ISSN 0915-3918

Honda R&D

Technical Review 2009

F1 Special (The Third Era Activities)

Technical Review

F1 Special

The Third Era Activities

2009

Honda R&D

RA106


HONDA
SHOWA
NTN
INTERCOND
NGK
SPARK PLUGS
LOOK RIGHT
MICHELIN
BBS

Special Issue – Honda's Third Formula One Era

On December 5, 2008, Honda announced its withdrawal from Formula One racing. For me personally, as for all the members of the Honda team directly involved in the Formula One projects and for everyone related to Honda racing, this was a great disappointment and represented a very difficult decision. Honda was not isolated from the global economic turmoil occurring at the time, and it was a period in which threats to the continued existence of the company were developing rapidly.

It is no exaggeration to say that Formula One was a symbol of the Honda brand, and the company's withdrawal from competition increased the sense of crisis throughout the Honda organization. This fact contributed significantly to the rapid implementation of a variety of crisis measures, and now, 10 months later, Honda is recovering somewhat more quickly than its competitors.

Honda's challenge during its third Formula One era was to compete and to win at the pinnacle of the world's motor races, and in doing so, to increase its technological capabilities, foster the development of its engineers, and boost its brand value. Above all, in our third era we set ourselves the tremendous goal of focusing not merely on the engine, but of achieving improvements in technologies throughout the vehicle.

While withdrawal from Formula One with the title within our grasp remains a great regret, I will be delighted if the discussions of the technologies that Honda developed during its period of competition, published in this special issue of the Technical Review, are read by a wide variety of specialists and prove useful in stimulating future advances in related areas.

Having achieved incredible development as the leading industry of the 20th century, the automotive industry is now confronted by a number of deep-rooted issues, including the reduction of CO_2 emissions, the implementation of measures to deal with the depletion of fossil fuel resources, and the determination of ways to improve the balance of supply and demand for resources on a global level. What is now demanded of Honda is for the company to demonstrate even greater energy and passion than it poured into Formula One in taking up the challenge of these global issues and carving out a new future for the automotive industry.

Takeo Fukui

Director and Advisor

CONTENTS

Descriptions of Chassis Technologies

Descriptions of Vehicle Control Technologies

Descriptions of Electronic Equipment Technologies

Descriptions of Material Technologies

Descriptions of Race Management Technologies

Activity Reports

Race Records

List of Companies Participating in Honda’s Third-Era Formula One Activities

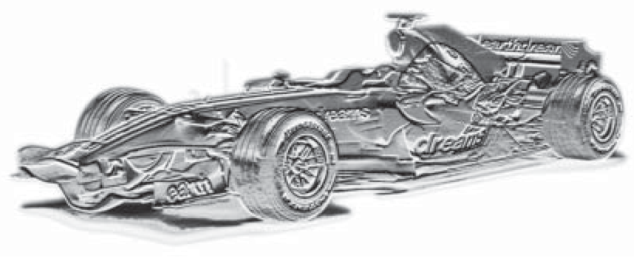
Comments from Project Members of Honda’s Third-Era Formula One Activities

< Prior Reports >

Fujita, H., Horiuchi, Y., Kubonoya, H., Baba, J., Sakai, K.: Development of Electric Power Steering System for F-1 Race Car, Honda R&D Technical Review, Vol. 14, No. 2, p. 45-50

Shimasaki, Y., Sakaguchi, J., Maki, H., Suzuki, H., Kato, A., Nishizawa, K.: Study on Combustion Monitoring System for Formula One Engines Using Ionic Current Measurement, Honda R&D Technical Review, Vol. 17, No. 1, p. 56-64

Development Overview



Looking back on Honda’s Third-Era Formula One Activities

Takeo KIUCHI*

In attempting to define the scope of the challenge that we were setting ourselves with regard to Honda’s third-era Formula One activities, we began from the standpoint of staking out a new area of challenge that would take us beyond the achievements of our second-era Formula One. We sought for a way of approaching racing that would differ from our second era, when we had focused on supplying engines. Our third-era activities got underway with us assuming a more comprehensive involvement not limited to engine supply, but also incorporating the development of vehicle chassis technologies and team management.

Viewed from this perspective, by contrast with our second-era activities our challenges extended beyond simply producing results in races. Basing ourselves in the UK and importing our own particular technologies, organizational management methods, and style of activities into a European framework can also be considered to be part of our new challenge, and overcoming the many, many issues that this generated might be said to be still another challenge that we set ourselves.

The main point of difference with our second era, however, was our full-fledged involvement in the area of developing chassis technologies. While Formula One vehicles are certainly cars, they are very different to mass-production vehicles, and the development of open-wheeled vehicles in which the major focus was on aerodynamic performance presented us with new technological challenges. In the area of total vehicle packaging, in addition to the basic issue – how to create a stiff but lightweight framework consisting of a monocoque, an engine, and a gearbox – it was also necessary to resolve other practical issues, namely how to lower the center of gravity and reduce the moment of inertia, and increase the degree of freedom of front-rear weight distribution. However, it is extremely difficult to reconcile these demands at a high level, and in reality development programs must balance them while achieving yearly performance advances. In proceeding in these areas in the initial stages, we formulated numerical targets indicating our goals for specific elements, but we could not predict how these would affect lap times.

No doubt our rival teams, with their long histories of development, understood how the changes made each year would affect lap times, and established priorities in moving ahead with the development of each element. A new team can only obtain this type of know-how from scratch, and we realized that this would be the area that would take the longest time.

Team management was another area in which Honda was inexperienced. In addition, the collaboration we established with a new team formed by a commercial production formula car constructor who had achieved success primarily in the US, which had been the result of our extended deliberations regarding the system that Honda should use in competing, had, while representing an ambitious endeavor, involved us in competitions with an management style preferred by European teams that had abundant Formula One experience and was hungry for victory. This was also a major issue to be addressed.

During our second era of Formula One, we had often been made aware of the differences between Japanese and US and European management styles - specifically, the difference between participation by every member of the team and an individual-focused, top-down approach that deployed specialists in each technological element. However, we had managed the issue successfully by completely separating areas of responsibility: the engines were Honda’s domain; and the vehicle chassis was the territory of constructors such as McLaren.

Bringing in the new element of joint development necessitated an entirely new and unprecedented management style: an equal fusion of styles based on different sets of values. There is a fundamental difference between personnel fostered in the survival strategies and cultures of companies working on contract, and those fostered in a culture in which outcomes are shared by everyone. This is perhaps a difference between responsibility that is shared among the group, and responsibility that is invested in the individual, and it is not possible to state categorically which is superior. When organizations of a similar scale come together on an equal basis, these differences will cause friction to arise at the points of contact between those organizations. In retrospect, this fact, and the effects of this friction, should have been more deeply considered in advance.

It may appear that it would have been best for us to simply proceed in our own way, but most companies, in the US and Europe at least, operate on a contract basis, and it was necessary for Honda, having for some time been expanding and engaging in global business

* Automobile R&D Center

activities, to also engage with this issue in the area of racing. However, we were unable to respond effectively, and as a result significant burdens were placed on the shoulders of those at the management levels of the practical development before development commenced, and it was not possible to organize a system that functioned at maximum efficiency.

On the technological front, our belief was that if the technology was excellent, it should be used regardless of whether it emerged from UK or Japanese research. However, for the UK side, if the technology was not based on an idea emerging from their group, it could not be applied. This was due to their contractual obligations, and they believed that they were not satisfactorily performing their duties if other ideas were used. Given that the engines were supplied by Honda, the effects of this were felt most strongly in the vehicle chassis domain, where we had proceeded on the basis of joint development, rather than in the area of engine development. Frequent conflicts arose between the Honda technical team, who were eager to engage in new challenges to taking victory, and the personnel on the UK side, who felt that the area of the vehicle chassis was their responsibility and sought to proceed on the basis of the knowledge they had attained through experience. Numerous adjustments were made in roles and modes of organization, but ultimately the only solution was to allow a culture of mutual understanding and respect to develop from a long involvement.

Control systems have advanced beyond conventional engine control, and we are now in an era of control systems that use vehicle parameters, such as traction control. Here also, a variety of efforts were made to achieve integrated engine and vehicle control, but some years had to elapse before we could overcome differences in concepts and measurement system environments, and bring together the efforts of both Japanese and UK sides.

We had gained experience of engine development in our second Formula One era, but in the ten years that had elapsed since Honda's withdrawal from factory works racing, the gap between our projected performance advancements and the technological level reached by the top manufactures had grown significantly. As a result, there was a considerable gulf in terms of performance between our team and our competitors when we first re-entered Formula One, and in particular the materials and design methods that we used seemed to be from another era in comparison with those employed by other teams.

Nevertheless, during our third era activities spanned a period of ten years, including the period of preparation, we overcame these issues and generated a significant number of fruits.

In the area of the engine, after we were able to analyze our competitors' high level of performance from a comparative perspective, within a short period we had reduced engine weight by 30 kg at the same time as boosting power by 50 kW. Our ideas and the results that they produced, as exemplified by a unified-head configuration, a single-piston-ring configuration, and IN EX camshafts geared at the front and rear ends, were generated by the coming together of the aspirations (the vision) of our engineers and technologies that made full use of recent 3D design technologies and precision machining equipment.

Our most sensational technology, and one which generated some astonishment in the industry, was a seamless shift that resolved the issue of shift-lag. This technological achievement had its beginning in the creation of an upshift mechanism, and was completed when we overcame issues in the area of downshift.

Aerodynamics-related development is, in certain senses, an area of race technology with which mass-production manufacturers are quite unfamiliar. In the development of standard mass-production vehicles, styling appearance is a major priority in the exterior design, and there is also, of course, a significant difference in the speeds at which production vehicles and Formula One vehicles are used. Because of this, we had accumulated virtually no experience in the development of vehicle forms to maximize Formula One aerodynamic performance. This was the area in which it took us the greatest amount of time to hone our skills, teasing out the intricacies of Formula One aerodynamics from basic theory until we reached the point at which we could confidently produce new vehicle shapes. The collaboration between engineers from Honda's aeronautics and styling design departments – in a certain sense, rather dissimilar fields – generated new designs and new methods of analysis, leading finally to the application of aero parts that would take us beyond the top teams.

Aerodynamic performance was a high priority, and even as we were constrained by the necessity of generating results in this area, the achievement of weight reductions and the development of chassis resulted in the creation of numerous analytic and measurement technologies, producing superior configurations and superior performance.

The development of the KERS (a hybrid system), scheduled to be introduced from 2009, succeeded in producing a compact, high-performance system within extremely restricted space and weight allowances in comparison to a mass-production vehicle. Regrettably, due to Honda's withdrawal from Formula One, the system could not be used in racing. However, we were able to test the system and demonstrate its effectiveness before our competitors, and we were able to construct a safe system before many of the other teams –also noteworthy accomplishments.

As a result of these efforts, we achieved 2nd place in the Constructors' Championship in 2004, and were victorious in the Hungarian Grand Prix in 2006. However, while the element technologies described above individually produced results, as a team Honda unfortunately did not achieve results that put us on par with the top teams.

In the world of racing, victory will not be achieved unless all of the many elements essential to competition are functioning at their top levels. As I have indicated above, we developed individual element technologies that boosted performance when applied, but were unable to coordinate the timing of these performance increases. In an arena in which, no matter how superb the element technology that is developed, competitors catch up instantly when the technology is employed in the field, what is necessary is to simultaneously develop and continually evolve an array of outstanding technologies. Our inability to supply technologies in this way, as well as our failure to maintain the lead position of superior element technologies until other elements could be supplied, remain as significant issues for us. However, the vehicle formed from the sum of these technologies displayed excellent performance in the 2009 season for the team that took over Honda racing team after its withdrawal. I can well imagine how this makes the members of the development team feel, and I hope that they are confident that the goals they formulated for themselves were the right ones, and that they had succeeded in expressing those goals in the form of technologies.

In the course of Honda's third Formula One era, numerous staff members left Japan to base themselves in the UK, separating themselves from the well-organized development environment that had, in a certain sense, been formed in our R&D centers. In particular, many young engineers, relying on their own skills, went head-to-head on an equal footing with engineers from other countries, and in this process, their levels of skill were recognized, and accordingly developed. A simple comparison is impossible, of course, but I feel that the burdens shouldered by the members of the development team were greatest in our overseas activities during our third Formula One era, when we failed to achieve our desired outcome. But I think that for exactly that reason, the growth that they experienced was also that much greater.

The outcomes of the efforts of the team have now been left behind, and it is my hope that all of our next generation of engineers, and in particular those I have discussed above, will mount new challenges in the near future in which they will reclaim those outcomes.

In conclusion, I would like to applaud and to thank all members of the Honda team involved in Honda Formula One projects, their families, and the staff members of cooperating makers for their tremendous efforts and their support over an extended period. Thank you.

■ Author ■

Takeo KIUCHI

Transition of Regulation and Technology in Formula One

Hiromasa TANAKA*

ABSTRACT

The Formula One regulations are established with consideration of fairness, competition, safety, sustainability, and entertainment value.

In order to ensure free technological competition, the technical rules of Formula One, under which constructors are obliged to compete using original vehicles, are essentially free, other than specifying minimum basic items.

From the beginning of the 1990s to 2009, despite continuous changes in the regulations to increase vehicle performance control and safety, lap times improved almost every year due to the development of technologies that exceeded regulation stipulations. As a result of these battles, the letter count of the Formula One technical regulations has increased more than three-fold over 19 years.

Detailed technological development spanning a broad range of elements and involving the use of computers, in addition to a comprehensive management approach that brings these elements together, are essential to the Formula One of the 2000s. In recent years in particular, Formula One has responded to rapid social change, for example with measures to cut costs in line with worsening global economic conditions, and the development of environmental technologies, as exemplified by responses to global warming, in addition to measures to highlight competition – the essence of Formula One racing – and to increase the spectacular nature of the races. This is a period in which Formula One is reflecting upon the very meaning of its existence.

1. Introduction

Teams entered in the Formula One World Championships are basically required to race their own original vehicles, and are termed “constructors.” Formula One racing, which commenced in 1950, encompasses both a driver’s championship, in which drivers compete to determine who is the world’s fastest, and since 1958 a constructor’s championship, in which constructors compete to determine which of their vehicles is the world’s fastest. However, the production of engines presents a considerable challenge to any participant other than an automaker or a specialized engine manufacturer that possess specialized technologies. In the case of engines, Formula One regulations permit acquisition or purchase from an external engine constructor. Therefore, in the naming of teams, the name of the chassis comes before the name of the engine, and the constructors’ title is awarded only to the chassis.

In the racing world, the term “works” refers to the fact that a maker itself manufactures the racing cars, racing engines, and the like, or manages the team. Divided into first, second, and third eras, Honda’s Formula One activities were organized as follows:

(1) First era (1964-1968): Raced under the Honda name as a full works team, including engine construction, chassis construction, and team management

(2) Second era (1983-1992): Supplied works engines as an engine constructor to chassis constructors including Spirit, Williams, Lotus, McLaren, and Tyrell

(3) Third era (2000-2008):
- 2000-2005: Supplied works engines to and conducted joint chassis development with the chassis constructor British American Racing (BAR); raced as BAR Honda
- 2006-2008: Raced as Honda Racing F1 Team (HRF1) as a full works team, including engine construction, chassis construction, and team management

In parallel with these activities, Honda also supplied engines in 2001 and 2002 to Jordan Grand Prix, and from 2006 to 2008 to SUPER AGURI F1 TEAM (SAF1).

For almost the entire period between Honda’s second and third Formula One eras, from 1994-2000, the company supported technical development efforts by MUGEN (now M-TEC), a company that supplied engines as an engine constructor, and was involved in Formula One as part of MUGEN-Honda.

This paper will first discuss the characteristics of the

* Automobile R&D Center

Formula One regulations, and will then go on to consider the main changes in Formula One regulations and technologies employed from the beginning of the 1990s, the last phase of Honda's second Formula One era, to 2008, the last year of the company's third Formula One era. It will also look at the changes in Formula One regulations and technologies up to 2009, for which Honda engaged in vehicle development, but ultimately did not compete due to the company's withdrawal from Formula One.

2. Characteristics of Formula One Regulations

Formula One regulations are based on the FIA International Sporting Code, which provides shared rules for all motor sports, and incorporate sporting regulations and technical regulations unique to Formula One championship[(1)]. The sections below will discuss the characteristics of these regulations, with a focus on technical regulations that directly affect technical development for Formula One.

2.1. Basic Principles and Structural Conflicts

Formula One involves competition between vehicles built from scratch by constructors. To help ensure competition, absolute prohibitions and mandatory observances have been minimized in the technical regulations, and constructors are free to approach any point not explicitly covered in the regulations as they choose. For example, there are no rules concerning the wheel base, and constructors may employ a wheel base of any length. One can associate the "Formula" in Formula One with the minimum standards in the regulations.

This basic concept is actually the antithesis of the concept of production car racing. In production car racing, a homologated mass production vehicle that has satisfied specific requirements regarding the number of units produced is used as a base, and only modifications that are specified in the regulations can be made to the vehicle.

Formula One racing is a contest of speed, and the performance of the vehicles therefore tends to improve year by year. To help ensure that the performance of the vehicles does not exceed the limits of the capacity of the circuit safety facilities such as run-off areas and guard rails, the rules of technical regulations are continuously enhanced in order to increase the safety of the vehicles themselves and to impose restrictions on vehicle performance. However, in order to take victory in a fierce competition, each team pursues technical development to the very limits of the regulations. Clear standards and accurate methods of judgment are therefore necessary for regulations designed to help enable fair and consistent judgment of the outcomes of this competition.

To this end, the regulations themselves should be finely subdivided and highly detailed. However, once highly detailed regulations have been formulated, even a slight deviation from the narrow scope they define enables developers to attain a freedom in their development just because it is not subject to the regulations. As a result of teams pursuing development of this type that attempts to sidestep the regulations, and of the continuing formulation of increasingly detailed regulations in an attempt to control it, the Formula One technical regulations increased in volume from approximately 40000 letters covering 65 items in 16 chapters in 1990 to approximately 86000 letters covering 140 items in 21 chapters in 2000, the first year of Honda's third Formula One era. By 2008, the final year of the company's third Formula One era, this had risen to approximately 122000 letters covering 157 items in 21 chapters.

A description of the detailed mechanism of formulation and amendment of Formula One regulations can be found in another paper[(1)], but in terms of Formula One history, when a regulation has been clearly deficient, the principle has been that the relevant item will only become prohibited following the publication of a formal codicil and the completion of the amendment procedures, and until that time teams are able to use it to their advantage. However, the right to specify interpretations within a scope in which no modification of the text of the regulations is necessary belongs to the Federation Internationale de l'Automobile (FIA) at any time, and the organization issues technical instructions called Technical Directives (TD) to all Formula One teams as necessary. Other than this, the FIA also has the authority to permit the Stewards of the Meeting to exclude vehicles that are considered dangerous from race events. Objections to a judgment at a race event may be presented before the International Court of Appeal (I.C.A.), the final court of appeal for motor sports.

2.2. Example of a Conflicting Interpretation

The 2005 regulations specified the following points: First, the regulations defined vehicle weight as the weight of the vehicle including the weight of the driver in full racing apparel, but did not indicate whether or not the fuel was included in this figure. The regulations further specified that the weight of the vehicle was to be no less than 605 kg during the qualifying practice session, and no less than 600 kg at all other times during a race event. Ballasts could be employed to reach the minimum weight, but the regulations specified that a ballast should be secured in a manner that necessitated the use of tools for its removal, and that it should be possible to affix seals if deemed necessary. Further, during the race, no substance may be added to the vehicle other than fuel and compressed gases.

Since 1994, it has been customary in Formula One to verify the weight of the vehicle with the fuel removed in the car inspection following the completion of the race, in order to help stop teams from attaining an advantage by running under the minimum weight during the race and bringing the vehicle up to the minimum weight after the race by adding fuel in the closing stages of the race. However, there was insufficient consensus between the FIA and the teams regarding the level to which fuel should be removed. In actual fact, to completely remove all fuel from the fuel system would necessitate dismantling part of the engine, which was unrealistic on the circuit. The teams therefore considered that under normal circumstances it was acceptable to include the remaining unusable fuel in the vehicle weight.

However, an issue arose concerning two BAR Honda vehicles that took 3rd and 5th place in the San Marino Grand Prix, the 4th round of the 2005 season. At the time, BAR Honda vehicles employed a sub fuel tank to help maintain a stable fuel supply pressure. The configuration was such that the fuel in the sub fuel tank could not be easily removed using ordinary methods. It was reported that when one of the vehicles was reweighed during the car inspection following the race after all fuel from the sub fuel tank and all remaining fuel from the main fuel tank had been drained, it was found to be 5.4 kg under the regulation minimum weight of 600 kg. Based on the report of the outcome of the car inspection, the Stewards of the Meeting interviewed the people concerned, finally upholding the original race result. However, the FIA brought an objection to the I.C.A.

The I.C.A. considered that it could not be physically proven that the vehicle had not been run during the race below the 600 kg minimum vehicle weight, and that insufficient evidence was provided by the combination of data and theory. In addition, it upheld the interpretation that because the use of fuel as ballast did not correspond to the rules concerning ballast, it could not be allowed. However, if this interpretation of ballast was followed, then oils and cooling water would also not be included in the minimum weight of the vehicle. In actual fact, however, vehicles are weighed following races containing oils and cooling water. Because addition of oils or cooling water during the race is not allowed, they cannot be used to bring the vehicle up to the minimum weight, and they are therefore customarily considered to be included in the vehicle weight.

The final verdict of the I.C.A. was that while it could not be proven that the team had deliberately set out to commit fraud, the team’s actions at the time that the fuel tank was drained for inspection, and their failure to request a clarification of the interpretation of the relevant regulation in advance, demonstrated negligence on their part. The two BAR Honda vehicles were disqualified from the San Marino event and the team was suspended from the next two events in the championship. The team was also suspended for a further period of six months, but this penalty was suspended for one year.

In Formula One considered as a sport, whether or not an inspection is conducted, competitors are regarded as having sworn that their vehicles are in compliance with regulations when they participate in a race. The onus of proving this rests on the teams. The authority to interpret the regulations is stipulated as resting with the FIA, the entity that is authorized to conduct the competition, and this authority should be respected. The author therefore has no intention of arguing with the I.C.A. ruling. However, a clause was added to Section 6.6. of the technical regulations in 2008 stipulating that all competitors should provide a means of removing all fuel from the vehicle, and this can be seen as a means of more certainly heading off any disagreement over the interpretation of the regulation.

3. Changes in Formula One Regulations

This section will discuss changes in Formula One regulations and technologies (Table 1), while providing some examples of development within each area of technology. For details of specific technologies, please refer to the relevant paper in this special issue of the Technical Review.

3.1. Engine and Powerplant

Regulations concerning the engine have been continuously revised in order to limit the performance of Formula One vehicles. In 1995, the previous maximum displacement of 3500 cm^3 was reduced to 3000 cm^3, and this figure was further reduced to 2400 cm^3 from 2006. Until 1999, the maximum number of cylinders that could be employed was twelve. However, as a result of a technical trend that valued the overall balance of the vehicle over the performance of engine unit teams gradually moved towards the use of ten cylinders, and by the time the regulations specified ten as the maximum number of cylinders in 2000, all Formula One engines already employed this number of cylinders. In 2006, together with the reduction in maximum engine displacement, the maximum number of cylinders was reduced to eight.

While it cannot necessarily be considered ideal from the perspective of seeking increased engine power, from the mid-1990s teams proceeded to increase the temperature of the cooling water in their engines, in order to reduce the aerodynamic drag generated at the openings for the cooling system. In order to put a stop to this trend, from 1999 the regulations specified a maximum cooling water pressure of 3.75 bar.

Maximum engine speed reached about 20000 rpm in 2006. While it had previously not been treated in the regulations, from 2007 maximum engine speed was limited to 19000 rpm, and from 2009 to no more than 18000 rpm. To help enable the vehicles to be run faster at a restricted engine speed, there was a trend towards an increased frequency of use of the high engine speed range, and this made developments aimed at increasing durability even more necessary.

In order to reduce costs by limiting changes to engine specifications and the number of engines used in a year, a (long mileage) regulation was introduced stipulating a minimum use distance for the engine. Previously, constant engine changes had been allowed during a three-day race event, but in 2004, regulations obliged teams to use one engine per race event. The following year, more stringent conditions were applied, with regulations stipulating that one engine should be used per

Table 1 Transition of F1 regulations

<table>
<tr><th colspan="2"></th><th colspan="2">1990</th><th colspan="2">1991</th><th colspan="2">1992</th><th colspan="2">1993</th><th colspan="2">1994</th><th colspan="2">1995</th><th colspan="2">1996</th><th colspan="2">1997</th><th colspan="2">1998</th><th colspan="2">1999</th><th colspan="2">2000</th><th colspan="2">2001</th><th colspan="2">2002</th><th colspan="2">2003</th><th colspan="2">2004</th><th colspan="2">2005</th><th colspan="2">2006</th><th colspan="2">2007</th><th colspan="2">2008</th><th colspan="2">2009</th></tr>
<tr><td rowspan="10">Engine and powerplant</td><td>Supercharging</td><td colspan="40">Prohibited from 1989</td></tr>
<tr><td>Capacity</td><td colspan="10">3500 cm^3 maximum</td><td colspan="22">3000 cm^3 maximum</td><td colspan="8">2400 cm^3 maximum</td></tr>
<tr><td>Number of cylinders</td><td colspan="20">12 maximum</td><td colspan="12">10</td><td colspan="8">8</td></tr>
<tr><td>Homologation or limitation of specifications</td><td colspan="32">Free</td><td colspan="2">Limitations on fundamental dimensions, weight, and materials</td><td colspan="6">Homologated</td></tr>
<tr><td>Mileage</td><td colspan="28">Free</td><td colspan="2">1 engine for 1 race event</td><td colspan="4">1 engine for 2 race events</td><td colspan="4">1 engine for 2 race events (Excluding FP1 & FP2)</td><td colspan="2">8 engines for 1 season</td></tr>
<tr><td>Rev limit</td><td colspan="34">Free</td><td colspan="4">19000 rpm</td><td colspan="2">18000 rpm</td></tr>
<tr><td>Inlet systems</td><td colspan="32">Free</td><td colspan="8">Variable geometry prohibited</td></tr>
<tr><td>Cooling system water pressure</td><td colspan="18">Free</td><td colspan="22">3.75 bar maximum</td></tr>
<tr><td>Hybrid systems</td><td colspan="18">Free</td><td colspan="20">Prohibited</td><td colspan="2">KERS permitted</td></tr>
<tr><td>Refueling during race</td><td colspan="8">Prohibited</td><td colspan="32">Free</td></tr>
<tr><td rowspan="2">Fuel</td><td>Properties</td><td colspan="4">–</td><td colspan="2">Specific gravity controlled</td><td colspan="12">Type used by general public</td><td colspan="8">Commercial fuel or fuel in development for future commercial use</td><td colspan="10">Commercial fuel</td><td colspan="4">Commercial fuel 5.75% (m/m) biofuel</td></tr>
<tr><td>Sampling</td><td colspan="10">–</td><td colspan="30">Sampling at circuit
Approval before use</td></tr>
<tr><td rowspan="6">Gear box</td><td>Automatic gear changing</td><td colspan="23">Prohibited</td><td colspan="5">Free</td><td colspan="12">Prohibited</td></tr>
<tr><td>Forward gear ratios</td><td colspan="8">Free</td><td colspan="28">Minimum: 4; maximum: 7</td><td colspan="4">7 maximum</td></tr>
<tr><td>Continuously variable transmission</td><td colspan="23">Free</td><td colspan="17">Prohibited</td></tr>
<tr><td>Clutch</td><td colspan="24">Free</td><td colspan="16">Twin clutch prohibited</td></tr>
<tr><td>Specification limitations</td><td colspan="36">Free</td><td colspan="4">Limitations on fundamental dimensions and materials</td></tr>
<tr><td>Mileage</td><td colspan="36">Free</td><td colspan="4">1 gear box for 4 race events (Excluding FP1 & FP2)</td></tr>
<tr><td>Minimum weight of car</td><td>Machine weight</td><td colspan="2">500 kg (without driver)</td><td colspan="8">505 kg (without driver)</td><td colspan="2">595 kg (with driver)</td><td colspan="8">600 kg (with driver)</td><td colspan="8">600 kg (with driver)</td><td colspan="6">605 kg for qualifying (with driver)
600 kg for others (with driver)</td><td colspan="6">605 kg (with driver)</td></tr>
<tr><td rowspan="2">Chassis</td><td>EPS</td><td colspan="24">Free</td><td colspan="16">Prohibited</td></tr>
<tr><td>Ballast</td><td colspan="20">–</td><td colspan="20">Moving ballast prohibited and area of skid block fasteners limited</td></tr>
<tr><td rowspan="12">Bodywork and dimensions</td><td>Overall width</td><td colspan="6">2150 mm maximum</td><td colspan="10">2000 mm maximum</td><td colspan="24">1800 mm maximum</td></tr>
<tr><td>Facing ground</td><td colspan="10">Flat bottom</td><td colspan="30">Stepped bottom</td></tr>
<tr><td>Rear wing maximum height</td><td colspan="6">1000 mm above ground</td><td colspan="4">950 mm above ground</td><td colspan="28">800 mm above reference plane</td><td colspan="2">950 mm above reference plane</td></tr>
<tr><td>Rear wing maximum width</td><td colspan="38">1000 mm</td><td colspan="2">750 mm</td></tr>
<tr><td>Rear maximum overhang</td><td colspan="2">600 mm</td><td colspan="26">500 mm</td><td colspan="12">600 mm</td></tr>
<tr><td>Rear center diffuser maximum width</td><td colspan="10">1000 mm</td><td colspan="28">300 mm</td><td colspan="2">1000 mm</td></tr>
<tr><td>Rear diffuser maximum height</td><td colspan="36">Free</td><td colspan="2">125 mm maximum (without center area)</td><td colspan="2">175 mm maximum</td></tr>
<tr><td>Front wing maximum overhang</td><td colspan="2">1200 mm</td><td colspan="4">1000 mm</td><td colspan="32">900 mm</td><td colspan="2">1000 mm</td></tr>
<tr><td>Front wing maximum width</td><td colspan="2">1500 mm</td><td colspan="36">1400 mm</td><td colspan="2">1800 mm</td></tr>
<tr><td rowspan="2">Front wing height</td><td colspan="10">Up to height of front wheel rims (end plates)</td><td colspan="12" rowspan="2">Between 50 mm and 250 mm above reference plane</td><td colspan="8" rowspan="2">Between 100 mm and 300 mm above reference plane</td><td colspan="8" rowspan="2">Between 150 mm and 350 mm above reference plane</td><td colspan="2" rowspan="2">Between 75 mm and 275 mm above reference plane</td></tr>
<tr><td colspan="2">Free</td><td colspan="6">At least 25 mm above reference plane</td><td colspan="2">At least 40 mm above reference plane</td></tr>
<tr><td>Enforcement of other bodywork limitation and deflection test</td><td colspan="10"></td><td colspan="2">No body work area around front wheel</td><td colspan="2">Height reductions just front of rear wheel</td><td colspan="4"></td><td colspan="2">Side wing prohibited
Front wing test (downward)
Rear wing assembly test (rearward)</td><td colspan="2"></td><td colspan="2">Test just front of rear wheel (downward)</td><td colspan="2"></td><td colspan="2">Splitter test (upward)</td><td colspan="2">Rear wing top element test (rearward)</td><td colspan="2">Rear wing front element test (downward)</td><td colspan="2"></td><td colspan="2">Rear wing element bridge mandated</td><td colspan="2"></td><td colspan="2">Designated front center wing</td></tr>
<tr><td rowspan="3">Tires</td><td>Tire width</td><td colspan="6">18 inch maximum</td><td colspan="10">15 inch maximum</td><td colspan="2">Front 305-380 mm
Rear 365-380 mm</td><td colspan="22">Front 305-355 mm
Rear 365-380 mm</td></tr>
<tr><td>Tire tread</td><td colspan="16">Slick</td><td colspan="2">Grooved
Front-3
Rear-4</td><td colspan="20">Grooved
Front-4
Rear-4</td><td colspan="2">Slick</td></tr>
<tr><td>Tire supplier</td><td colspan="18">–</td><td colspan="4">Single supplier</td><td colspan="12">–</td><td colspan="6">Single supplier</td></tr>
<tr><td rowspan="5">Electronically controlled devices</td><td rowspan="2">Traction control</td><td colspan="10" rowspan="2">Free</td><td colspan="13">Prohibited</td><td colspan="13" rowspan="2">Free</td><td colspan="4" rowspan="2">Prohibited (Single ECU)</td></tr>
<tr><td colspan="4"></td><td colspan="9">Software validation in advance</td></tr>
<tr><td rowspan="2">Launch control</td><td colspan="10" rowspan="2">Free</td><td colspan="13">Prohibited</td><td colspan="5" rowspan="2">Free</td><td colspan="12" rowspan="2">Prohibited
(FIA logger observation)</td></tr>
<tr><td colspan="4"></td><td colspan="9">Software validation in advance</td></tr>
<tr><td>Chassis devices</td><td colspan="8">Free</td><td colspan="32">Prohibited</td></tr>
<tr><td rowspan="2">Materials</td><td>Specific modulus</td><td colspan="20">–</td><td colspan="2">40 GPa/(g/cm^3) limit for engine moving parts</td><td colspan="18">40 GPa/(g/cm^3) limit for all parts</td></tr>
<tr><td>Material designation</td><td colspan="32">–</td><td colspan="8">Designated for engine parts</td></tr>
<tr><td rowspan="4">Safety</td><td>Frontal crash test speed</td><td colspan="10">11 m/s</td><td colspan="8">12 m/s</td><td colspan="2">13 m/s</td><td colspan="14">14 m/s</td><td colspan="6">15 m/s</td></tr>
<tr><td>Side impact test</td><td colspan="10">–</td><td colspan="30">Mandated</td></tr>
<tr><td>Side panel penetration test</td><td colspan="20">–</td><td colspan="2">Specific designation</td><td colspan="18">Mandated</td></tr>
<tr><td>Suspension tether</td><td colspan="18">–</td><td colspan="4">Single tether</td><td colspan="18">Double tether</td></tr>
</table>

two race events (from 2007, the Friday free practice (FP) session was excluded from the regulation). In 2009, the total number of engines that a team could use per year was limited to eight.

From 2006, in order to avoid cost increases resulting from excessive development competition, common basic dimensions for engines were established, and minimum weight and center of gravity of an engine were specified. Then, from the next year, 2007, a system of engine authorization (homologation), under which major engine specifications could not be modified once they had been registered, was introduced one year earlier than initially scheduled. At the same time, the use of variable geometry intake manifolds, previously allowed, was prohibited. The introduction of this homologation system represented a significant change in orientation for the Formula One philosophy, which as indicated above is based on free technical competition. Broad-ranging development efforts, seeking even minor enhancements of basic performance, were concentrated in the period immediately before the submission deadline for engine homologation applications, but following this, engine development was largely limited to subtle modifications within the allowable scope or tuning of characteristics.

In order to more thoroughly embody the way of thinking that sources of drive power other than engines should not be basically allowed, and to prohibit the hybrid systems that some teams were said to be using, in 1999, power sources other than 3000 cm^3 engines were prohibited, and the maximum recoverable stored energy was restricted to 300 kJ (the maximum stored energy recoverable at a rate greater than 2 kW was restricted to 20 kJ). From 2009, however, the use of a type of hybrid system called a Kinetic Energy Recovery System (KERS) has been allowed with the intention of promoting the development of environmental technologies through the medium of motor sports. These are the only systems excluded from the regulations discussed above. The maximum energy that can be input to or output from a KERS is 60 kW, and a maximum of 400 kJ can be recovered per lap.

Refueling during a race was prohibited for some time, but this prohibition was lifted from 1994, focusing attention on pit stop timing and the amount of refueling as aspects of race strategy. In line with this, technologies associated with aircraft refueling systems were applied, and from 1995 regulations stipulated the use of identical refueling systems able to safely and rapidly supply at a fixed refueling speed.

With regard to the composition of the fuel employed, while the interpretation that Formula One fuels should fundamentally be based on commercial fuels was upheld in the 11th round of the 1992 season(2), from 1999 to 2002, detailed rules concerning the ratio of various hydrocarbons in the fuel made it possible to use fuels that had been developed for the purpose of enhancing commercial fuels in the future. In addition, as a response to worldwide efforts to ameliorate global warming, from 2008, the regulations have stipulated the use of fuels containing a 5.75% ratio of biofuels.

3.2. Gearbox

A semi-automatic gearbox based on a sequential shift mechanism developed in the early 1990s, by means of which gears are shifted using a paddle attached to the steering wheel, has been continuously used as the standard Formula One gearbox type. As will be discussed below, from mid-2001, a significant relaxation in the regulations concerning electronically controlled systems temporarily enabled the use of automatic shifting systems that were not operated by the driver, but regulations were tightened to once again prohibit these systems in 2004 (as part of driver aids ban) over concerns that they would obscure differences in levels of skill between the drivers.

With regard to the number of forward gear ratio, there has basically been no change since the regulations stipulated a maximum of seven in 1994, but a prohibition on the use of continuously variable transmissions was added from the fifth round of the 2001 season. Following this, in 2002, regulations prohibited dual-clutch transmissions, which enable uninterrupted switching to the next pre-engaged gear stage. It was considered that a de facto prohibition existed on shift mechanisms that produced no discontinuity in torque, the ideal form of gearbox, but in 2005, BAR-Honda introduced a seamless shift using a one-way mechanism that the team had developed based on a close study of the regulations. Other teams sought to keep pace, and this has now become the standard technology in use in Formula One.

Since the 1980s, every team has employed original designs and methods of manufacturing gearbox case, which is a structural element of the chassis to which the suspension is mounted. In the first half of the 1990s, most gearbox cases were cast from magnesium, but after this a variety of methods came into use in attempts to produce lightweight, high-rigidity cases, including the manufacture of the cases from thin-walled titanium castings and welded plates, and the use of CFRP reinforcing. In 2004, BAR-Honda became the first team to produce a practically-applicable all-CFRP gearbox case.

With regard to the gears and shafts inside the gearbox, even in the early 2000s it was still standard practice, with some exceptions, to employ existing parts manufactured by gearbox makers. However, teams began to develop their own original gears as development competition seeking weight savings and greater compactness intensified. Based on concerns over spiraling costs, from 2008 the regulations have stipulated that forward gears should be manufactured from iron, and have also specified minimum gear width, gear weight, distance between shaft centers, and the like.

In 2008, the long-mileage concept was also applied to gearboxes, and a single gearbox is now required to be used for four race events. However, the adjustment of the gear ratios to match the characteristics of specific circuits and the replacement of the dog rings is allowed conditionally.

3.3. Chassis

In line with advances in electronic chassis control technologies in production vehicles, at the beginning of the 1990s the application of a variety of technologies was tested, including active suspension, traction control, four-wheel steering, and anti-lock brakes. Semi-active suspension had a particularly significant effect, enabling the realization of ride heights control that produced stable aerodynamic performance. However, based on considerations of performance control, and to control cost increases and ban driver aids, electronic chassis control systems were prohibited in 1994.

Power steering had been used in Formula One since the early 1990s, in order to respond to the increased steering force resulting from advances in aerodynamics and increased tire grip. From 1994, the use of this system was limited to the provision of assistance to the driver's physical effort. Initially, electronic power steering (EPS) systems were allowed if they satisfied this condition, and from the 12th round of the 2000 season through 2001, Honda led other teams in using a Formula One EPS system in racing[(3)]. However, because of the challenge represented by the verification of and judgment on the details of the control, making it possible that the systems incorporated control that functioned as a driver aid, the use of EPS was entirely prohibited from 2002, and hydraulic power steering systems are now used exclusively in Formula One.

The minimum weight of the vehicle was initially specified in regulations without the driver, and was 500 kg in 1990, increasing to 505 kg in 1993. From 1995 the driver was included in rules regarding minimum weight. In 1995 the figure was 595 kg, increasing to 600 kg in 1996. In 2004 the minimum weight was increased to 605 kg for the qualifying practice only, and in 2007 this was made the minimum weight for the entire race event. Compared to the previous circumstances, the minimum weight figures have not changed significantly in themselves.

In 1993, the maximum width of the vehicle, including the tires, was reduced from the previous 2150 mm to 2000 mm. The figure was further reduced to 1800 mm in 1998. In the case of formula cars, this directly means a reduction in track, and this regulation was applied in order to reduce the cornering limit speed of the vehicles by increasing load transfer on the left and right tires. While reducing the weight of their vehicles, teams began to make efforts to lower the center of gravity by even a small amount and thus increase the cornering limit speed, positioning corresponding ballasts, manufactured from a tungsten alloy with a high specific gravity, as low as possible in the vehicle. As a result of these efforts, the top teams came to employ ballasts weighing 70 kg or more. This enabled the front-rear weight distribution to be maintained towards the front of the vehicle while helping to ensure the dimensions of the front half of the vehicle (the distance between the front wheels and the side pontoons) necessary from an aerodynamic perspective, which was also effective in increasing warming of the tires and boosting the dynamic performance of the vehicle. In 2000, from considerations of safety, the regulations banned the use of movable ballasts and limited the area of the skid block fastener, which is used as the lowest ballast on the vehicle.

In an attempt to significantly lower downforce, which had previously been continuously increasing, and thus to reduce the cornering limit speed, in 1995, flat bottom rule was replaced by a stepped bottom rule that would increase the distance between the bodywork facing the ground and the road surface, and the use of skid plates that would increase the ground clearance was also stipulated. However, a high nose configuration that increased downforce by encouraging air flow beneath the vehicle had been developed in 1990, and this configuration was applied by all teams in 1996. Efforts were also made to increase the contribution of the shape of the front suspension to vehicle aerodynamics. A twin keel mount was introduced in 2002 and a zero keel configuration was introduced in 2005. Nevertheless, since the front suspension stroke is extremely small in Formula One vehicles, the tendency has been to emphasize rigidity over geometry.

In parallel with this, from about 1991, the regulations concerning the dimensions of the front and rear wings and the diffusers positioned in the rear of the bodywork facing the ground, which were making an increased contribution to aerodynamic performance, began to be reviewed. Since 2000, the regulations have been made more stringent on an almost annual basis. In addition, given various advancements in aerodynamic components such as vortex generators, bargeboards, and winglets, designed to increase the overall downforce of the vehicle by adjusting the flow of air around different parts of the body, a large number of additions have been made to regulations concerning the dimensions of parts of the vehicle body.

In addition, as a result of the use of flexible wings to reduce aerodynamic drag at high speed, since 1999 regulations have stipulated measurement of the level of deformation of each part of the vehicle body and the fitting of stiffening elements, among other requirements.

The previously used slick tires, which had no grooves, were replaced by grooved tires from 1998, in order to reduce the absolute level of grip. In 1999, one extra groove was added at the front of the tires for a total of four front grooves and four rear grooves. This configuration was employed until 2008. The stiffness of the tread rubber of grooved tires is low; they are prone to delay in response and present issues of controllability. In addition, they are also challenging to handle, with the blocks between the grooves being deformed to incline and abrasion termed graining tending to occur commencing from the edge. Appropriate temperature management of the tires is necessary for the effective use of these grooved tires. At the same time, it is also necessary to increase controllability, maintaining the overall vehicle balance and a minute transient response characteristic. In 2009, there was an extensive review of aerodynamic issues, and aerodynamic regulations were introduced based on the concept of lowering cornering performance and reducing the effect of the wake of the vehicle in front, in order to enable active overtaking. At the same time, high-controllability slick tires were reintroduced.

3.4. Control

It is the fate of control systems in Formula One to be constantly subject to regulations from the perspective of driver aids ban. It is more challenging to regulate engine-based torque control systems such as traction control than chassis-based control systems, in the case of which it is comparatively simple to make a judgment based on whether a device is fitted or not. Since 1997, a software validation in advance has been mandatory, but even if the FIA inspections were to confine themselves to a hotel room with the team members for several days for each team in order to check source code, it would still not be possible to achieve complete restriction. Perhaps because regulations relating to this area had reached their limit of operability, from the 5th round of the 2001 season, regulations were abolished for all forms of software excluding safety-related software, and engine torque control became effectively free.

This did not only affect traction control. Development has also proceeded in other areas, including engine brake control and over-run control, which achieves a type of ABS effect by using engine torque during braking to help keep the rear wheel brakes from locking.

Launch control, which increases acceleration by making maximal use of the tire grip at race start, which changes moment by moment, also advanced considerably. However, from 2004, launch control was prohibited through detailed regulation of the method of using the engine and the clutch, and the introduction of a method of using standardized FIA data loggers which should be fitted to vehicle to monitor the way of using them.

Following this, from 2008, torque control, including traction control, was once again prohibited because the requirement to fit a standardized FIA ECU to vehicles made it possible for the first time to accurately manage the regulation.

3.5. Materials

There had previously been few rules in the regulations concerning materials of formula one vehicles, but attention was directed to them when the use of beryllium alloys, which are harmful to human health, became an issue, ultimately leading to restrictions on the use of expensive lightweight and high-rigidity materials. In 2000, materials with a specific modulus of elasticity of greater than 40 GPa (g/cm^3) were prohibited from use in any part other than the moving parts of the engine, and in 2001 this prohibition was extended to the entire vehicle. In 2003, methods of testing compliance were detailed.

While materials development programs had previously been focused exclusively on the achievement of low weight and high rigidity, in response to these changes, fine control began to be applied to the composition and process of production of materials, and more advanced technologies were pursued, for example by developing surface treatments and conducting research at the molecular level, in order to realize desirable materials that are able to achieve high performance levels while remaining compliant with regulation rules concerning composition and physical values.

Since 2006, detailed regulations have been in effect concerning the types of materials that can be used for each engine part, and development has continued within the scope defined by the regulations.

3.6. Safety

The deaths of Roland Ratzenberger and Ayrton Senna in accidents in 1994 resulted in a large-scale revision of the regulations seeking a rapid enhancement in passive safety. Continuous changes to the regulations have also been made since then, and as a result no Formula One driver has been killed in an accident in the intervening 15-year period. More details can be found in two previously published papers[(1),(4)].

In the opening round of the 2001 season, the BAR-Honda vehicle being driven by Jacques Villeneuve ran head-on into another vehicle and became airborne before suffering a severe crash. Protected by a robust carbon monocoque, Villeneuve escaped with almost no injuries. Tragically, however, a wheel torn off the vehicle struck and killed a track official. Since 1999, regulations had already stipulated the use of suspension tethers between the uprights and the chassis in order to help reduce this type of accident, and standards have been raised each year. However, given the fact that the achievement of complete safety is not a realizable goal, the regulations have continued to be enhanced since the following year.

4. Trends in Formula One Regulations and Their Effects

Formula One regulations are formulated based on considerations of fairness, competitiveness, safety, sustainability, and entertainment value[(1)]. If the changes in the regulations discussed above are summed up as a whole, certain trends can be observed from the 1990s to 2009, and these trends have also had a significant impact on the direction of technical development. These trends will be discussed below.

In 1994, the first driver death in eight years made the 1990s a period in which reconsideration of Formula One safety was pushed ahead rapidly. From 2000, when Honda reentered Formula One racing, until today, the application of limitations to vehicle performance and the achievement of increased vehicle safety have been constantly promoted. The limitation of vehicle performance has mainly focused on restriction of driving performance, primarily by means of engine regulations, and the application of regulations concerning chassis dimensions, aerodynamics, and tires, focusing on the reduction of cornering speed.

In order to seek enhanced performance and help to ensure competitiveness against the background of these regulations, teams have pushed ahead with the development of new high-efficiency mechanisms that reduce loss and modifications based on heat and flow analyses conducted using computers. Developments focusing on the achievement of weight savings, increased compactness, and higher rigidity – universal and important elements of development for racing – have not rested with simple substitution of materials, but have entered a new stage, with limit design using structural analysis technologies and the development of sophisticated technologies to engineer characteristics and functions into materials, among other initiatives.

In addition, a change in technical orientation that seeks to increase competitiveness by enhancing not only the performance of each technical element but also the overall balance of the vehicle as a package, which could already be observed in the 1990s, became particularly important in the 2000s. The reduction of the maximum width of the vehicles discussed above reduced the clearance between the tires and the bodywork, and the efforts to realize a low, slim rear cowl with consideration of the flow to the rear wings was related to the reduction in the size of the engine, the use of a narrow longitudinal shaft gearbox, and, further, the length and arrangement of the exhaust pipes.

A variety of measurement, analysis, and simulation technologies have come to be employed to help ensure the competitiveness of the vehicles, but for the evaluation of the results of developments, it has become necessary to once again consider what is after all the fundamental predicate of Formula One, i.e., having actual drivers run the vehicles at high speeds. The rapid advancement of computer technologies has enabled Formula One developers in the 2000s to process large volumes of data at high speeds, and to push ahead with sophisticated technical developments spanning a broad range; at the same time, Formula One in this era is a world in which systems of comprehensive technical management that bring these technologies together and technical philosophy themselves are under scrutiny.

Figure 1 shows how lap times in Formula One qualifying practice sessions have changed since 2000 as a result of these changes in the regulations and subsequent technical developments. For the base data, a track was selected on which there had been no change in the length of a lap in nine years; results for rainy weather were excluded. The figure shows the transition in the rate of reduction in the fastest qualifying lap times with the time for 2000 as the benchmark. As the results make clear, despite ongoing changes in the rules aimed at restricting vehicle performance, such as the 20% reduction in engine displacement introduced in 2006, there has been almost no increase in lap times. Times have either been reduced, or have at least been held steady.

This indicates that Formula One is technically incomplete, and there is still margin for enhancement. In particular, the importance of the effect of tires on performance in contemporary Formula One is demonstrated by the rapid reduction of lap times in the period when multiple tire suppliers were in competition, and the deterioration in lap times in 2005, when tire changes were prohibited from the qualifying practice to the completion of the final race.

It has long been feared that the fierce competition in development for Formula One will lead to spiraling costs, and specific regulations have been rapidly put in place to reduce costs against the background of the recent deterioration in global economic conditions. These

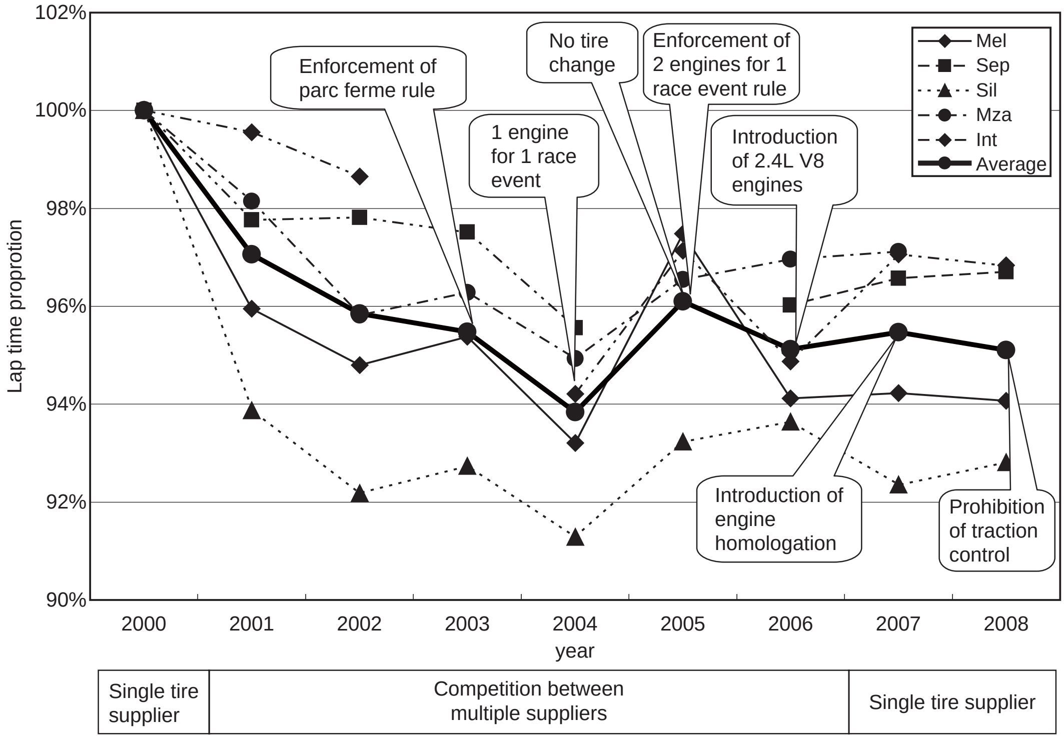


Fig. 1 Transitioin of lap time

regulations do not restrict themselves to regulations concerning the materials that can be employed and the like; a variety of new regulations have been added, including regulations extending the life of engines and gearboxes, regulations concerning a homologation system that will limit development itself, regulations restricting the distance and the amount of days for running tests, and a parc ferme rule designed to reduce the amount of operations at the circuit by prohibiting repair or adjustment of vehicles from the qualifying practice until the completion of the final race. However, if for example running tests are restricted, developers will turn to bench tests and simulations, and the development of these methods themselves will increase in importance. In response, the FIA is seriously considering the introduction of measures, including restrictions on the amount of time that development facilities can be used and a cost cap system that will control total development costs.

Environmental measures are also being emphasized in response to global warming, and the recent recommendation of technologies enabling energy to be reused and the introduction of obligation to control total CO_2 emissions from fuels are noteworthy points.

Efforts are also once again being made to highlight the competition that is fundamental to racing, and to increase the excitement and spectacle of Formula One. The ability to make the spectators feel the differences in skills and the individuality of the drivers, and to transmit the thrill of a race fought out at white heat will be a necessary element in helping to ensure the sustainability of Formula One into the future.

As indicated by the discussion above, we can point to present day as an era in which Formula One is reflecting upon the very meaning of its existence as it responds to rapid social changes. A vehicle that is responsive to any driver input will run fast, and eventually be safe, non-fatiguing, and enjoyable to drive. While the environments in which and the speed at which they are operated are very different, Formula One vehicles and production vehicles share this fundamental characteristic. The author can only express his hope that technical development for Formula One will continue to contribute to fundamental advances in vehicles.

5. Conclusion

Formula One encompasses a variety of aspects, including competition, technical rivalry, and entertainment, and the like. Having failed to achieve a good record of victories, it would be difficult to say that Honda’s activities during its third Formula One era achieved their purpose in increasing recognition of the company or enhancing its image. However, we applied ourselves with a respect for sportsmanship, a dedication to technical development, and an indomitable spirit. The various technical outcomes and primary experiences that were obtained are valuable assets to the Honda, and the process itself can be said to be emblematic of the Honda approach.

References

(1) Tanaka, H.: The Transition of Regulation and Technology in Formula One (20074177), JSAE SYMPOSIUM, No. 22-06, p. 32-41 (2007)
(2) Shirai, H.: Evolution of F-1 Regulation, Honda R&D Technical Review, F-1 Special, p. 6-9 (1993)
(3) Fujita, H., Horiuchi, Y., Kubonoya, H., Baba, J., Sasaki, K: Development of Electric Power Steering System for F-1 Race Car, Honda R&D Technical Review, Vol. 14, No. 2, p. 45-50
(4) Tanaka, H.: Safety Technology in Formula One Racing, Journal of Japan Society of Mechanical Engineers, Vol. 109, No. 1048, p. 148-149 (2006)

■ Author ■

Hiromasa TANAKA

Summary of Honda Formula One Engine in Third-Era Activities

Kazuo SAKURAHARA*

ABSTRACT

Honda has entered six models of V-type engines with 10 cylinders (below, V10 engine) and three models of V-type engines with eight cylinders (below, V8 engine) as third-era Honda Formula One engines. The goals of development were to achieve output approaching that of the second era, the turbocharged engine era, with natural aspiration, and to realize a smaller, lighter engine with a low center of gravity, focusing on the vehicle's height of gravitational center, weight distribution and aerodynamics. Revising the structures of different parts, modifying materials and surfacing processes, and making design engineering and evaluation technique progress among other advancements resulted in per-liter power that is 100 kW/L greater than a second-era naturally aspirated V10 engine, as well as having approximately 6500 rpm higher engine speed for peak output, being more than 50 mm shorter in length and about 70 kg lighter, having about 50 mm lower crank center height and more than three times the mileage. Honda has proceeded with development of V8 engines with the goal of high engine speed and won a third-era victory. However, because of regulations restricting maximum engine speed and an engine development freeze due to homologation regulations, issues relating to drivability (below, DR) have been left unsolved.

1. Introduction

Development of the third-era Honda Formula One engine began in the autumn of 1998 with the program of returning to racing in 2000. Honda has launched a total of nine engines: six V10 engines starting with the RA000E in 2000 and continuing through the RA005E; and three V8 engines from the RA806E to the RA808E. This article gives an overview of Honda's third-era activities, looking back on second-era engines and recounting the movements through the third era, then reviewing the progress made by comparing second-era and third-era V10 engines.

2. Trend from Second Era to Third Era

2.1. Regulations

Development of second-era engines continued for 10 years, 1983-1992, with Honda as an engine supplier. During that period as well, engine regulations were changing greatly, and in 1983, when Honda restarted Formula One racing, the 1.5 L turbocharged engine was the most common type. Subsequently, regulations limiting boost pressure and total amount of fuel consumption were put into place to limit outputs of more than 740 kW (1000 hp), and as Table 1 shows, the transition to 3.5 L naturally aspirated engines began in 1987, while from 1989 all engines became naturally aspirated with a maximum of 12 cylinders. In 1995, after Honda had taken a break from being a works engine supplier, engine displacement was diminished to 3.0 L and the V10 engine became the mainstream. From that point on, changes in regulations became less significant, and the era of the 3.0 L V10 engine lasted 11 years until 2005. The third era began in 2000 and lasted for six years under these regulations. Subsequently, the engine regulation changed to a 2.4 L V8 in 2006, and in the inaugural year of the V8 engine, our victorious third-era engine was realized.

2.2. Output

The key requirements of second-era racing engines were output, drivability, reliability, lightness of weight, compactness, and fuel consumption, which are the same elements needed today. A different era and different regulations, however, mean different priorities, so second-era engines, built when vehicle aerodynamics were still simple and slick tires were allowed, were developed strongly focused on output.

As Fig. 1 shows, the Honda Formula One engine

* Automobile R&D Center

Table 1 F1 engine regulations

Year	Amendments to F1 Engine Regulations
1984	Engine capacity (with supercharging) : 1.5 L maximum (without supercharging) : 3.0 L maximum Number of cylinders : 12 maximum Fuel storage capacity : 220 L maximum Fuel RON : 102 maximum
1986	Engine capacity (with supercharging) : 1.5 L maximum Fuel storage capacity : 195 L maximum
1987	Engine capacity (with supercharging) : 1.5 L maximum (without supercharging) : 3.5 L maximum Manifold pressure (turbocharged engine) : 4.0 bar maximum Fuel storage capacity : 195 L maximum
1988	Manifold pressure (turbocharged engine) : 2.5 bar maximum Fuel storage capacity (turbocharged engine only) : 150 L maximum
1989	Supercharging prohibited Engine capacity : 3.5 L maximum
1991	Fuel RON : 102 maximum , Fuel MON : 92 maximum
1992	Fuel RON : 100 maximum , Fuel MON : 90 maximum
1993	Fuels of a kind used by general public mandatory
1995	Engine capacity : 3.0 L maximum Fuel sampling at circuit Fuel approval before use
1999	Throttle and pedal relationship fixed whilst car is in motion Restriction for engine and clutch control (traction control prevention) Cooling system pressure 3.75 bar maximum Fuel incorporate 2000 EU limit
2000	Number of cylinders : 10 maximum Fuel incorporate 2006 EU limit
2001	High specific modulus of elasticity material (Be-Al, etc.) completely prohibited Traction and launch control permitted
2002	Spraying substances other than fuel into engine prohibited
2003	Parc ferme rule (no engine change between qualifying and race)
2004	1 engine for 1 race event Launch control and fully automatic gear shift prohibited
2006	90 degree V8 2.4 L engine 1 engine for 2 race events
2007	Engine homologation (freezing of engine main component development) Rev limit : 19000 rpm
2009	8 engines throughout 1 race season Rev limit : 18000 rpm KERS (hybrid system) permitted

already had output of 440 kW (600 hp) during the 1983 races, and the era of the 1.5 L V-type engine with 6 cylinders equipped with turbochargers (below, V6 engine) further enhanced output by increasing boost pressure. By 1986, the engine had achieved 770 kW (1050 hp), actually increasing output by more than 300 kW in three years, with the result that regulations were changed to prohibit supercharged engines.

Another factor that increased output in this era was the development of fuels. As Table 1 shows, under the regulations of this era, there were few limitations on fuel, and in particular no restrictions on energy density, so fuels came into use that deviated substantially from commonly available products. Starting with the 1992 Hungarian Grand Prix, teams were obliged to use premium levels of market fuels and the use of special hydrocarbons was prohibited, so output actually dropped by 32 kW (43 hp) as compared to the German Grand Prix that preceded it. This example demonstrates how fuel contributed to engine output.

In the third era, even stricter regulations came into effect on premium levels of market fuels in 1999 and thereafter to prevent excessive competition, and there were no major increases in output as a result of fuel as in the second era. In the V10 engine era, however, when the free development of engines was allowed, the performance of naturally aspirated engines advanced to rival the output of the second era (the turbocharged engine era). This shows that the technology of the engine itself had advanced at high speeds, and some of the content relating to this will be mentioned below in the comparison of second-era and third-era engines.

2.3. Weight

This part discusses changes in the weight of engines. As Fig. 2 shows, the weight of the engine alone was 120-130 kg even during the V6 era, and there have even been some naturally aspirated V10 or V12 3.5 L engines weighing 155-160 kg. Magnesium and titanium were used at the time as materials to make engines lighter, but even so engines could hardly have been called very light.

In 1992 and beyond, although this was not a works activity for Honda, engine output seemed to stagnate, as Fig. 1 shows, in part because of the impact of regulation change and, the lack of competition among works engine suppliers. For that reason, the chassis constructor teams started emphasizing weight distribution and aerodynamics, and making the engine lightweight with a low center of gravity became more important than before. As a result, engine development advanced year by year during the era when Honda was racing as Mugen-Honda, and engine weight reached as low as 122 kg by 1998.

Coming into the third era, there was even greater need for lightness of weight, and development has continued until V10 engines finally weighed just 89 kg.

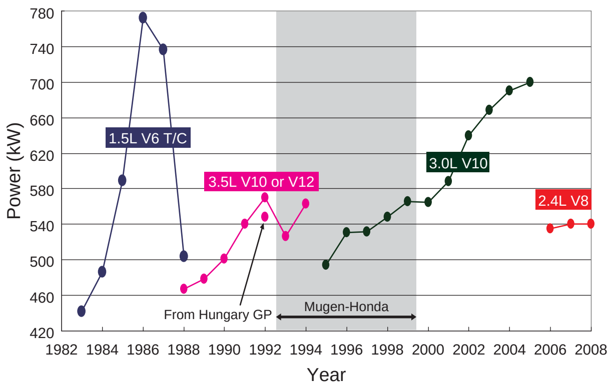


Fig. 1 Honda F1 engine power

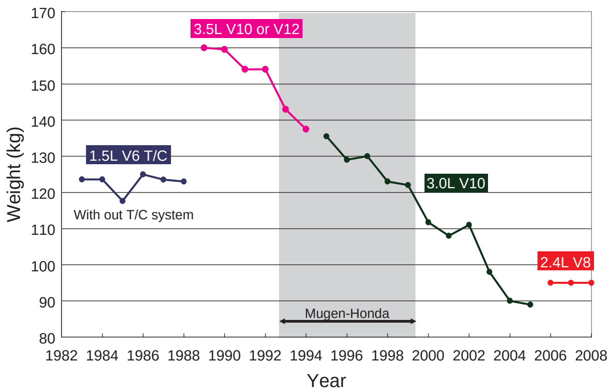


Fig. 2 Honda F1 engine weight

Table 2 RA100E versus RA005E

	RA100E	RA005E
V-Bank angle (degree)	72	90
Power (kW/L)	143	240
Ps peak Ne (rpm)	12250	18700
Engine length (mm)	633.0	581.5
Engine weight (kg)	160	89
Crank center height (mm)	109.0	58.5
Mileage (km)	400	1400

3. Comparison of RA100E and RA005E

3.1. Comparison of External View

The following compares the RA100E in 1990 and RA005E in 2005, two engines of the same V10 configuration from the second and third era, respectively, to discuss in broad terms the progress that was made over 15 years. Table 2 shows comparative values.

As the photos of the external view shown in Fig. 3 indicate, the RA005E is not so high. This difference has to do with the change in V-bank angle from 72° to 90°; naturally, an engine with a large V-bank angle will have a low center of gravity. In a V10 engine, when the left and right banks have a common crank shaft pin, a V-bank angle of 72° with even firing intervals would theoretically be advantageous, considering the load placed on the pin. However, in the third era, engines were already using an 80° angle as early as 2000, the inaugural year of that era, daring to choose uneven firing intervals. In 2001, there was an angle of 80°, in 2002, 94°, and since 2003, 90°, so priority was not given completely to even firing intervals. The reason is that enhancement of aerodynamics and a lower center of gravity took priority as elements determining vehicle speed in Formula One cars in the third era; the basic structure prioritized the total car packaging layout instead of chasing some ideal for the engine itself, which is after all just one component of the vehicle. Renault went so far as to use V-bank angles of greater than 100° for a time in its try for a lower center of gravity. The V8 engines of the time had to have an angle of 90° by regulation; there was no choice in the matter. With V10 engines, all teams were already using 90°, so there was no problem with this. However, it means that no unique engines such as Renault's would come into existence thereafter.

RA100E RA005E

Fig. 3 Engine photo

3.2. Comparison of Output

A comparison of output per liter shows that the RA005E's power is 240 kW/L, or about 100 kW/L greater than the RA100E's 143 kW/L. The elements of enhancing performance are to increase volumetric efficiency and combustion efficiency and to reduce friction; these are fundamental to engine development and there is nothing special about them. Since this concerns naturally aspirated engines, engine speed is the dominant factor in volumetric efficiency, and the RA005E's peak output was achieved at 18700 rpm, 6500 rpm higher than the RA100E's 12250 rpm. Necessary conditions for achieving high engine speed are to stabilize valve behavior by reducing valve train equivalent mass and to increase durability and reliability of the reciprocating system. The Honda Formula One engine reduced equivalent mass specifically by changing from a direct-driven tappet system to a rocker arm system, and furthermore switching materials from titanium to a titanium/aluminum alloy for the valves and making the stems more slender. In the reciprocating system, Honda was able to reduce the weight of moving components by using aluminum matrix composite (AMC) pistons and box-structure connecting rods (below, conrods) made with an intermetallic bonding production method. By additionally using an alloy with good heat conduction for the conrod bearing (plain metal), the engine reduced the temperature of sliding against the crankshaft pin and achieved reliability at high engine speed.

3.3. Comparison of Overall Length

A comparison of engine dimensions shows that, whereas the RA100E is 633 mm long, the RA005E is 581.5 mm, or 51.5 mm shorter. Cylinder bores are 93 mm and 97 mm, respectively; the RA005E's bore diameter being 4 mm larger means that its length would be 20 mm longer, or 653 mm. Converted from that basis, it is 71.5 mm shorter.

Broadly speaking, there are three technical elements that made the shorter engine possible. The first was that it became possible to join the cylinder block (below, block) and cylinder liner (below, liner) into a single piece. In the earlier Formula One engine, the block and liner were separate, and they were generally built into wet liner structure so that cooling water could make direct contact. The separate structure has commonly been used because the liner undergoes special surface finishing where it slides against the piston, which makes it more convenient if it is separate from the block. The separate structure has had the additional advantages of offering a high degree of design freedom for the water jacket of the block itself, and during rebuilding, only the liner needs to be replaced. However, recent advancements in block casting technology and piston sliding surface-processing technology have made it possible to integrate these into a single piece. Moreover, higher engine speed has decreased block life, whereas engine assembly life has increased due to regulation requirements. Thus, engine assembly life and block life are now about the same, so there is less meaning in replacing only the liner; in the third era, the movement to a single piece has advanced, and this is more or less the case with other engine suppliers. In addition, the structure of the gasket that seals between block and cylinder head has changed greatly, so the gasket is now an O-ring type, sealing just the bore perimeter, instead of the old sheet form. As a result, the dimension between cylinder axes is shorter, making it possible to reduce engine overall length.

The second technical element is that advancements in technology to make gear materials stronger and reduce gear vibration have enabled a change of valve gear train with a reduction system to one with a serial system, thus making it thinner.

The third element is that advances in conrod bearings have allowed them to become narrower, which has reduced the amount of offset of the left and right banks.

3.4. Comparison of Weight

The weight of the RA100E is 160 kg while that of the RA005E is 89 kg. This reduction of about 70 kg in 15 years represents a weight reduction of more than 40%. Even just looking at the third era, the RA000E in 2000 was 112 kg, so engines reduced 23 kg in five years: an average yearly reduction of 4 - 5 kg. Although the push to lower weight stagnated for a time in 2002, the engine resumed reducing weight in 2003. Magnesium and titanium, useful for reducing weight, were already being used in the second era, so in the third era it has been thought that there are few advantages of using these materials. Advances in design technology can be mentioned as a reason for even further weight reduction. From the second era to sometime in the third era, design methods were mostly 2D, but starting around 2003, 3D gradually came into use, and with the implementation of CATIA V5 shown in Fig. 4, completely 3D design and CAE became easy to use at the designer level, and engines became correspondingly lighter. In order to prevent excessive competition, current regulations stipulate that V8 engines must weigh at least 95 kg, and they prohibit the use of magnesium, titanium, resin, and CFRP as materials. If these restrictions were not in place, calculations suggest that designs weighing about 78 kg would be possible, which indicates how much the technology to reduce engine weight has evolved.

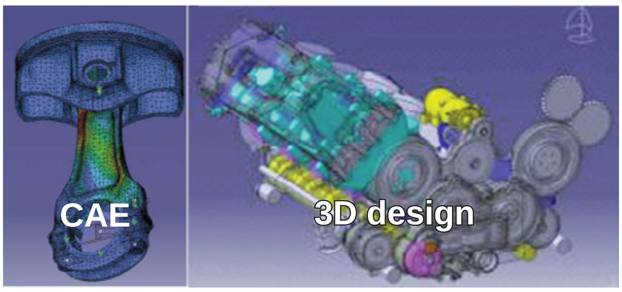


Fig. 4 CATIA V5

3.5. Comparison of Crank Center Height

Crank center height is a factor that determines the engine's center of gravity and affects the vehicle's overall height of gravitational center, and as such is an important dimension determining vehicle dynamics. Current regulations thus stipulate that it must be at least 58 mm high to prevent excessive competition, and furthermore that the engine's center of gravity itself must be at least 165 mm from the bottom of the engine. Whereas the RA100E has a crank center height of 109 mm, in the RA005E it is 58.5 mm, or more than 50 mm lower. Broadly speaking, there are three technical elements that made this possible. The first is stronger ferrous materials, which allow for more slender crankshafts, as well as more slender conrod bolts. As a result of combining these newly developed parts, smaller conrod locuses are realized. The second element is advances in crankshaft counterweight design engineering. Materials with high specific gravity, such as tungsten, that are used as balance weights can now be bolted down directly under even high levels of centrifugal force such as at 20000 rpm, which has reduced the crankshaft's radius of gyration. The third element is modifications in clutch friction material and construction, so that even with a smaller diameter it is still possible to get sufficient torque transmitting capacity.

3.6. Comparison of Mileage

In 1990, when the RA100E engine was in use, a single driver used at least three engines during one racing event, because they were allowed to use one for the Friday free practice, one for the Saturday qualifying practice, and one for the Sunday final race. Since the mileage of the final race is about 300 km, the engine only needed to have reliability of 400 km, including the Sunday morning warm-up session. The RA005E engine, on the other hand, had to have durability of about 1400 km, because it arrived in the second year of regulations, begun in 2004, which intended to reduce high engine costs by requiring that a smaller number of engines be used. After that, each driver had to use just one engine during two race events. This restriction remained in force until 2008; beginning in 2009, each driver was permitted to use a maximum of eight engines in a year. As noted above, advances in the engine itself and in evaluation methods can be mentioned as technical elements that more than tripled the durability and reliability of engines as compared to the second era. In addition to the previously mentioned advances in the engine itself, there have also been advances in surface modification technology, as exemplified by diamond-like carbon (below, DLC) coating. The evolution of DLC has been particularly remarkable, so that, in almost all of the major sliding areas of a Formula One engine, DLC of one specification or another is currently used. If this were to be prohibited, it is likely that both reliability and performance would diminish greatly. The next advancement is in evaluation methods, the biggest factor in which is the use of low-inertia transient dynamos (dynamos with the same inertia as wheels and tires), which allows personnel, using the engine alone, to perfectly simulate driving a circuit. Engine durability and reliability are affected when using traction control or by irregular revolution resulting from ignition cuts when engine speed is restricted, but in the past it was only possible to reproduce this by actual driving, so durability and reliability could not be definitively guaranteed in advance for all 16 or more circuits.

4. V8 Engines

Development of V8 engines, the use of which has been obligatory since 2006, began in November 2004. In May 2005, before any other team did so, Honda conducted test runs on the Jerez circuit in Spain with a prototype engine, a V8 adapted from the 2004 V10 model. Having a goal of maximum engine speed of 20000 rpm in Honda’s 2006 racing engine, development was carried on at a quick pitch, with first firing test on dynamometer in August 2005. The result was Honda’s only third-era victory, at the summer Hungarian Grand Prix, which proved the excellence of Honda’s V8 engine. Development was proceeding that sought an even higher engine speed for 2007, but regulations limited engines to a maximum of 19000 rpm. Meanwhile, homologation regulations limited annual development of engines themselves, and engine specifications have been determined that have become the basis for the development freeze since 2008, specifications that Honda was unable to deal with satisfactorily. As a result, transient combustion has not stabilized, and the 2008 prohibition on traction control has additionally caused DR issues to emerge. Honda has taken measures as far as development is possible, but has not achieved any fundamental solutions. Since DR was still insufficient, the season was a disappointing one. Talks with the FIA resulted in a partial lifting of the development freeze for 2009, and Honda began development in October 2008 to solve DR issues in anticipation of the opening of the next season, and confirmed in December that performance was as targeted. Unfortunately, Honda has withdrawn from Formula One activities, so it has not been possible to prove the results of enhancements. The emphasis in current Formula One engine development is not to put the foremost priority on maximum output, which has always been considered Honda’s strength, but rather on improving output characteristics, particularly in the range where they affect DR, and on somehow eliminating irregular combustion. The reason is because even if output were increased by 10 kW, it would only improve lap times by less than 0.1 seconds, whereas in comparison poor DR makes the driver lift off the throttle pedal while coming out of the corners, resulting in a loss of more than 0.5 seconds. Honda engineers need to do a lot of reflecting on the fact that there has been little development emphasizing this.

5. Conclusion

This paper has looked back on Honda’s Formula One engines, using numerical values to compare the second and third eras, which reminds us anew of how amazing the technical advances have been over the course of 15 years. Because there are presently so many regulations, there is little room for further advances such as these, but it is still supposed that the technology will advance bit by bit hereafter. Regardless of whether Honda returns to Formula One racing, the company feels it must endeavor to preserve the information network it has created and pursue further technical advances (concepts). Otherwise, in such a rapidly advancing world, it may take a great deal of time to catch up if we again get the chance to take part in Formula One engine development.

■ Author ■

Kazuo SAKURAHARA

Overview of Gearbox Development for Formula One

Atsushi MANO*

ABSTRACT

In the racing world, the transmission is ordinarily referred to as the gearbox. Honda first tried developing an Formula One racing gearbox with its third-era Formula One activities. To do this, it was necessary not only to develop the technology, but also to solve several problems, including production, supply and operation. This article recounts how Honda overcame these problems to bring a number of technological firsts to the racing world and gives an overview of the advancement of Formula One gearbox technology.

1. Introduction

Generally, when people talk about Formula One technology, engine power and aerodynamic performance are often mentioned, but there are few opportunities to bring up gearbox technology. This article, therefore, will start by discussing the basic elements essential to gearboxes and recount how Honda's development advanced these elements.

2. Formula One Gearboxes

2.1. Regulations

The following is a brief excerption (not a verbatim quotation) from the Formula One technical regulations on transmissions as stipulated by the Federation Internationale de l'Automobile (FIA).

(1) Only two-wheel drive is allowed
(2) It must be possible to cut the clutch manually when the vehicle is stopped

In the event that a vehicle stops along the course because of vehicle trouble or any other reason, course attendants must be able to push it out of the way, so vehicles have a manual clutch cutoff switch that even works when the engine is shut off.

(3) The minimum number of forward gear ratios is four and the maximum is seven, and CVT is prohibited
(4) Vehicles shall have a reverse gear
(5) Left/right torque transfer is prohibited

In addition to the above, the following rules were added in 2008 to control costs.

(6) A single gearbox must be used in four consecutive races
(7) Gear ratio pairs must have a minimum thickness of 12 mm wide with at least 85 mm between centers, and each set of gears must weigh at least 600 g

Other than the parts newly established to avoid excessive technical competition, these restrictions are only a general framework, and there is overall a very high degree of freedom in the design of gearboxes. Because of that, each Formula One team is aggressively developing technology with the major objectives being lightness, compactness and high efficiency.

2.2. Required Performance

The following lists the performance required of a Formula One gearbox.

(1) Durability and reliability
(2) Lightness and compactness
(3) Lower center of gravity and low yaw inertia moment
(4) Quicker gear shifting
(5) High transmission efficiency
(6) Stiffness of casing

Unlike mass-produced vehicles, the rear suspension is directly attached to the casing (Fig. 1), so sufficient

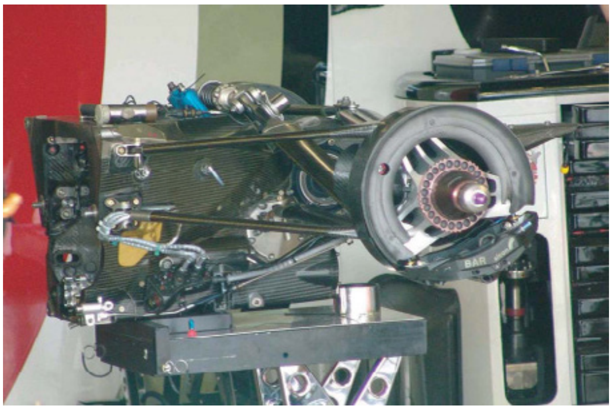


Fig. 1 Appearance of F1 gearbox

* Automobile R&D Center

stiffness and strength are required, and this has an impact on vehicle behavior.

(7) Must contribute to aerodynamic performance

The gearbox must be as small as possible so that it does not impede airflow to the rear wing and diffuser at the back of the vehicle. In particular, the rear part must be narrow.

(8) Easy to maintain

It must be possible to complete ratio gear changes in the interval between racing sessions.

In addition, the internal mechanism transmits engine torque input through the clutch through seven sets of gears, a bevel gear and final gear, as shown in Fig. 2, and from there through the drive shaft to the tires.

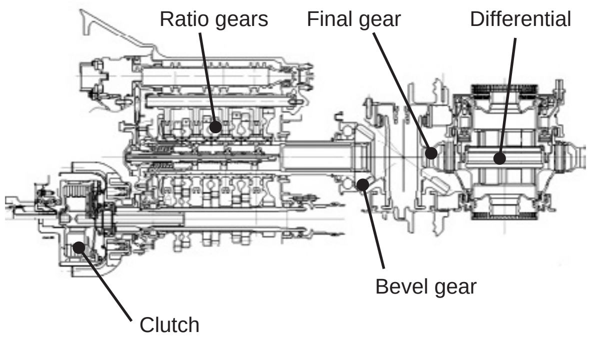


Fig. 2 Structure of F1 gearbox internals

3. Preparation for Original Development

3.1. Establishment of Joint Development Partnership with Racing Team

Honda's third-era activities began in the form of joint development of racing cars with British American Racing (BAR). Because development of the gearbox was mainly done by the race team BAR, the first issue to resolve was how to proceed with joint development. The development conducted by BAR at the time was on the part concerning the overall package, including the casing, while detailed design and production of internal components like the gear shaft were contracted out to manufacturers specializing in racing transmissions. Therefore, although BAR was able to present its requirements relating to gear and shaft layout, it was difficult to make these parts lightweight and develop innovative mechanisms faster than other teams, and even if it came up with good ideas, it was difficult for BAR to use them exclusively. On the other hand, Honda's strengths were its ability to manufacture casings, gears and shafts internally and the fact that it had bench testing equipment to evaluate the strength and reliability as well as performance of gearboxes, something which other racing teams at the time did not have. If Honda therefore could provide its competitive technology in place of the specialized manufacturers, the advantages to BAR would be significant, such as being able to develop pioneering technologies and use them exclusively. For that reason, the two sides agreed that Honda's Tochigi Automobile R&D Center (HGT), a development base in Japan, would oversee development of technology, especially for gears and shafts, and full-scale development began in HGT in 2002. Subsequently, development continued with the same division of responsibilities even after the team reorganized into the Honda Racing F1 Team (HRF1) in 2006.

3.2. Learning Formula One Gearbox Development Technology

Before development could proceed, Honda first had to learn the level of technology in Formula One gearbox. The following four major issues had to be addressed in order to do this.

(1) Design engineering

As for the type of gear, mass-produced automobiles use helical gears, where the emphasis is put on gear noise. In contrast, racing cars use spur gears, with more emphasis on efficiency and strength. Although there are differences, the design tools are the same, and the basic design techniques of the former could be used in the latter. The conditions under which racing cars are used, however, made it impossible to use the historical data from mass-produced automobiles as reference, and Honda learned that it is necessary to deal with early onset gear surface pitting and to make revisions to gear teeth tips, taking into account elastic deformity of the gear teeth caused by impact load input from such forces as shift shock.

(2) Production technology

In order to ensure gear strength and maximize transmission efficiency, all gears underwent heat treatment and subsequent gear grinding (that is, grinding of the entire tooth down to the bottom), and then for the surface finish underwent a treatment to reduce surface roughness of the tooth (including barrel grinding) and shot peening to enhance fatigue strength. In particular, the grinding process required numerous rounds of trial and error and new arrangements in order to get the same level of surface roughness as the specialist manufacturers' gears, which were used as reference. In addition, during shot peening, in order to give the material areas with higher surface residual stress and deeper residual stress, Honda had to spend much time catching up to the specialist manufacturers' performance, not only selecting media but using multistage shot and choosing the right process pressure and times.

(3) Materials technology

The gear materials that were being used in the racing industry were the same materials used in mass-produced gears but with extra strengthening, and to be competitive it was necessary to develop specialized high-strength gear material. Honda worked with steel manufacturers and had to spend many hours until it could get satisfactory material performance.

(4) Preparing a cooperative prototype production system

To develop competitive technology in timely fashion, one absolutely must have production support in the form of fast prototype-production times by a number of manufacturers specializing in the area of prototype production. Honda received support from many proven prototype producers of gears and was able to manufacture gears and shafts with relative ease, but it faced more difficulty in prototype-producing bearings. When developing new gearboxes for racing purposes, the use of bearings of a specialized form contributes to performance enhancement. Bearings, however, are highly specialized functional components, and there are no manufacturers who prototype-produce them. As a result, Honda had to depend on manufacturers of mass-produced bearings to prototype-produce some, but the manufacturers had difficulty meeting Honda's need for small quantities of many types, so the prototype-production period grew longer and there were frequent situations where the manufacturers could not meet Honda's development needs. However, thanks to development support from NTN Corporation, Honda was able to build a cooperative development system for Formula One bearings. As a result, the time from examination of specifications to prototype production and supply of components was shortened and it was possible to work in a timely manner.

3.3. Technical Development Goals

To address the issues mentioned in the preceding sections and ensure competitiveness in the area of gearboxes, the major development goals were narrowed down to the following two, which are big factors in lap times.

(1) Lightness and compactness

Regulations dictate that racing vehicles must weigh no less than 605 kg, including the driver, so a lighter gearbox does not lead directly to a lighter vehicle. However, by increasing ballast weight to adjust overall weight, one can try to increase the degree of freedom for vehicle weight distribution and lower the center of gravity. Also, greater compactness not only leads to lighter weight but also enhances aerodynamic performance by giving the gearbox a slim form. These contribute to the vehicle's competitiveness.

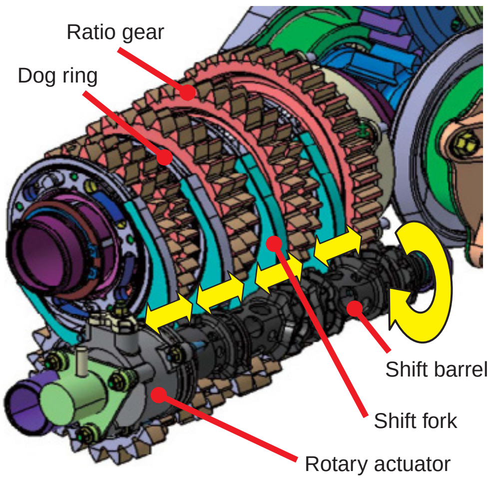


Fig. 3　Appearance of F1 gear-change system

(2) Faster gear shifting time

Beginning in the early 1990s, Formula One autos used a sequential shift mechanism consisting of a dog clutch and a shift barrel driven by a hydraulic actuator, as shown in Fig. 3, a system that works on the same principle as that used in motorcycles. Particularly when upshifting during acceleration, the vehicle can decelerate with a force of as much as 1.0 G from air resistance because of interruption of torque delivery, so reducing shifting time is a crucial objective directly linked to better lap times.

3.4. Establishment of Racing Component Supply System

In order to bring competitive, unique technologies, one must not only develop those technologies but also produce and supply the number of components needed for operations with stable quality. Prior to this development, Honda had supplied assembled complete engines to a race team, but this was its first experiment supplying individual components. For that reason, it was necessary to create a system, including the organization, in order to achieve the two following points.

(1) Production quality assurance

For those important components associated with the driving function, the history of each component is controlled (this includes control of production lot and driving lifecycle). For this purpose, the part number, serial number and production lot are controlled. Infrastructure needs to be prepared so that this information can be placed directly on components by laser marking. Honda has not only introduced a laser marker, but also established rules, routes and a system for conducting this work.

In addition, it was necessary to attach an inspection report card to each component as evidence that it satisfied blueprint quality. To do this, Honda arranged inspection systems for all components and decided on serial number engraving and key point control dimensions. Honda additionally established rules for writing concession reports on rescue measures for components slightly outside the tolerance which were judged to be no issue functionally, and a system was set up at both HGT and HRF1 to determine whether components presented any functional issues, and if so, to rescue them. This effort sought to prevent unnecessary cost increases and stabilize component supply.

(2) Component shipping system

Since HGT did not previously have a shipping function for individual components, Honda prepared the processes from packaging to shipping of finished individual components. To make sure that ratio gears, final gears and so on could be efficiently stored and assembled at the factory or circuit, Honda created special packing boxes designed to prevent rust, cushion impact and simplify component identification.

4. Details of Technical Advancement

Competition among racing teams to develop technology has caused gearbox technology to evolve during the third-era Formula One activities. This section introduces and reconsiders the details of this advancement, looking at changes in the trends at other teams and specific technologies worked on at HGT.

4.1. Casing

Since the casing also functions as part of the chassis, it must be highly stiff as well as lightweight. Techniques used by Honda to achieve this were research into a variety of materials and development of production methods. In the past, sand casting of aluminum alloys and magnesium alloys was standard procedure, but in recent years the paths that racing teams have chosen have gone in two different directions[(1), (2)]. The first path is to advance the casting production methods and save weight by casting with thinner walls. This approach, which used rapid prototype technology, makes the walls thin and at the same time, since there are no restrictions on draft or undercut, it is possible to eliminate useless material. Aluminum or titanium is chosen as the casting material. The second path is to use carbon fiber reinforced plastic (CFRP) as shown in Fig. 4. CFRP can offer the advantage of much greater specific stiffness than metal materials and is also superior in terms of both lightness and stiffness. It takes a long time to manufacture, however, and the production cost is disadvantageous compared to casting. As of 2008, the various racing teams' production methods have diverged into the two methods mentioned, and there is no evident trend of unification to one or the other. At HGT, the engineers worked to develop a lightweight magnesium casing using a CAE program to optimize stiffness, and this was released at the end of the 2002 season. BAR, however, developed a CFRP casing that could be made even lighter, which was put into use starting in 2004, so HGT stopped developing casings.

4.2. Gears

To procure gears, the top teams set up their own technology development environments in which gears were designed and produced internally. In contrast, lower ranking teams procured gears from specialty manufacturers. Internal mechanisms standardized for Formula One use were customized into packages for each team's vehicle. As stated above, greater compactness of the gearbox itself contributes significantly to the vehicle's competitiveness, and therefore active initiatives have been taken to reduce the size of gears. Techniques for doing this include using high-strength gear material, optimizing the safety factors by using FEM, and advancing the production methods. HGT pursued independent development starting with the gear material, and by enhancing root bending strength and tooth surface pitting strength, it was able to reduce gear width by 1 mm for first through third gears and by 2 mm for fourth through seventh gears, as compared to the gears produced by a specialty manufacturer which had been the base up to that point. This also reduced total gear weight by approximately 1 kg. The new gears were released with the final stage of the 2003 season and thereafter were the standard. At first there was some unexplained gear tooth damage (breakage from the root), but analysis revealed the mechanism: shock loads more than anticipated while gear shifting and sudden changes in vehicle behavior caused some gear teeth to undergo plastic deformation, lowering fatigue strength, and causing breakage in a short span of time. As a result, design standards were established and applied to prevent tooth deformity in ordinary use. In terms of design, however, increasing strength and reducing weight are contrary to each other when dealing with greater than anticipated inputs, and this reduces competitiveness. Therefore, the engineers developed controls to keep excessive input from occurring, and by integrating torque sensors they put a system in place to monitor input values on gears at all times, so that if a gear underwent more than the allowed torque, that gear would not be used again. This effort helped to reduce gear weight in balance with gear strength. The engineers used a gear tester to identify the torque at which tooth bending occurred and found the correlation to design stress, which made it possible to set a highly precise design allowable stress.

Fig. 4 Appearance of CFRP gearbox casing

4.3. Gear Shifting Mechanism

The sequential semi-automatic gearbox that appeared in the early 1990s became established as the basic structure, with no change until recent years. In the meantime, attempts to reduce gear shifting time focused on reducing operating time by reducing component weight and friction, and on torque optimization control by means of cooperative control with the engine, but no great progress was made. To achieve faster gear shifting times, one of their key development goals, the HGT engineers developed a seamless gear-change mechanism which eliminated interruption of torque delivery while shifting. Theoretically, this mechanism allows the change in gears to take place instantly by letting the next gear engage while still driving at the previous gear. Broadly speaking, there are two major issues involved as described below to achieve this, and the Honda engineers dealt with each of them.

(1) Avoiding interlock from double engagement of gears

Interlock generally results when engaging two gears simultaneously, which causes gear or shaft damage. To make double engagement of gears tolerable, therefore, a one-way clutch structure was incorporated, enabling the one-way clutch to idle even when the former gear switches to a deceleration tooth surface after selecting the next gear, and making it possible to avoid interlock. However, if a one-way clutch is always in operation, issues occur, such as being unable to use the engine brake during deceleration, so the one-way clutch is locked except when shifting, which prevents it from idling.

(2) Increase in shock while gear shifting

Instant gear shifting causes instant change in torque equivalent to the change in inertia based on the gear ratio, so that spike torque increases, in addition to the engine torque. In response, since a major portion of the change in inertia is engine inertia, the engineers reduced spike torque by developing optimization control over the amount of clutch during gear shifting. Combining this control with the mechanism described above made it possible to achieve both this objective and gear strength.

Results of repeated bench tests and circuit tests indicated that these issues had been resolved, so the technology was used in races starting with the 2005 season. Lap times dropped by 0.4 seconds per lap, a very significant advantage for a single technology. Moreover, with no interruption of torque delivery while upshifting and with reduced spike torque, there was less drive fluctuation while shifting and it became possible to shift even while cornering or when there was low tire grip, such as in rainy conditions. Honda was the first to use this technology for eliminating interruption of torque delivery when shifting in a race, but since it was so effective, other teams began eagerly developing the similar technology, so that by 2008 all teams were using it as seamless shifting. It is said that the structure used by other teams, however, had two shift barrels, with the even-numbered gears and odd-numbered gears controlled separately. In contrast, the structure used by Honda consisted of just one shift barrel and was unique in that it prevented double engagement of gears, giving it superior reliability and lightness.

The engineers furthermore undertook to develop an innovative gear shifting mechanism with the goal of producing a technology to reduce the overall length of the gearbox. As Fig. 2 shows, the length of the gearbox can be broadly divided into the clutch, ratio gear, bevel gear, final gear and differential portions. Here, the engineers focused on shortening the ratio gear portion and began a series of investigations based on the idea of eliminating the dog ring and shift fork located between gears. The result they hoped to achieve was a structure that, without losing the advantage of seamless shifting, would reduce overall length by 19% and weight by 12% by putting the shifting components inside the main shaft and aligning the ratio gears without any gap. Although it was difficult to ensure durability of an internal shifting mechanism, the given target performance was achieved as a result. It had been decided to use this technology in circuit tests at the end of 2008, but this plan was canceled when Honda decided to withdraw from Formula One racing.

4.4. Clutch

In order to maximize launch performance at the start of races, development was begun with the goal of stabilizing clutch transmission torque. Conventional mechanisms controlled clamp load against clutch friction material by adjusting the piston force that hydraulically operates the preload set by the diaphragm spring. This structure made it difficult to control launch torque with much precision because individual assembly statuses caused variance in the clamp load, and wear of the friction material caused diaphragm spring characteristics to change. Figure 5 shows the Direct Push Clutch (DPC) that was developed to address this. By using this mechanism, the clamp force of the friction material was acquired directly from the hydraulic force, eliminating the element of variance and enabling high-precision clutch-torque control. The greater part of load hysteresis during operation, a factor that can reduce control characteristics, is seal friction. To reduce this, the type of hydraulic piston used was one that placed small-diameter plungers at two points, minimizing the impact of seal friction. This was first used as original technology in 2006 and subsequently became standard in HRF1.

4.5. Differential

The differential is one of the components showing the most progress toward compactness in the past 10 years, and according to data from a specialty manufacturer, differential assembly weights were just 6.1 kg in the 2004 specifications, a reduction of more than 50% from the 12.5 kg in 1998[2]. This progress is primarily the result of advancements in package technology and optimized safety factors. Since the differential is located higher than the vehicle's center of gravity, making it lighter is a very effective way of lowering the height of

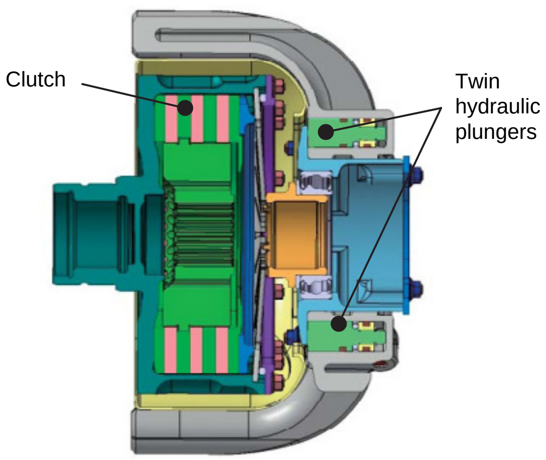


Fig. 5 Appearance of Direct Push Clutch (DPC)

the gravitational center. Reducing differential width, moreover, helps enhance aerodynamic performance and makes longer drive shafts possible, so the sweepback angle of the drive shaft is smaller, which enhances transmission efficiency. Differential mechanisms include a bevel gear set-type and planetary gear-type, but the latter is superior in terms of weight and total width. The HGT engineers also actively engaged in an effort to make differentials more compact, and developed a number of original technologies. This section introduces the Ultra Short Diff (USD), which used a unique mechanism to try to achieve a more compact differential. Figure 6 compares USD to RA108, the 2008 specification, in cross-section. Using a full-engagement double-pinion planetary has resulted in 20% less distance between centers and 14% lighter weight. By the end of 2008, the engineers had finished ascertaining performance, durability and reliability and were aiming to introduce the technology to racing in 2009.

4.6. Enhancement of Transmission Efficiency

The original key development goals were lighter weight, greater compactness and faster gear shifting, and Honda's unique development yielded results on these goals and enhanced competitiveness, so in 2006 the enhancement of transmission efficiency was addressed as a key issue. Table 1 gives the results of measuring transmission efficiency under maximum gearbox load at the time.

As is evident, most of the transmission loss came from the meshing of the gears. For that reason, the engineers added surface finishing to their research with the goal of reducing friction losses in gear engagement. Diamond-like carbon (DLC) coating yielded good results, and adding DLC coating to all ratio gears, the bevel gear and final gear made it possible to reduce transmission loss by 3.4 kW. On the subject of coating durability, at first there was an issue with the coating peeling off the bevel gear, which is subject to a particularly high level of contact surface pressure. This was solved by revising the materials and production methods, namely enhancing adhesiveness by reducing the underlying surface roughness and making the film finer, so this technology was used in part starting midway

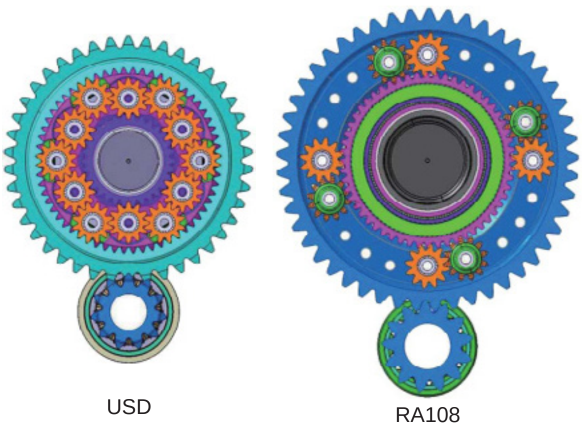


Fig. 6 Comparison between USD and RA108

Table 1 Classification of transmission losses (loaded)

Bearings	Gears	Seals	Churning	Pump
20%	62%	2%	14%	21%

through the 2007 racing season and on all gears starting with the 2008 season opening race.

The engineers additionally addressed how to reduce transmission losses through gearbox oil development. Generally speaking, lowering the oil's viscosity reduces churn resistance, but simultaneously reduces oil film thickness, which increases friction on tooth surfaces when in a boundary lubrication status and, as a result, lowers transmission efficiency. By reviewing various base oils, the engineers were able to address both churn resistance and friction loss on tooth surfaces, cutting transmission losses by 0.4 kW.

5. Conclusion

Although the third-era Formula One activities resulted in only one race victory, they also gave birth to seamless shifting and a number of other industry-first technologies, so Honda's engineers could at least take pride in such developments as they faced the closing of this era. As development began, it was marked by one failure after another, but the engineers learned many things and grew along the way. This is the prize won from participation in Formula One racing and will certainly be an asset for Honda into the future.

References

(1) Mano, A.: Development of Gearbox Technology for Formula 1, Journal of Society of Automotive Engineers of Japan, Vol. 59, No. 9, p. 8-11 (2005)
(2) McBeath, S.: F1 transmission trends, Racecar Engineering, p. 34-42 (2005)

■ Author ■

Atsushi MANO

Summary of Honda Third-Era Formula One Chassis Development

Akio TONOMURA*　Shun KAWABE*　Nobuhisa NAKAYOKU*

ABSTRACT

The third era of Honda's Formula One participation started in 1998 with a stage of preparation for the return to actual racing, which began in 2000 and extended up to their withdrawal in 2008. A summary view of the Honda Formula One cars that were developed every year during this period of participation in competition is presented from the perspective of chassis technology. An overview will be given of each year's aims regarding the chassis and the technology that was required for third-era Formula One racing, accompanied by photographs of the chassis. Each year's regulations and the aims of chassis development will also be surveyed from the separate perspectives of aerodynamics, the chassis, and the like. The points on which development was successful will be discussed on the basis of racing results, and points calling for rethinking will also be examined.

1. Introduction

The Formula One cars of recent years cannot succeed just by one prominent performance capability, but have to achieve a high-level balance of all factors in order to win. Even if one car has the most powerful engine, if it is heavy it will damage the rear tires. If efforts to perfect an aerodynamic advantage limit the space available for the suspension, the chassis will be lacking in stiffness and will become unstable. Going too far in reducing the weight of parts will detract from rigidity and reliability. The question of how to discern the points of optimal balance as a whole for conflicting approaches like these is the key to chassis development.

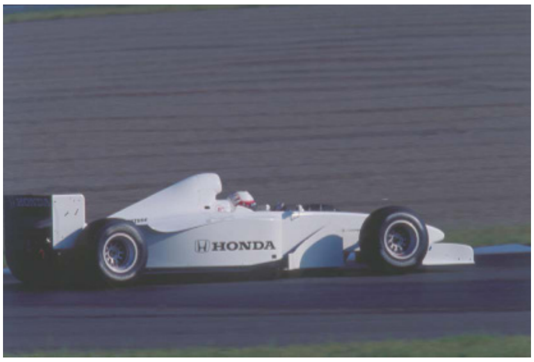
The team conducts actual driving tests in order to search out the optimal balance points. However, testing on a slow car will only yield low-level balance points. For example, when a car has unstable braking, the brakes cannot be applied hard, and since the brakes cannot be applied, the tire temperature will not rise to the working range, and the cornering grip will be reduced. Low grip on corners means that the corner exit speed will be lower, which reduces the top speed on the straight-end. In this way, a sequence of various different factors can fall into a vicious cycle. A team that has a fast car faces the opposite situation, and the difference cannot readily be overcome. These are the reasons that a new team finds it so challenging to come into its own, and it takes a considerable number of years to raise all the factors to a high level. This is because there are various constraints placed on the development environment.

Formula One vehicle testing is carried on with a variety of different limitations, including the funding for team operations as well as limitations on the distance of test drives, limitations on the number of tires, and so on, in addition to which the circuit conditions are constantly changing due to weather, dirt, and tire rubber deposits. When determining the sensitivity contribution of a single factor on vehicle performance, it is necessary to test it with all the other factors held constant. Under these conditions, however, it always seems as though either nothing more than a very rough evaluation or a mistaken evaluation can be done. Therefore, every team competes fiercely on the development of evaluation methods, using bench tests in a stable environment that also includes inexpensive simulations and wind tunnel testing. One might say that the determining factor lies in these evaluation methods. However, it is in the nature of simulations and bench tests that the results may diverge from what is obtained using the actual machine. In order to compensate more or less correctly for this divergence and to decide the direction for development, it is essential to obtain correlation with the results of repeated testing with the actual machine. This means that tests run with the actual machine are necessary, and so the development is caught in a dilemma. This is what makes Formula One development such a challenge: it is a race to overcome dilemmas like these, to find reliable evaluation methods and evaluation criteria, and to quickly discover the car's balance points.

This article will relate how the Honda team dealt with this development race.

* Automobile R&D Center

2. The RA099: 1998-1999



The RA099 came into being in late 1998 when Dr. Harvey Postlethwaite, who had been working with Tyrrell until 1998, came to Honda Racing Development as technical director. The chassis fabrication and simulation analyses were carried out on contract by the Dallara Automobili company in Italy.

Honda was aiming toward entry into Formula One competition from 2000, and the RA099 was intended as a prototype to be test-driven the year before that. The engine was a Mugen-Honda V10. The concept was true to the basics: to build up a proven system by driving it steadily as a dedicated test car and gathering definitive tire data, aerodynamic characteristics, and directions to take in vehicle setup so as to develop a car for actual racing in 2000. The primary focus in this was the aerodynamic concept that there was little variation in the downforce (hereafter referred to as the aerodynamic pitch sensitivity) due to changes in the chassis pitch angle. If this sensitivity was excessive, then changes in the chassis attitude when braking and accelerating would result in excessive aerodynamic changes, causing instability. This effect could not be mitigated through other tuning elements such as the suspension springs or the weight distribution, and the result was that the tire characteristics could not be employed to full advantage. Ascertaining the tire characteristics required raising the tire temperature to the working range of approximately 70°C and maintaining a speed that would keep the temperature at that level. It was therefore crucial to assure the car's aerodynamic stability.

The front suspension was largely like those of the other Formula One vehicles of recent years, in a double wishbone configuration that has torsion springs located in front of and behind the rocker arm shafts. The rear suspension employed a double wishbone with a coil spring coaxial damper positioned perpendicularly to the chassis and connected by a large rocker arm protruding from the aluminum cast gearbox. This layout of the rear suspension opened up space on the top surface of the gearbox, allowing a third damper to be added easily in later testing. The third damper consists of a spring and damper that expands and contracts when the chassis moves vertically with the wheels on either side in phase (i.e., heave). Its purpose is to control pitching.

The layout around the gearbox today, when the Coke-bottle shape and top exhaust represent the mainstream, is designed for a gearbox made as narrow as possible. At that time, however, the mainstream type had exhaust pipes that passed between the top and bottom of the suspension to vent to the rear, and this was the form followed in the RA099.

The only adventuresome step taken amid this orthodox fabrication was in the top wishbone in front. The arms were made of Carbon Fiber Reinforced Plastic (CFRP), the fittings that attached them to the body were made of titanium, and they were bonded together with adhesive. Since that was the time when the use of CFRP for suspension arms was just getting started in the Formula One world, one arm was selected as an example to begin development. The other suspension arms were all made of steel.

The weight distribution on the front axle of the chassis (hereafter referred to as the weight distribution) was designed using calculations based on tire characteristics data. The objective was 45%, and the design faithfully achieved this. The wheel-base was 3020 mm. The height of the center of gravity was 250 mm above the floor, which was lower than that of the following year's car, the BAR002. A crucial issue in race car design is how to assure traction when cornering. This was addressed by an effort to bring the jack-up power that arises from the suspension link mechanism down toward zero by locating the rear roll center more or less on the rear floor. The RA099 was built in an orthodox manner without any remarkable measures to reduce its weight, but even so it turned in a series of good lap times in 1999 tests. The car looked promising for the next year's races, but Dr. Postlethwaite suddenly passed away in April. That May, the project was put back on the drawing board, and RA099 development was brought to a halt.

3. The BAR002: 2000

In 1998, Tyrrell was bought up and British American Racing (hereafter BAR) was founded. The BAR001 was built starting in 1999, and was entered in Formula One competition. The BAR002 was the model from the second year of BAR.

Honda contracted with BAR in 1999 to supply engines and conduct joint development of the chassis from 2000. The work on the BAR002 therefore represented Honda's debut in the third era. The BAR001 from the previous fiscal year had repeatedly experienced trouble every time it raced, and with two cars in a total of 16 races, it had 22 retires. The reliability was therefore upgraded, but even though the BAR002 placed fourth and sixth in the opening competition, it experienced mechanical trouble that resulted in Jacques Villeneuve retiring from the race five times and Ricardo Zonta two times out of 17 races. There was no choice for the development effort throughout 2000, therefore, but to focus on upgrading reliability, and weight reduction was sacrificed accordingly. The fact is that the car was able to load 27.5 kg of ballast in the opening race, but the amount was down to 3 kg in the final race. The Formula One rule is that the car must be driven at a weight of 600 kg or more, including the driver, and considering the weight of the driver’s equipments, the maximum-target weight for the chassis would be approximately 500 kg. The chassis is made as light as possible, and then ballast is loaded to bring the total weight up to 600 kg. As much ballast as possible is loaded, and it is placed in as low a position as possible. This lowers the center of gravity and allows adjustment of the weight distribution, and explains why the amount of ballast is directly tied to chassis performance. By the end of the season, the BAR002 had almost no ballast loaded.

Another issue was the 42% weight distribution. It is not known why this weight distribution became a starting specification. Considering that the RA099 had a figure of 45% using the same Bridgestone tires, this means there was a difference of as much as 3%. A weight distribution that is skewed to the rear is advantageous for acceleration but becomes unstable in the deceleration range. To control that, the rear wing is turned up to increase the downforce. The higher up the rear wing is turned, the more the aerodynamic efficiency is degraded and the drag increases. It is also necessary to suppress fluctuations in the aerodynamics in order to limit the vehicle’s instability, and there was no choice but to stiffen the springs in order to limit fluctuations in vehicle height. The actual spring wheel rate was 25 kg/mm, which was stiffer than the vertical spring rate (24 kg/mm) of the tires themselves. In the lower speed range, therefore, the vehicle would bounce and lose its grip. Having the weight distribution skewed to the rear also placed a burden on the driving wheels and caused the tires to rapidly deteriorate. When the situation progressed, the tires would generate excessive heat, causing blisters to form.

The way that races developed for the BAR002 was to do a rocket start and pass two or three cars before the first corner. After that, the rear tires would degenerate, and the car would drag back and lose its position. This pattern was repeated. The weight distribution was gradually enhanced toward the final races of the season, reaching 44%. In conjunction with this, the distribution of the aerodynamic downforce exerted on the front wheels (hereafter the aero balance) changed from 33% to 38%, but the aerodynamic efficiency ended up decreasing. The reason for this was that the aero balance target value served as a reference in promoting the development of all the aerodynamic parts so as to raise the aerodynamic efficiency to a maximum at the balance point, and making mid-season changes to the target value therefore would cause a fatal delay in aerodynamic development. Similarly, the weight distribution target values that are calculated from the gripping force of the front and rear tires should also be decided initially when beginning chassis development. The reason is that the weight distribution target values end up determining the dimensions, weight, and positioning of the monocoque, fuel tank, engine, transmission, and suspension, so that trying to make changes later causes the yaw inertia moment and the height of the center of gravity to increase.

Another notable development topic was electric power steering (EPS), which was used starting with the Hungary Grand Prix in August, ahead of the other teams. No one doubts its effect any more, but when it was initially developed, people wondered whether power steering really would reduce the lap time. This was because the increase in lap time due to the addition of 2.6 kg to the system weight was clear, but the enhanced stability caused by reduction of the burden on the driver could not be quantified. At that time, however, a self-alignment effort of 40-50 Nm was considered to be necessary for the steering on the high-speed corners in Barcelona and the USA. This means that drivers would have to exert fine control on a 10-inch diameter steering wheel while supporting a 20-kg load on it with their arms extended. The drivers were in a situation where keeping the steering aligned was all they could manage without exerting further control, and it was therefore decided to adopt the EPS. The size of the motor was a constraint on the EPS, and it was developed to be capable of a maximum assist of 15 Nm.

The issues faced with the car during that year were the inability to optimize the height of the center of gravity and the weight distribution because of the weight increase caused by reliability upgrades. With the following year in mind, therefore, weight reduction of the various parts was given priority in order to properly establish those parameters.

4. The BAR003: 2001

Making use of lessons learned in the previous fiscal year, a weight reduction of 33 kg was accomplished and a weight distribution of 45% achieved. Typical weight

reductions are as follow:
(1) Use of CFRP in suspension: 13 kg
(2) Overhaul of gearbox structure: 6.5 kg
(3) Hydraulic systems: 1.5 kg
(4) Thinner walls for tubes in aluminum radiator and enhancements to fins: 3.0 kg
(5) Optimization of orientation and number of monocoque laminates as well as of honeycomb thickness using computer-aided engineering (CAE): 3.0 kg
(6) Achieving a balance of collision performance and weight reduction by using CAE to optimize nose cone, front wings, and rear wings: 4.4 kg
(7) EPS motor enhancement to increase assist torque from 15 Nm to 27 Nm while reducing weight: 0.2 kg
(8) Use of titanium for seatbelt buckle: 0.5 kg

These are some of the ways in which the parts were overhauled and refined.

The aerodynamic performance achieved the lift/drag ratio (L/D) of 3.0 under the aerodynamic conditions of a downforce of 12500 N at a speed of 250 km/h, which was the initial objective when the BAR003 was first raced. Development was pursued with the aero balance objective of 42%.

The wheel spring rate came out at 14 kg/mm, finally bringing the car’s basic characteristics and specifications to appropriate values, and making this a year in which development could raise the car to a whole new level. The weaknesses of the BAR003 were its instability from braking up to the point when the steering wheel started to be turned, understeering (US) at the cornering apex, and the kind of sudden oversteering (OS) called snap-oversteering that occurs when the accelerator is pressed and the car is building up speed coming out of a corner. In other words, the steering characteristics during cornering were inconsistent, requiring constant corrective steering. This was addressed by altering the control in the electronically controlled differential to increase stability by directly linking the left and right wheels when the brakes were applied, then freeing the wheels after that, which was effective in resolving US in the middle of a curve as well as snap-OS when the car was coming back up to speed. It was necessary to develop optimal control, and even before that it was necessary to enhance the basic characteristics of the suspension. In other words, the camber angle-change characteristic (hereafter suspension compliance steer) and the toe angle obtained by input from the tires became the toe out when lateral force was input to the rear suspension. Measurements made after the season showed that these were in the direction of instability. Carrying out measurements with a suspension tester reveals these clearly, and it is taken as the accepted wisdom that new cars should ordinarily be tested once they have been built and before they are driven. Under pressure from the testing and racing schedules, however, there was no choice but to give priority to circuit testing over bench testing, so that practice could not be implemented. As result, however, the discovery of an essential issue was delayed, and the development ended up taking a detour.

5. The BAR004: 2002

This car represented a further evolution with a weight reduction of as much as 14 kg, as well as increased stability achieved by having control play a larger role. While traction control during acceleration was a matter of course, over-run control was adopted to transfer torque from the engine when the rear wheels were detected as locked during braking. Having the rear wheels lock during braking is ordinarily very likely to result in a spin. In order to avoid this, the brake force distribution to the front and rear brakes must allow an extra margin of distribution to the front, but this over-run control now allows the rear-wheel brakes to be applied even closer to their limit.

Efforts were also made in this year to enhance the analysis of maneuverability. One rather basic matter was to enable quantitative evaluation of driver comments about the degree of OS and US from the chassis data. Tools for evaluating stability by means of simulations that take suspension compliance steer characteristics into account were also adopted. All of these originated in chassis performance evaluation technology for mass-production vehicles.

The weight reduction measures included the development of materials for disks and pads in carbon disk brakes. Friction material made with carbon is subject to progressive wear when they are oxidized due to heat, so they are made thick enough to last through the running distance of a race. Curbing that oxidation and thereby reducing the amount of wear enabled reductions from 24 mm to 20 mm in the thickness of disks and from 18 mm to 15 mm in pads made to Suzuka Grand Prix specifications. This also led to reduction in the total width of the calipers themselves, achieving a total weight reduction of 1.6 kg in the system. Weight reduction was pursued by going so far as to implement proprietary development of parts that ordinarily must be purchased from the brake manufacturers. Efforts were also begun to reduce the weight of the materials employed in many different CFRP applications in the chassis. Carbon laminates are made up of carbon fiber and resin, and in order to raise the fiber volume fraction (VF), extra-thin prepregs were applied. These were made from carbon fibers that had been opened up and crushed and then lined up side-by-side without any gaps between them. Their use in the engine cover achieved a weight reduction of 3.1 kg.

New attempts were also made regarding chassis stiffness. Air intake to the engine passes through holes in the roll hoop above the driver's head to the air box where it is conducted to each cylinder. The air box was treated as more than just an air vent, however, by making it a structural member that fastens the top of the engine to the monocoque in order to provide a dramatic increase in chassis stiffness. This measure enhances torsional stiffness 8%, vertical bending stiffness 55%, and lateral bending stiffness 6%. When tested in actual driving for comparison with and without this air box, however, the results showed no difference, and the regular air box was returned to use partway through the season. This led to the formulation of a rule of thumb that driving stability is insensitive to chassis bending stiffness in the vertical direction, and this provided an excellent reference point for implementation of weight reduction the following year. Advances were made in weight reduction and control development during this year. Although the car was balanced, however, it was slow. The balance that had formed was still at too low a level, and it would be necessary to upgrade performance in all directions.

6. The BAR005: 2003

During the previous fiscal year, every element had been reviewed from the ground up under the guidance of technical director Geoffrey Willis, who had moved over from Williams. Dramatic progress was made in the quality control of carbon parts and the reliability of bonded parts in particular. Faults in carbon parts occurred much less frequently after that. Rigorous quality control of other parts was implemented, as well.

Measures to reduce weight also extended beyond simple weight reduction. The effort to reduce weight above the rear axle, to take a particular example, expedited selective weight reduction in the chassis rear end in order to lower the rear center of gravity, making the gearbox lighter and more compact and the engine 16 kg lighter, which was expected to promote stability. These measures had a significant effect in reducing the yaw inertia moment, and even though the wheel-base was extended by 61 mm, from 3133 mm to 3194 mm, the yaw inertia moment remained largely the same at 688 kgm^2. The overall height of the center of gravity was lowered from 243.1 mm above the floor of the previous fiscal year's BAR004 to 226.6 mm above the floor, making the BAR005 the lowest of all the third-era Formula One cars. Although further measures to reduce weight were also pursued in subsequent models, changes in the collision safety regulations turned the focus toward how to reduce the resulting weight increases. Consequently, the year-to-year changes in the height of the center of gravity tended to show either leveling off or slight increases.

With this machine, the basic elements of chassis performance, such as the height of the center of gravity, the weight distribution, and the moment of inertia, as well as the aerodynamic performance, reached the highest level standard. Even though the BAR005 showed signs of being fast during winter testing before the season, weight reduction had resulted in diminished reliability in the gearbox and hydraulic systems, so that Jacques Villeneuve retired eight times out of 16 races while Jenson Button retired five times. The car ended up with fifth place in the constructor's standings. This was the result of pressing hard in development, and taking drastically aggressive measures, in order to catch up with and overtake the top team, and the reliability of the points in question was steadily upgraded in the following year.

7. The BAR006: 2004

Of all the third-era Formula One cars, this was the most competitive machine throughout its season. Of the 18 races, Jenson Button finished with points in 15 races and went to the victory stand 10 times, while Takuma Sato finished with points in 10 races and went to the victory stand once, with the BAR racing team placing second in the constructor's standings. The reasons for this breakthrough were the way in which the BAR005, which had great potential, matured through evolution along normal lines to secure its reliability, and the change in evaluation methods used in aerodynamic development. Target values were set for the variation range so that the aero balance could be maintained constant as much as possible during cornering. The aero balance change from braking to corner exit can increase by as much as 10% in some cars, and these measures were part of an attempt to bring that figure down to zero. The conventional approach was to evaluate aerodynamic pitch sensitivity by mimicking the changes in vehicle height at the front and rear during cornering. This was now augmented with a crosswind mode that subjected the car to a diagonal wind from the front, taking the actual vehicle slip angle during cornering into consideration, and also adding a mode in which the tires are steered. These efforts placed the wind tunnel evaluation modes on a well-established basis. The establishment of these aerodynamic evaluation methods held an important meaning: making the aero balance stable meant that drivers could attack corners with confidence, and this is a key to building fast racing cars. It would be ideal, essentially, if the degree of confidence with which a driver can attack a corner could be expressed in quantitative terms, and if it could then be predicted how many seconds of lap time that would be equivalent to. This quantification, however, is no easy matter, so it is necessary to ascertain and evaluate the causal relationship between the aero balance change and the driver’s confidence.

Advances in methods for analyzing tire characteristics also enabled a better understanding of how to use Michelin tires with the weight distribution configured at 48%. This was another major factor in the upgraded competitiveness of the BAR006.

Another gain in competitiveness came from the front clutch pack. When the driver begins turning the steering wheel while braking, the normal load on the inside front wheel is reduced, and that tire tends to lock up. Once it has locked, the braking distance increases, and not only is lap time lost, but flat spots form on the tire surface, vibrations are generated, and subsequent driving is impeded. The front clutch pack is a system that avoids the above by transferring torque from the outside wheel to the inside wheel in order to prevent locking. It connects the left and right front wheels by means of the clutch. The results for this year were achieved because the basic elements of chassis performance reached their highest level, reliability was upgraded, and on top of that, the car possessed another powerful advantage. Unfortunately, the front clutch pack ended up being prohibited in the following year’s regulations.

8. The BAR007: 2005

Measures were taken with the aim of further increasing the range of adjustment of the weight distribution toward the front by 1% (to a maximum of 48.9%) compared to the previous year’s BAR006, and to reduce the yaw inertia moment (-9%). The result was a packaging layout that moved the front axle 46 mm to the rear while keeping the location of the rear axle unchanged and placing the differential center 5 mm farther to the front. The wheel-base was 3085 mm, and the gearbox was changed from a 3-speed to a 7-speed with seamless shift to reduce the time loss when shifting gears to zero. As the speed of the cars increased year after year, the FIA had adopted a policy of placing limits on it, so that the regulations this year aimed at reducing the downforce by raising the height of the front wings 50 mm and moving the rear upper wing 150 mm forward, new regulations were introduced for the diffuser area, and so on. The FIA was eying a 20% reduction in the downforce, whereas the Honda team decided to limit the reduction target to 10%. Although the target was met, the other cars achieved smaller reductions than anticipated, so there was a considerable gap at the start of the season. Every effort was made in the subsequent development, and a 13% increase in the downforce was achieved in the course of the season, while the year-round figure had usually been 7%. However, the other cars achieved similar advances, and the gap was not closed. The drag had also increased to a maximum, and enhancement of the L/D remained an issue. The lack of downforce resulted in excessive deterioration of the rear tires, which also had an impact on traction and braking performance. In combination with insufficient warm-up of the front tires, this caused conspicuous delays due to OS in high-speed corners and US in low-speed corners. In the later competitions, “push rod on upright” was introduced as a front suspension measure to achieve better weight-shift distribution, and this provided a better grip in low-speed corners. In “push rod on upright,” the push rod is attached to the upright instead of the lower arm, where it is conventionally, and also offset from the kingpin, so as to control the vehicle height and weight shift during steering. Although this measure yielded its effect, it did not increase performance enough to overtake the top team, and this was a year that left unresolved issues in tire handling.

The fourth competition at the San Marino Grand Prix would have been a double victory, the first time in the season for both drivers to earn championship points (Jenson Button in third place and Takuma Sato in fifth). The car inspection after the race, however, found a violation: when all the fuel in the fuel collector tank was removed, the cars came in under the regulation minimum weight (600 kg). The team responded by disclosing their data and insisting as follows: the fuel tank is not a special item, and the car did not go under the regulation minimum weight of 600 kg under any conditions whatsoever while running in the race. The FIA responded that data can be altered and cannot be recognized as evidence. They took the view that, structurally, there was a possibility that the cars fell below the minimum weight, and they deemed the cars to have infringed the regulations. The race results were forfeited, the team was suspended from entering the fifth and sixth competitions – the Spain and Monaco Grand Prix – and the team was suspended from racing for six months, with a probation period of one year. This year, therefore, was also one in which severe rulings were imposed.

9. The RA106 (BAR008): 2006

Honda bought up all the BAR shares during this year, so that it was 100% Honda-owned. The chassis designated BAR008 was renamed during the development stage as RA106. Since regulations required that the engine be changed to a 2.4 L V8, the total engine length was reduced by 90.5 mm, affecting the chassis package. In order to maintain stability, the monocoque and gearbox case alone were lengthened, and a driveshaft with lightweight constant velocity joints was adopted. The wheel-base retained the same 3085 mm length as in the previous year.

With the new engine, the lateral mounting points were spaced with a larger pitch and the chassis lateral stiffness was increased 33% to heighten the stability. In the gearbox, the seamless shift that had been introduced the previous year was expanded to all the gear shifting stages. A direct-push clutch was adopted with the aim of stabilizing the clutch friction performance necessary to control tire slipping at the start, which came with the reduced engine displacement, and this measure also increased acceleration from a standing start. Aerodynamic measures were taken in light of the reduced power with the change to a V8 engine, and a policy change from emphasis on downforce to emphasis on efficiency aimed to enhance efficiency by 13%. In order to obtain downforce without increasing the drag, steps were implemented to increase the rate of flow across the bottom surface of the floor, and the nose was raised higher and the keel was eliminated in order to heighten performance. This setup sought to overcome the excessive wear of the rear tires that had been an issue the previous year by taking steps to maximize the traction performance of the tires, and the front-rear ratio of the weight-shift distribution, which was caused by the lateral force generated when building up speed coming out of a corner, was changed to give it a forward shift. This allowed the temperature of the front tires to be brought up to the working range of 70°C or higher.

The RA106 (BAR008) produced top-level times in the early qualification rounds when the season was beginning, but the car was not producing winning times in the actual races, and this became a conspicuous issue. The effects of aerodynamic enhancement brought out turning performance that exceeded the other cars', but delays in developing technology to reduce rear wing drag at high speeds meant that this machine was far outstripped by the top teams at maximum speed. In the area of tires, the excessive deterioration of the rear tires was resolved. However, warm-up became an issue on certain circuits, making it necessary to further upgrade tire temperature management. Although it was helped by the dropout of the top teams in the 13th race, this year the RA106 did achieve Honda's first win (with Jenson Button) of the third era of Formula One racing. Various measures were taken to upgrade the car's competitiveness even from the 14th race, such as extending the wheel-base by 50 mm at the front axle to enhance its braking stability, and these yielded positive effects, but not enough to put its performance on a par with that of the top teams.

10. The RA107: 2007

Chassis development up to the RA106 had continuity, and consisted basically of carrying on with the previous year's development while adding refinements. In June 2006, however, technical director Geoffrey Willis left the team, and for this among other reasons, the RA107 represented a very challenging attempt to discard the preceding base and build instead from a new concept on up.

Development of the RA107 was begun with "top priority to aerodynamics" as its slogan. The wheel-base, made longer than in the RA106 by shifting the front axle 30 mm forward, was 3165 mm. In order to minimize the chassis inertia, the gearbox interior was shortened and the final gear center was moved approximately 50 mm forward. As a result, the sweepback angle of the driveshaft also became larger than in the RA106.

Significant changes were also made in the layout of the interior where they are not visible from the outside. For example, the oil and water radiators located on either side of the driver inside the engine cover were reshaped to be longer and more slender, slanted forward, and contained inside the side pontoons. A layout was also attempted that led the exhaust pipes from the engine forward from the left and right banks of four cylinders, gathered them together, then turned them outside at a point close behind the driver. While the purpose was to reduce the moment of inertia of the chassis by any amount possible, the increased vibration and noise from the exhaust were not welcomed by the drivers.

Another major change made this year was the return to the use of Bridgestone tires, because Michelin, whose tires had been used for the preceding three years, withdrew from Formula One racing. The differences in the deformation of the Michelin and Bridgestone front tires when running had a greater effect on the aerodynamics than had been anticipated, and the time it took to understand it caused delays in development.

The aerodynamics of the RA107 were the first to be developed making use of the full-scale wind tunnel that entered operation in 2006. At the start of the design process, the objective was development to reduce movement of the aerodynamic center as much as possible, from the point of braking just before cornering up to the zone of acceleration at the exit. When testing began with the actual car, however, the amount of movement was greater than anticipated, counter to the plan, so that the chassis behavior showed unstable characteristics from the point of braking throughout the entire corner. Furthermore, the response emphasized downforce at the sacrifice of aerodynamic resistance, with the result that the car's straight-line speed fell short of the top teams' by 10 km/h or more. A number of changes were made partway through the season in order to enhance the aerodynamic characteristics, prominent among them a change in the shape of the side pods, and the aerodynamic performance did gradually enhance. The other teams, however, also advanced with similar rapidity, and the RA107 did not achieve superiority in performance.

The result was that the team did not mount the victory stand even once throughout the entire season, they earned points three times, and were eighth in the constructor's standings. Even in the Driver's Titles, Button was 15th and Barrichello was 20th. These were the poorest results since the start of the third era.

11. The RA108: 2008

One of the major causes of the lukewarm results achieved by the RA107 was its aerodynamic performance. Therefore, the RA108 also adopted the slogan of "top priority to aerodynamics" and development was begun with the goal of pursuing enhancement of L/D (20% better than the previous year) as well as yaw and pitch sensitivity reduction.

One major change in the development was the introduction of the concept of "usable downforce" as an index for aerodynamic development. In this procedure, wind-tunnel test results conducted under various conditions, including front and rear vehicle heights, are multiplied by weighting factors according to importance. These are then integrated to obtain a single scalar value by which analysis is carried out. Setting up the factors requires expertise, but the method expresses performance using a single numerical value, and it has the advantage that, when further refined, it can be used to keep the development orientation in clear focus. Target usable downforce values were set for each update from the opening race and the major Grand Prix competitions, and development was carried out so as to realize those targets.

The monocoque was made more slender and longer (by 45 mm) to the external appearance, and the side pod locations were also moved toward the rear, largely due to aerodynamic requirements. As a result, the rear axle was shifted rearward while the sweepback angle of the driveshaft was maintained unchanged, and the wheelbase was increased 45 mm over the previous year's to 3210 mm. The nose was also raised and the bottom of the side pods narrowed, resulting in a chassis shape that takes the flow over the chassis bottom into consideration.

However, development that placed top priority on aerodynamics also had negative effects in some areas. For example, the overall width of the gearbox was narrowed out of concern for the flow along the side to the rear wing at the rear of the car. In the rear suspension, this caused a diminishment in the mounting stiffness and deterioration of the damper ratio. Traction control was forbidden starting in this year, and it emerged as an issue of driveability that the engine's low-speed torque did not increase smoothly when the throttle was applied. The RA108 was troubled throughout the season with unstable cornering and inadequate traction. During the last half of the season, in particular, graining of the tires (in which just the tire surface peels off as rubber grains, and these are rounded into roller-like shapes that cause the tire to lose its grip) caused by low-temperature road surfaces also became an issue.

In order to enhance the damper ratio and the stiffness of the rear suspension, the damper was changed to a rotary type. This enhanced suspension was adopted starting with the 11th race – the Hungary Grand Prix – but it did not amount to a fundamental enhancement.

Another factor was that Ross Brawn, who joined as Team Principal in November 2007, adopted the policy of concentrating development resources on the 2009 car. Mid-season upgrades and enhancements to the RA108, therefore, tended to be omitted if they were not items that could be treated as advance tests for 2009.

As a result, the car scored points only five times, and the team came in eighth in the constructor's standings (McLaren was divested of its title due to a spying incident, so the actual ranking would have been ninth) with Rubens Barrichello the only third-place winner to mount the victory stand in the ninth competition, at Silverstone in the rain, when the pit strategy was successful. The driver rankings had Rubens Barrichello at 14th and Jenson Button at 18th place, making this another year of low rankings to follow those of 2007.

12. Conclusion

The decade-long flow of Formula One chassis development in the third era can be generally summarized as follows. The years from 1998 to 1999 were the period of initial involvement with Formula One racing in recent years, with the fundamentals being learned from Dr. Postlethwaite and the Dallara engineers. The years from 2000 to 2001 were a time of working to get the cars at least up to a certain level by reducing the weight in every part in order to reduce the height of the center of gravity. The other main focus was assuring reliability. In 2002, an attempt was made to take a step forward from the fundamentals and address such issues as suspension stiffness and body stiffness. However, the vehicles that were the base for this effort had individual unit variations because of the manufacturing quality, the driving data evaluation methods and related matters did not evolve to match the effort, and the development stalled. The year 2003 brought advances in the manufacturing quality of CFRP and all other parts, as well as a new ability to engage in design while making limited predictions of how it would affect the maneuverability of the overall chassis package, including the engine. This became the year of transition from development of constituent technologies to development with a view to the balance of a single whole vehicle. The year 2004 represented a maturation and an evolution along normal lines from 2003, with new aerodynamic evaluation methods proving their worth. The car reached the top level. The second place in the constructor’s ranking earned in this year, however, turned out to be a peak. In 2005 and thereafter, efforts were made to overcome the differences between phenomena in the actual machine and testing in a wind tunnel, and to determine techniques for obtaining the optimal performance from the tires, but these core technologies proved elusive, and the car slipped back in the rankings. It became apparent in 2007 that the regulations on aerodynamics would be changed significantly in 2009. This was taken as an opportunity to recover from a moribund situation, and work was begun on development of 2009 specification aerodynamics. The year 2008 turned out to be the final year of the third era. Although the car finished eighth in the rankings, the preparations for 2009 were steadily taking shape, and after Honda’s withdrawal, that technology was inherited by the Brawn GP Formula One team and contributed to their finish in the two top places in the opening race.

Formula One development in recent years has been a puzzle about how to bring mutually antagonistic factors that govern car performance into balance with each other so that the level of the car as a whole can be raised. In order to achieve a high-level balance among complexly interrelated factors, it is necessary to evolve technology that can accurately grasp the degree of contribution made by each individual factor, as well as the relative merits of different combinations of factors in terms of total performance. In the latter part of the third era, efforts were therefore concentrated on understanding of the gaps that exist between bench test results and simulations on the one hand and the actual machine on the other, as well as establishing methods of evaluation. Step-by-step tests and simulations had to be constantly repeated in order to obtain accurate tools and judgment criteria. Once these are in hand, a team will not lose sight of the direction of development, no matter how the regulations may change. Furthermore, any reduction in the necessity for tests using the actual machines enables a reduction in development costs, as well. As people increasingly call for the cost of Formula One development to be lowered, this style of development will no doubt continue to accelerate.

The movement to reduce development costs and to perform evaluation of cars as much as possible on the desktop and using bench tests is by no means limited to Formula One development. This is clearly a major trend in the development of mass-production vehicles, as well. It is a matter of firm belief, therefore, that the knowledge acquired during this 10-year period of development can definitely be used to advantage in the development of mass-production vehicles. All of that knowledge cannot be related in the pages of the Technical Review, but it will be fortunate indeed if some hints of it can be conveyed herein.

■ Author ■


Akio TONOMURA


Shun KAWABE


Nobuhisa NAKAYOKU

Development of Materials during Third Formula One Era

Masami HOSHI*

ABSTRACT

The realization of high-power and high-efficiency powertrains, lightweight and compact vehicle frameworks, and a high level of durability and reliability are necessary to increasing the competitiveness of Formula One race vehicles, and material technologies are an essential factor in this. Honda has developed high-strength, high-stiffness, and low-density materials such as titanium-aluminum and aluminum metal matrix composite (MMC) materials in order to achieve weight-savings in structural parts. The company has also developed low-friction and high-durability surface modification technologies, in particular DLC coatings, for friction materials. For use in hybrid systems, Honda has developed magnetic materials for use in motors and highly functional materials such as high-thermal-conductivity materials. These material technologies have enhanced vehicle performance and increased race competitiveness.

1. Development of Materials during Honda Formula One Third Era

The role of Honda's Materials Division in the company's Formula One activities during the 1980s centered on the development of technologies for the enhancement of materials that would enhance engine performance. In the process of development of race vehicles for the company's third era activities commencing in 2000, that role evolved to the development of advanced materials of the type used in the aerospace industry and their application throughout the vehicle, from the powertrain onwards. This process involved rapid changes of a type not previously experienced, and the Formula One third era became a demanding period in which the Formula One team faced a variety of technological development challenges.

At the beginning stages of Honda's third era activities, the European and U.S. teams had already established a technological foundation in advanced technologies developed in the aerospace industry, and were striving to apply these technologies in Formula One. Having ceased its Formula One activities, Honda was delayed in its own efforts in this direction, and the company fell behind its competitors in terms of technological advancement. Precision casting technologies for titanium and steel materials for gears and cranks can be pointed to as representative examples. Because of this technological lag, the enhancement of Honda's development program to enable the company to quickly catch up with and overtake the leaders in the field, the European teams, was an important issue at the time. Given this, the Materials Division actively sought out and introduced not only domestic but also European and U.S. technologies, and proceeded with development programs aiming to create materials technologies surpassing those of other teams. While a later change in circumstances reduced the need for precision casting technologies, in the area of structural steels the company developed high-strength gear materials and high-toughness steels for use in cranks, and began to employ them in races. Later, the practical use of high-strength, high-rigidity materials such as titanium and aluminum metal matrix composite (MMC) materials became feasible, and Formula One teams entered a stage of fierce development efforts. Although research efforts in these areas had diminished in Japan, these were fundamental technologies for the aerospace industry, and research was ongoing in Europe and the U.S. Honda also made concerted efforts at development, and established its own processes for manufacturing high-strength materials. This resulted in the development of titanium-aluminum valves and piston pins and aluminum MMC pistons, which were subsequently employed in race vehicles.

In 2002, Honda made the transition from conventional bucket tappet valvetrains to rocker arm valvetrains, which enabled the achievement of increased engine speed and enhanced performance. Tribological technologies for the friction components formed an important aspect of these development efforts, and surface modification technologies were an essential research focus which would affect the feasibility of the new mechanisms. As in the case of structural materials, Honda sought out technologies, including technologies developed overseas, and developed a diamond-like

* Automobile R&D Center

carbon (DLC) coating as a new surface modification technology. This technology significantly increased the durability of the friction components of the camshaft and rocker arms of the new mechanism, and contributed to making it technologically feasible.

Moving in another direction to the development activities outlined above, discussions intensified among the Federation Internationale de l'Automobile (FIA) and participating teams regarding the reduction of the excessive development budgets resulting from competition for technological development, and from 2006 materials regulations were enhanced.

In order to minimize the possibility for losses from having technologies that had already been developed and put to use rendered ineligible for use, Honda continued development efforts situated within the scope of the materials regulations. As a result, new ideas emerged, and Honda proceeded with the development of conrod bearings using a high-thermal-conductivity material as the back metal and technologies for the enhancement of the performance of the DLC coating. These technologies increased powertrain efficiency. In particular, in the area of DLC coatings, a technology resistant to high surface pressures was developed enabling the previous limit in surface pressure to be exceeded. The application of this technology from the shift to the bevel gears of the transmission increased Honda's race competitiveness. In the period from 2007, the final period of the company's third era activities, Honda also focused on development efforts in the area of functional materials for the development of hybrid systems, the use of which changes in regulations had made possible from 2009 onwards. Aiming to reconcile hybrid device performance, the framework structure, and durability, the company conducted development efforts in the areas of magnetic materials for use in motors and high-thermal-conductivity materials to enhance the cooling performance of power control units, and succeeded in developing new materials technologies.

2. Direction of Materials Development

This chapter will discuss the direction of development of materials technologies for race vehicles as a whole, including the technologies discussed above. The elements composing race vehicles are engines, transmissions, chassis, and hybrid systems, and as Fig. 1 shows, there are four basic directions for the development of materials enabling the realization of the necessary characteristics of each element. It is essential to maximize the functions of these basic materials and thus enhance the competitiveness of the race vehicle.

High-strength, high-stiffness, and low-specific-gravity materials, both metallic and non-metallic, are a key technology in the achievement of improved performance and durability in race vehicles, from the engine and transmission components and the framework structures to the suspension and the brakes.

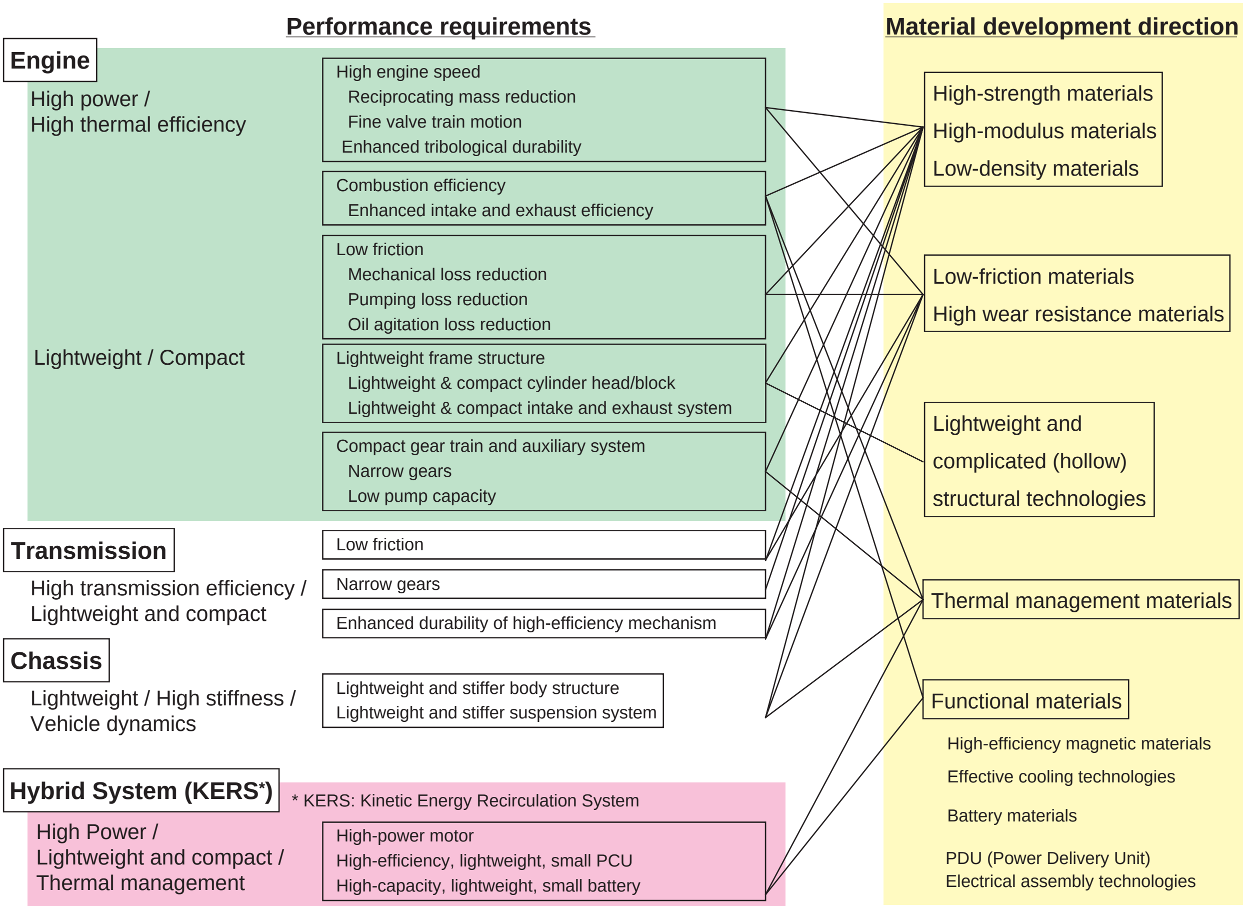


Fig. 1 F1 material development

Low-friction and low-wear materials are an important technology directly related to the achievement of enhanced vehicle efficiency by reducing friction resistance in the engine and in friction components throughout the vehicle and enabling components to be used under extremely severe conditions.

Thermal management material technologies, which enable control of the heat balance, are important in enhancing the functions of each part of the vehicle, including the enhancement of thermal efficiency around the engine and brakes and the cooling of the hybrid system. This is achieved, for example, by means of exploitation of the thermal conductivity of base materials and the heat absorption and dissipation characteristics of surfaces.

In the area of functional materials, magnetic materials for use in fuel injectors are necessary in engines, and magnetic materials, materials with excellent thermal conductivity and thermal insulation characteristics, battery materials and others are important technologies in enhancing the functions of the motors, PCUs, and energy storage devices used in hybrid systems.

3. Conclusion

For examples and more detailed discussion of Honda’s work in the development of materials technologies, the reader is referred to the special edition of the Technical Review dealing with materials[(1)-(18)]. While few of these materials are employed in mass-production vehicles due to cost, the knowhow and guidelines in the area of element technologies gained from their development do encompass aspects that can be extensively used in development for mass production. In conclusion, the author wishes to express his sincere gratitude to Honda’s suppliers, both domestic and international, for their generous assistance in these development projects.

References

(1) Ito, K., Asami, M., Koinuma, H.: High-performance Shell Bearing from New Material, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 240-242

(2) Mizoue, K., Kawahito, Y., Mizogawa, K.: Development of Hollow Crankshaft, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 243-245

(3) Munemura, T., Yamada, H., Ohnuma, T.: Development of Titanium Exhaust Pipe, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 266-268

(4) Konagaya, D., Tanahashi, T., Tanaka, T.: Development of High-performance Gear Materials, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 271-272

(5) Sanada, T.: Development of Titanium Hollow Valve and Study of Sodium-potassium Valve, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 257-259

(6) Murai, Y., Tsuchiya, M.: Development of Lightweight and Low-friction Resin Materials for Oil Pump Rotor, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 269-270

(7) Murai, Y.: Research on Coolant Using Nano Additives and Heavy Water, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 271-272

(8) Murai, Y., Okumura, M.: Development of High Thermal Conductivity Material for Heatspreader, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 278-279

(9) Tsuchiya, M., Tsunashima, S., Hashimoto, H.: Development of Titanium Aluminide Valve, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 254-256

(10) Ito, N., Yonehara, M., Yakubo, K.: Development of DLC Coating on Camshaft and Rocker Arm, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 260-262

(11) Miyoshi, T.: Development of High Induction Stator Core, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 280-281

(12) Ishizuka, K., Kusakai, K., Imai, N.: Development of Metal Matrix Composite Piston, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 246-247

(13) Minato, S., Onaha, Y.: Development of Lightweight Titanium-aluminide Piston Pin, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 248-250

(14) Minato, S., Mizogawa, K.: Development of Hollow Titanium Connecting Rod, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 251-253

(15) Imai, N., Nishida, H.: Development of Laser Clad Welded Valve Seat, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 263-265

(16) Tanahashi, T., Onaha, Y.: Development of DLC for Transmission Gears, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 278-279

(17) Yamada, Y.: Weight Reduction Study for Side Panel of Formula One Monocoque, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 282-285

(18) Yanase, A., Yamada, H.: Development of Brake Caliper Production Process with High Strength Al-Li Material, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 286-288

■ Author ■

Masami HOSHI

Descriptions of Engine Technologies

Explanation of Honda's Third Era Formula One Engine Development

Kiyoshi KONDO* Yasuhiro MOTOHASHI* Kazushi OGIYAMA*
Hajime ENDO* Satoshi NAKAMURA* Nagao YANAGISAWA*

ABSTRACT

During the third era Formula One engine development, as a measure for increasing the engine power through an increase of engine speed, the enhancement of the induction static efficiency and weight reduction of the moving parts was focused on. From 2007, restrictions on the maximum engine speed and development of the engine except for the induction and exhaust system were introduced. Therefore, the focus of the development changed to practical use of dynamic effect of induction and exhaust pressure. Moreover, the driver-aids including traction control was banned in 2008, so attention moved to the development with the objective of shortening lap times, and including drivability enhancements.

Also, to comply with new regulations concerning the longer use of engines, the restriction of use of materials, and to respond to changes in actual usage in circuit running in accordance with the change of regulations, the subdivision of the endurance mode and results feedback was performed, which achieved durable reliability in parallel with performance enhancement.

This document describes the evolution of the engine as outlined above, and includes descriptions of technology such as the lap time evaluation method and tools.

1. Introduction

Honda's third era Formula One activities began in 2000, with Honda competing with the other constructors to develop engines with higher engine speed and output. In 2006, immediately before the International Automobile Federation (FIA) introduced the engine speed restrictions, some competitors had achieved a maximum engine speed of 20000 rpm. At that time, the maximum engine speed that Honda had reached in a race was 19600 rpm.

In order to competitively increase engine speed, technology was required for reducing the reciprocating mass of the valve train system and the reciprocating system. To achieve this, in addition to developing materials and manufacturing technology, predictive computer aided engineering (CAE) was introduced for each part. In particular, a system had been firmly established by 2003 for designers themselves to produce 3D models and perform CAE analysis, enabling design that is optimized for structural strength, resonance, flow rate, temperature distribution, thermal stress and other factors to be performed in a short period of time. Induction potential was also required to increase engine power. To achieve this, a higher peak power engine speed was required, and technology that increases induction efficiency was focused on.

Because friction and vibration increase quadratically with the rise in engine speed, attention was also given to technology that reduces mechanical loss, and to dealing with a vibration acceleration that reached 300 G.

Engine failure on the circuit may cause immediate retirement from the race, so high reliability was also required. The load on the engine was estimated for each circuit from the throttle opening angles and the duration of continuous wide open throttle (WOT), a driving mode suitable for the load was formulated, and then an endurance test on a test bed was performed.

From this, operating conditions for the race such as the temperature of oil and water and the individual ignition timing for each cylinder were established in advance, and measures were taken to help ensure high performance and reliability, and make one engine last for two races.

In 2007, the FIA restricted the maximum engine speed to 19000 rpm, and at the same time enforced a freezing of development of the main engine parts (homologation).

After that, attention moved to output characteristics that would shorten lap times, concentrating on the development of induction and exhaust system parts. Simulation tools for lap times and fuel efficiency were

* Automobile R&D Center

also developed for simulating circuit running conditions on dyno.

In 2008, the use of a common ECU was made compulsory and traction control was banned. This gave rise to issues of drivability when coming out of corners. Work was started on enhancing drivability by modifying the engine hardware, but as of 2009, this development issue remains.

2. Power

2.1. Era of High Engine Speed and High Output (2000 to 2006)

2.1.1 Increased engine speed and enhanced induction performance

If Pse: Shaft output, Pmi: Mean effective pressure, Vs: Stroke displacement, Ne: Engine speed, Psf: Friction, then: $Pse = (Pmi \times Vs \times Ne/2) - Psf$.

Thus, an effective way to increase the power is to raise the engine speed, and Honda competed with the other teams in increasing the engine speed and power.

Figure 1 shows the trend for the maximum engine speed.

Honda's V12 engines in 1992 had an engine speed of only 14400 rpm, but by the time the third era started in 2000, the level was 17000 rpm. Although the introduction of mileage extension regulations to make one engine last over one race event, and then two race events, slowed this growth, by 2006 it had still reached 20000 rpm in tests on dyno. However, for reasons of lap time contribution and engine durable reliability, an upper limit of 19600 rpm was set for the actual races.

When the target of the maximum power and engine speed are fixed, the power peak engine speed is determined from maximum engine speed, and the output value can be estimated as shown in Fig. 2. This means that the target output shortfall should be compensated for by reducing the friction and enhancing the combustion.

The induction performance target for the power peak engine speed was decided using a steady flow test, and development was performed of the cam profile and port layout, including the valve diameter, that was required for achieving the desired horsepower by controlling engine speed.

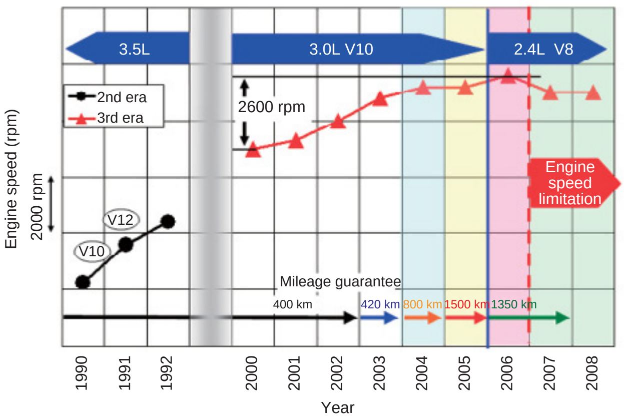


Fig. 1 Transition of max engine speed

The valve diameter size was decided from a good balance with the cylinder bore diameter, which was determined by also taking the piston speed into account.

The bore diameter started at 95 mm in 2000, but by 2002 it had been increased to 97 mm. In 2004, laser-clad welded valve seats were used in order to increase the valve diameter.

From 2000 to 2001, the tappet type was used for the valve train system, but because the valve lift was limited by the lifter bore, in 2002, the valve train system was changed to the rocker arm type in order to increase valve lift.

Cylinder head port development continued throughout the season, and over 10 types were evaluated. However, just because the induction performance was high in the steady flow test, this did not mean that the power would increase, and so development for enhancing the combustion, including fuel distribution, was required.

Immediately before the engine homologation in 2006, power had been increased by developing head port shape that maximized the dynamic effects, using not only the steady induction performance, but also 3D-2D coupled ηv simulation.

2.1.2. Making moving parts more lightweight for increasing the engine speed

With the increased engine speed, moving parts needed to be made more lightweight from the viewpoint of reducing friction and durable reliability. To make these parts more lightweight, materials with high specific stiffness and high-temperature fatigue strength were used, and their shapes were optimized using CAE.

In 2004, an aluminum Metal Matrix Composite (MMC) material with increased high-temperature fatigue strength was used for the piston, reducing it to a weight of 210 g at a size of ϕ 97.

After the engine mileage extension and the banning of MMC materials in 2006, the piston weight increased, but through CAE thermal stress analysis and with the enhancement of piston cooling using oil jets, a weight of 230 g was achieved using A2618 enhanced materials.

At the same time, reducing the shaft diameter was studied to minimize heat generation in the connrod bearing in the reciprocating system, and to achieve a good balance with stiffness, it was reduced to ϕ 34. Enhancements for durable reliability were also performed for the bearing itself, with a copper alloy with high thermal conductivity being used from 2005.

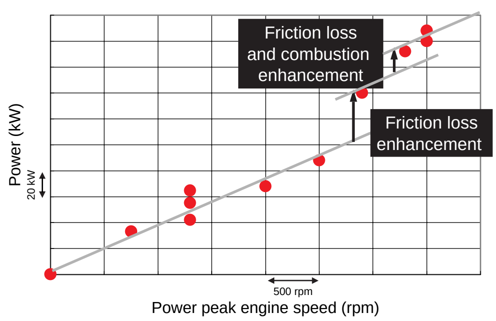


Fig. 2 V10 F1 engine power peak engie speed - power

The factors that were important in reducing the reciprocating mass of the valve train system were the use of rocker arms for the mechanism and the use of Titanium-Aluminide (TiAl) as the valve material. The reciprocating mass was reduced from 71.4 g in 2000 to 47.2 g. Even after the regulations on intermetallic compounds that were introduced in 2006, an equivalent reciprocating mass could be achieved with a titanium alloy using CAE analysis.

Figure 3 shows the positioning of the specific stiffness for the materials that were used in races. From the perspective of cost reduction, the specific stiffness was reduced to less than 40 GPa/g/cm^3 in 2003, and in 2006, the use of MMC materials, intermetallic compounds, and magnesium alloys was banned. Regulations were also introduced for the main elements of the piston materials, allowing only conventional materials to be used.

2.1.3. Combustion

In Formula One engines that require higher engine speed and power, increasing the bore diameter is an effective method of enhancing the induction and exhaust performance.

However, increasing the bore diameter also increases the combustion duration, which leads to decreased thermal efficiency, power, resistance to misfiring, and the like. As a result, measures for making combustion faster were required. In the third era Formula One engine, the bore diameter was increased to ϕ 97, and combustion was enhanced to raise the compression ratio to 13.0. In 2003, a compound angle was added to the inlet valve to enhance the in-cylinder flow and make the combustion faster. Figure 4 shows the effects of shortening the main combustion duration.

2.1.4. Friction

When reducing the engine friction, a factor that had a particular effect was the friction in the valve train system. The behavior of the valves was enhanced by reducing the reciprocating mass and minimizing the angular velocity fluctuations, which reduced the spring load. In addition, the oil agitation resistance in the cylinder of the Pneumatic Valve Return System (PVRS) was reduced. These measures reduced the valve drive system friction of 35 kW in the V10 engine to 12.8 kW by 2005.

Further studies were performed for reducing the agitation and shear resistance of oil, the sliding friction of the journal and reciprocating system, and the ancillary drive loss.

2.2. Power Curve Characteristics for Shortening Lap Times (from 2007)

2.2.1. Lap time simulation

In 2007, the development of new systems for the engine except for the induction and exhaust systems was frozen in order to reduce costs, and the parts that could be developed were limited to the air box, exhaust, and fuel systems.

After the maximum engine speed restriction was introduced, to maximize the team's competitive strength, the use of the maximum engine speed was extended to the 7th gear end of straight. Figure 5 shows the changes in engine speed for each gear level, while Fig. 6 shows

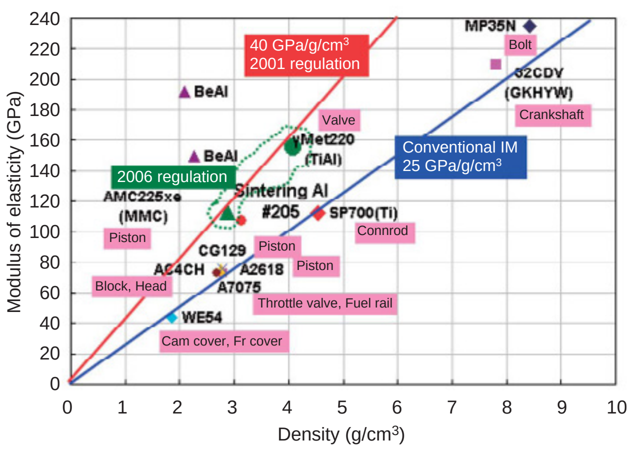


Fig. 3 Comparative rigidity of race engine materials

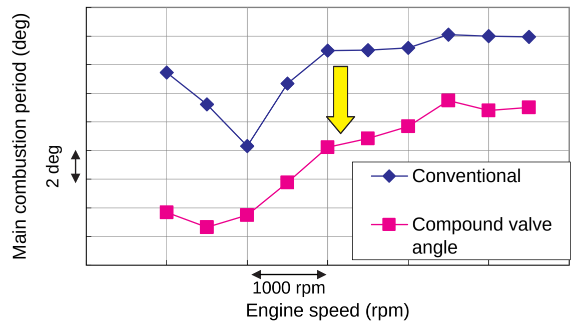


Fig. 4 Effect of compound inlet valve angle

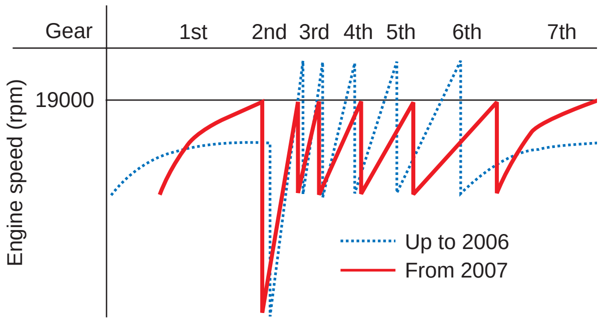


Fig. 5 Change of engine speed usage

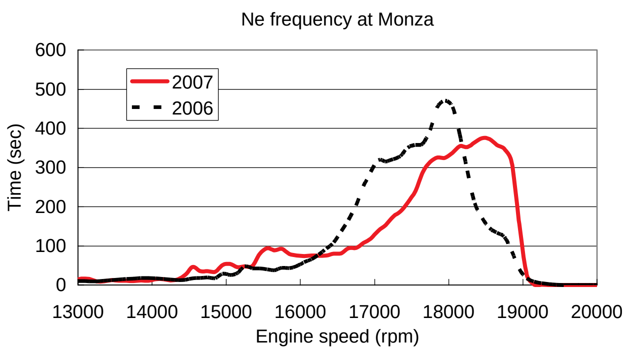


Fig. 6 Ne frequency difference due to maximum engine speed restriction

the engine speed frequency.

Figure 7 shows a comparison of lap times when the end of straight engine speed is 18500 rpm and 19000 rpm. It can be seen that at every circuit, the higher the end of straight engine speed, the shorter the lap times.

Figure 8 shows the effect of the power peak output engine speed on the lap time and maximum speed at the Shanghai circuit. It can be seen that the best lap time is at 18200 rpm, and that the time drops sharply with engine speeds below this level.

The shift-up engine speed, end of straight engine speed control (EOS) settings and gear ratios were optimized for each Formula One circuit to shorten the lap times.

Also, it became possible to theoretically calculate how the power curve characteristics would affect the lap times, and the calculation results were reflected in the development.

For the Monza circuit, with the power curve characteristics shown in Fig. 9, even though the power curve characteristics of Curve B is 12 kW less at the horsepower peak, the lap time was shortened by 0.04 seconds.

In the development, the lap time calculation was performed before the race was held in order to optimize the power curve characteristics.

2.2.2. Drivability

In 2008, the use of a common ECU was made compulsory, and driver aids were banned. In the engine, attention was focused on the fact that the banning of traction control and restrictions on control resulting from the common ECU affected lap times because of issues of drivability (abbreviated here to “DR”) when coming out of corners. Figure 10 shows two DR issues: (a) the initial torque following at 8000 rpm, and (b) the torque hole generated by misfiring as the air-fuel ratio becomes richer from 11000 to 12000 rpm.

It was thought that this decrease in DR would lead to a drop in the lap time of about 0.2 seconds.

Regarding these DR issues, for issue (a), spit-back was reduced by the early closure of the inlet valve and fuel sticking to the port wall was enhanced by changing the injector form to a beam. For issue (b), the amount of residual gas in the cylinder was reduced by tuning the exhaust, and the torque characteristics were enhanced by the early closure of the inlet valve. These addressed the issues of torque following and inconsistent combustion, reduced the disturbance from transient throttle operations, and enhanced the DR.

2.3. Engine Oil and Fuel Development

2.3.1. Engine oil development

Engine oil development was important in order to reduce friction and achieve durable reliability. Joint development was performed with Nippon Oil Corporation.

The power can be increased by lowering the base oil viscosity and the HTHS viscosity, but lower viscosity results in decreased durable reliability. As such, selecting the base oil and developing oil additives suitable for the parts and coating materials were performed continuously. Oil deterioration was not an issue because the replacement intervals were short.

2.3.2. Fuel development

The number of regulations, including for distillation characteristics, has increased since 1995, eliminating the prospect of increasing the output through fuel development. This situation continued for fuel

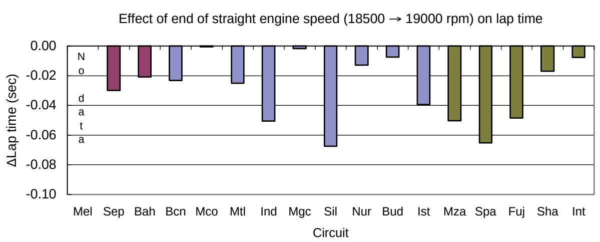


Fig. 7 End of straight Ne - lap time

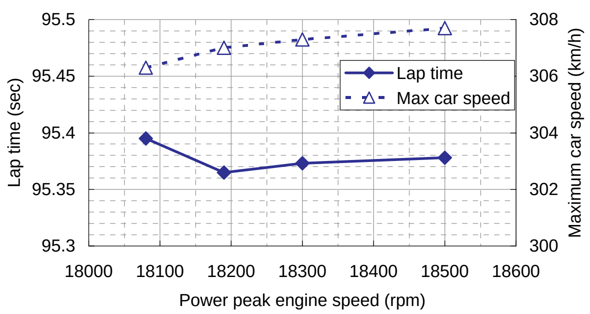


Fig. 8 Pe peak Ne - lap time / Vmax

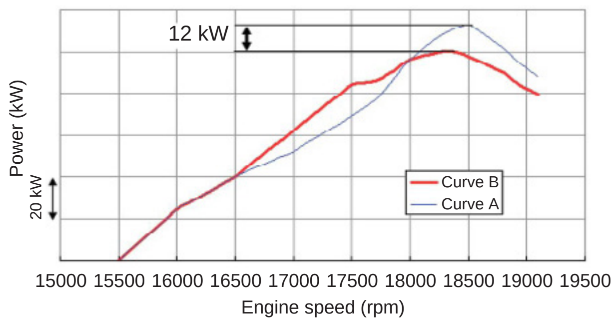


Fig. 9 Power curve shape comparison

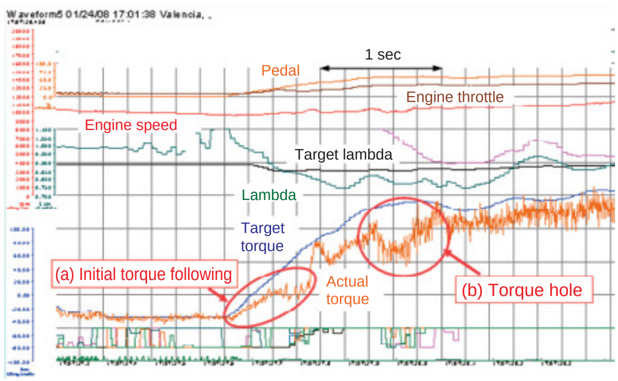


Fig. 10 Drivability issues

development in the third era of Formula One. From 2003, a small modification by mapping the fuel properties in line with the engine characteristics was performed, within the regulatory boundaries. In addition to power, specific gravity from the perspective of the fuel efficiency and fuel weight when the vehicle is filled with fuel were considered. Also, properties were studied that would be suitable for the atomization due to modified fuel pressure and injectors.

From 2008, the use of bio-gasoline (5.75% additive) became compulsory in Formula One, in consideration of environmental issues.

2.4. Achieved Power

Figure 11 shows the achieved power and power per liter for every year. The power increase trend became lower due to the engine mileage extension and engine development regulations, but even so, the power increased every year. By 2008, Honda had achieved a power increase of 23% over its level in its first year of participation in 2000.

3. Reliability

3.1. Guaranteed Endurance Distance

In the third era Formula One, engine operation (use distance) has changed greatly since 2004 due to changes in the regulations. For this reason, each year a review was held of the guaranteed durable reliability distance for the engine. Descriptions of the specific regulations for each year are omitted here, but the required guaranteed distance for each session was determined from the previous year's results and the regulation changes, and was used in the endurance tests on dyno (Fig. 11).

3.2. Guaranteed Endurance Mode

In order to guarantee the Formula One engine durable reliability, an endurance test mode needed to be set. However, this could not be set only through engine operation (use distance); the loads of each circuit also had to be taken into account.

In the third era Formula One, loads were classified into 5 levels to guarantee durable reliability.

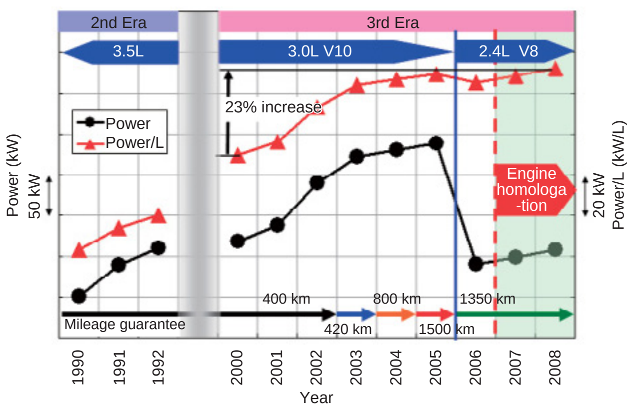


Fig. 11 F1 engine performance transition

Table 1 Endurance test loads classification

Load classification	Condition of wide open throttle		Typical circuits
	Frequency	Continuous time	
Minimum load	Less than 50%	Less than 10 sec	Monaco
Low load	Less than 70%	Less than 13 sec	Magnycours, Budapest
Medium load	70% and over	13 sec and over	
	Less than 70%	Less than 13 sec	Bahrain, Barcelona
High load	70% and over	16 sec and over	
	Less than 70%	16 sec and over	Istanbul, Fuji
Maximum load	70% and over	16 sec and over	Monza, Spa

Table 1 shows the load classifications (5 levels) in the endurance test. Up to 2004, the guaranteed durable reliability was evaluated with one endurance test mode for each load. However, from 2005 the regulation changed to one engine for two races, so the endurance test mode combined different loads to determine the guaranteed durable reliability.

Included in this, verification was performed on dyno for various operating conditions such as the temperature of oil and water, the individual ignition timing for each cylinder, the air-fuel ratio, the ignition cut and retard amount (T/C and O/C control) settings, shift cut during blipping, end of straight engine speed control (EOS), torque-drive, over-revving, idling, and the oil replacement timing, in order to maximize performance.

As a result of the endurance test described above, the engine speed frequency (Ne frequency) was calculated, and this was used as the reference for operating in the actual race. Figure 12 shows the representative Ne frequencies for the endurance test results and the race results.

With the increased engine mileage, the current method for confirming the guaranteed durable reliability takes a lot of man-hours and time. In the future, it will be necessary to take measures to guarantee durable reliability by using Miner's rule.

3.3. Reflecting the Endurance Test Results

After the endurance test on dyno and the end of the race, the changes in dimensions of various parts such as the tappet clearance, the wear amount of the piston clip groove, gudgeon pin hole diameter and connecting rod bearing, and the piston temperature were all measured. These were then progressively reflected in engine hardware studies and in operating condition settings such as the ignition timing settings and the engine oil

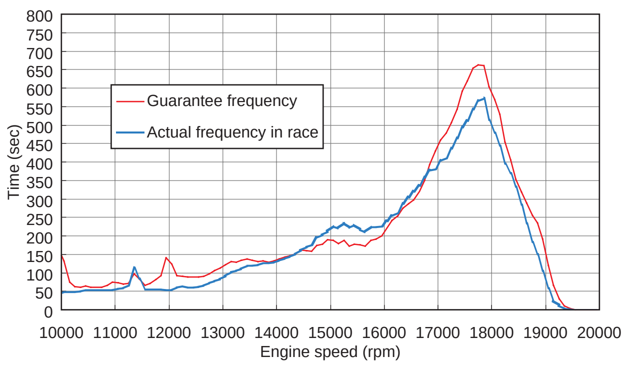


Fig. 12 Engine speed frequency (2006 Rd 17-18)

temperature control. Figure 13 shows the differences in the piston roof temperature in line with the cylinder bias (retard amount from the set ignition timing). It can be seen that by advancing the average ignition timing for all cylinders by 1.2 degrees, the roof temperature rises by about 10°C. Feedback was performed for these results, and the optimum balance for material strength was set for each circuit based on the power changes, cylinder pressure changes, and temperature changes that occurred due to the cylinder bias.

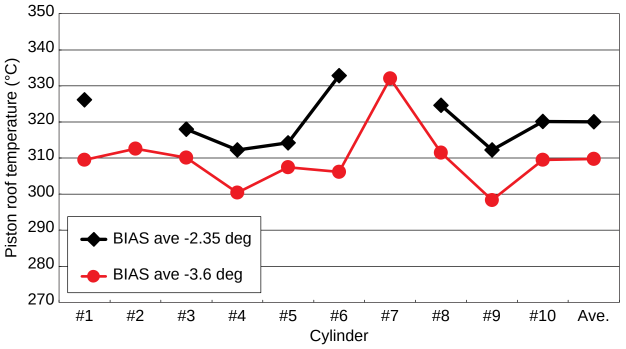


Fig. 13 Cylinder BIAS - Piston roof temperature

4. Conclusion

In the third era Formula One development from 2000 to 2008, high performance Formula One engines were developed that achieved high engine speed and high output, and durable reliability for one engine over two races, with the following results.

(1) An increase in the specific output of more than 230 kW/L was achieved by increasing the engine speed, using the dynamic effects of the induction and exhaust pressure wave, enhancing the combustion, and reducing friction in various areas.

(2) Circuit simulations were used to optimize the use of the power peak engine speed, shift-up engine speed, and end of straight engine speed for each circuit, and lap times were shortened. Also, knowledge was gained about the output characteristics for shortening the lap times.

(3) Verification of various operating conditions was performed on dyno through the segmentation of the endurance evaluation mode, maximizing performance and providing durable reliability.

(4) In response to regulations regarding material restrictions and extension of the use of engines, durable reliability was provided by using CAE technology and through the development of engine oil.

Acknowledgements

The authors would like to express their deep gratitude to Nippon Oil Corporation for their cooperation in engine oil development and in the transportation of oil for the Grand Prix.

■ Author ■

Kiyoshi KONDO

Yasuhiro MOTOHASHI

Kazushi OGIYAMA

Hajime ENDO

Satoshi NAKAMURA

Nagao YANAGISAWA

Technical Description of Formula One Engine Structural Design

Kiyoshi KONDO*　Hajime ENDO*　Tetsuo GOTOU*

ABSTRACT

Through the third Era of Honda Formula One challenge, it was the technical trend that the Formula One engines were considered to be one of the elements of entire race vehicle. The engines were carefully designed to have such characteristics as compact size, light weight, low center of gravity and enough stiffness. These characteristics of engines were very important to increase aerodynamic and vehicle dynamics performance in order to reduce lap times on the race tracks. Moreover, Honda engines were always expected to have the highest power output amongst the all Formula One engine constructors.

This paper describes how the engines, including the lubricating and cooling systems, were designed from the early stage of development in order to satisfy all demands above.

1. Introduction

Figure 1 shows the positioning of the engine in Honda's 2006 Formula One vehicle. The engine formed one element of the overall vehicle framework, for it was rigidly mounted on the monocoque and the gearbox by bolts. Therefore, it was important to reduce engine size and increase its stiffness for the sake of the aerodynamic and dynamic performance of the vehicle.

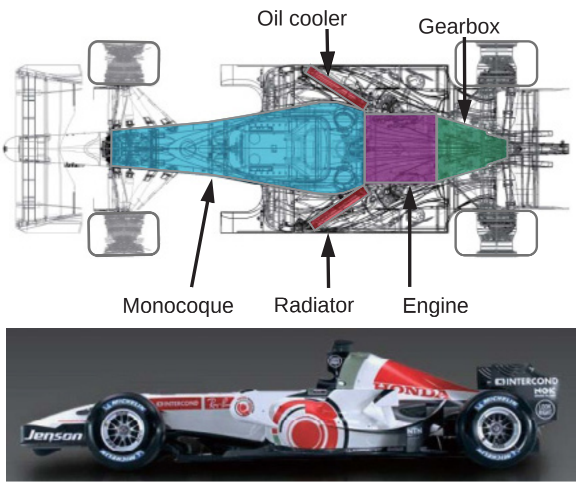


Fig. 1　F1 chassis and powertrain configuration

In addition, Formula One regulations state that a race vehicle must weigh no less than 605 kg with the driver aboard. The engine represents approximately one-sixth of the total weight of the vehicle, and the achievement of engine weight savings was therefore of great significance. If the engine weight was reduced to the limit considering reliability, it would be possible to increase the weight of the ballast which is positioned around the floor and near the center of gravity of the vehicle by the same amount as the engine weight reduction. This would not only reduce the vehicle's roll and yaw moments, but would increase the degree of freedom in setting the vehicle's weight distribution, and would thus contribute to enhanced dynamic performance. For the same reasons, a low and forward center of gravity was demanded in the engine itself.

The stiffness of the engine was also important as same as compactness, light weight, and low center of gravity. The dynamic performance as the whole vehicle is greatly affected by its lateral bending stiffness. Therefore, increasing stiffness in the engine can contribute significantly to the dynamic performance of the vehicle, for the engine itself is one of the vehicle framework components.

Figure 2 shows an overview of the engine as installed, with its lubricating and cooling systems. The cooling water is cooled in a radiator positioned on the left side of the vehicle, where the wind is

* Automobile R&D Center

Table 1 Basic engine configuration and dimensions from 2000 to 2008

Year	(Mugen)	2000	2001	2002	2003	2004	2005	2006	2007	2008
Capacity - configuration	3.0L-V10	3.0L-V10						2.4L-V8		
V-angle (deg)	72	80		94	90					
Length (mm)	620	588		600	604.5	581.5		490		
Weight (kg)	122	111.7	108	111	99	90.9	88.6	95.2	95.1	
COG (mm)	194.7	191.3	191.3	177	177.2	171.6	163.5	165.5	165.1	
Bore (mm)	94.4	95		97						
Bore pitch (mm)	106	106		108	106	103		106.35		
Bore to bore (mm)	11.6	11			9	6		9.35		
Crankshaft height (mm)	70.5	68.5	68.5		66	63	58.5			

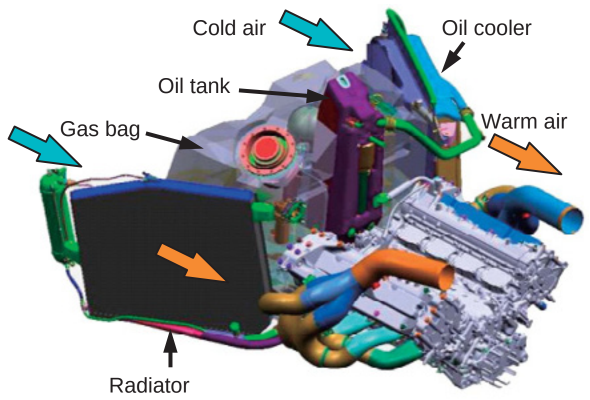


Fig. 2 Engine installation with cooling system

passing through the side pontoon. The engine oil cooler is positioned on the right side of the vehicle with the gearbox oil cooler. The engine oil tank is mounted at the front of the engine, and is fitted in a recess in the monocoque. The entire layout was optimized in order to reduce lap times on the circuit, and almost the same layout is adopted by all teams in current Formula One racing.

2. Engine Technologies

2.1. Engine Size, Weight, and Height of Center of Gravity

Figure 3 shows annual changes in the total length, weight, and height of the center of gravity (COG) of Honda Formula One engines. Table 1 shows the engine

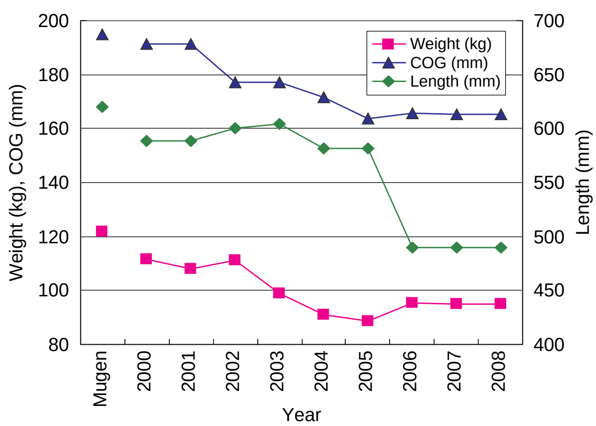


Fig. 3 Change in engine length, weight and COG

specifications that determined these parameters.

The plots at the left of the graph are figures for the Mugen Honda engine supplied to the Jordan team in 2000, with a total length of 620 mm and a weight of 122 kg. Honda works engines returned to Formula One in the same year, and were supplied to the British American Racing team. Almost every engine part, including the auxiliary parts, was newly developed with a focus on the achievement of compactness, light weight, and a low center of gravity. This helped to reduce the engine's total length to 588 mm and its total weight to 112 kg. However, in 2000, other teams were already using engines weighing less than 100 kg. Honda's engines of the time were rather large and heavy.

Variety of technologies were applied to reduce total length, weight, and COG every year. As a result, in 2005, Honda achieved the engine weight of 88.6 kg, the lightest at the time. However, in 2006, FIA regulations were revised to change the engine type from 3.0 L V10 to 2.4 L V8. At the same time, limits of at least 95 kg, 165 mm, and 106.5±2 mm were set for the engine weight, COG, and bore pitch, respectively. This put all the teams on basically the same level, and no further changes have been made since then. This paper will discuss individual framework-related technologies for each item below.

2.1.1. Reduction of total engine length

The total length of the engine is basically determined by the bore, the bore pitch, and the cam gear train layout.

The bore diameter is determined by the maximum engine speed, by the diameter of the intake and exhaust valves which would enable the achievement of the target power of the engine, and by the valve layout. The bore diameter was once increased from ϕ95 in 2000 to ϕ97 in 2002 during the third era of Honda Formula One. ϕ97 was an optimal setting for engine speed of 19000 rpm. All engine constructors appear to have used bore diameters of between ϕ97 and ϕ98 when engine homologation was introduced in 2007.

The method of cooling between the cylinders is one factor that determines the bore pitch, and it is also an important element affecting the concept of water flow throughout the engine. Figure 4 shows the evolution of engine water cooling systems. From 2000 to 2002, separate thin-walled cylinder sleeves were employed to form separate cooling water paths for each cylinder. The bore bridge dimension in this configuration was 11 mm. In 2003, the thickness of the bore bridge water jacket core was reduced to 3 mm and a closed deck configuration was employed, reducing the bore bridge to 9 mm.

From 2004, the direction of cooling water flow was modified from a cross flow for each cylinder to a longitudinal flow. At the same time the bore bridge water jacket core was abandoned and a Siamese cylinder block configuration was employed. With respect to the bore bridge cooling water path, $\phi 3$ machined holes were employed to guide cooling water from the heads to the bore bridge, enabling to reduce the bore bridge dimension to 6 mm.

Figure 5 shows the reduction of the cam gear train width in 2000. Compared to the 2000 Mugen engine, which employed standard reduction gears, the diameter of the crank gears was reduced in the 2000 Honda engine to use a gear train layout without reduction gears (shown at the bottom of Fig. 5). In combination with the use of a thin front cover, this contributed to reduce total engine length.

The method of mounting the clutch would be also an important factor, if the total length of the engine including the clutch fitted at the posterior end of the engine was considered. In Honda's 2004 engine, the clutch basket was inserted in the rear end of the crankshaft, and was fastened with a single bolt. This helped to put the clutch position forward by 9.5 mm, compared to the previous year's engine.

2.1.2. Reduction of engine weight

Between 2000 and 2005, before the advent of the minimum weight regulation, a diverse range and large number of technologies, regardless of the magnitude of their effects, were undertaken to achieve weight savings. An engine weight of 88.6 kg in a 3.0 L V10 configuration had been achieved by the 2005 final engine specifications. Many of the effective technologies in reducing engine weight were closely connected to manufacturing technologies. Some examples are presented below.

Figure 6 shows the closed deck cylinder block employed from the latter half of 2002. The key to the development of this technology was a quality enhancement technology for casting that employed high-strength shell cores, which was essential to reducing the thickness of the bore bridge water jacket shell core to 3 mm, and a surface treatment technology by means of which a Nikasil surface treatment was directly applied to the inner surfaces of the cylinders. The use of these technologies resulted in weight savings due to the reduction of the total length of the engine, in addition to reductions in friction and blow-by gas with the achievement of increased stiffness around the bore.

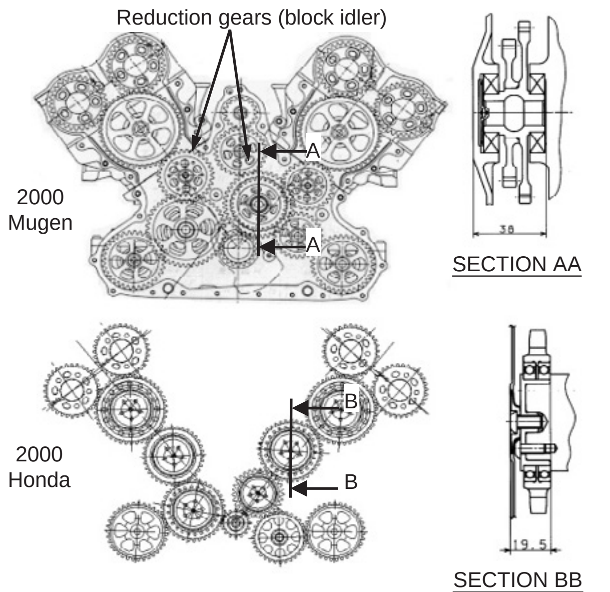


Fig. 5 Gear train configuration

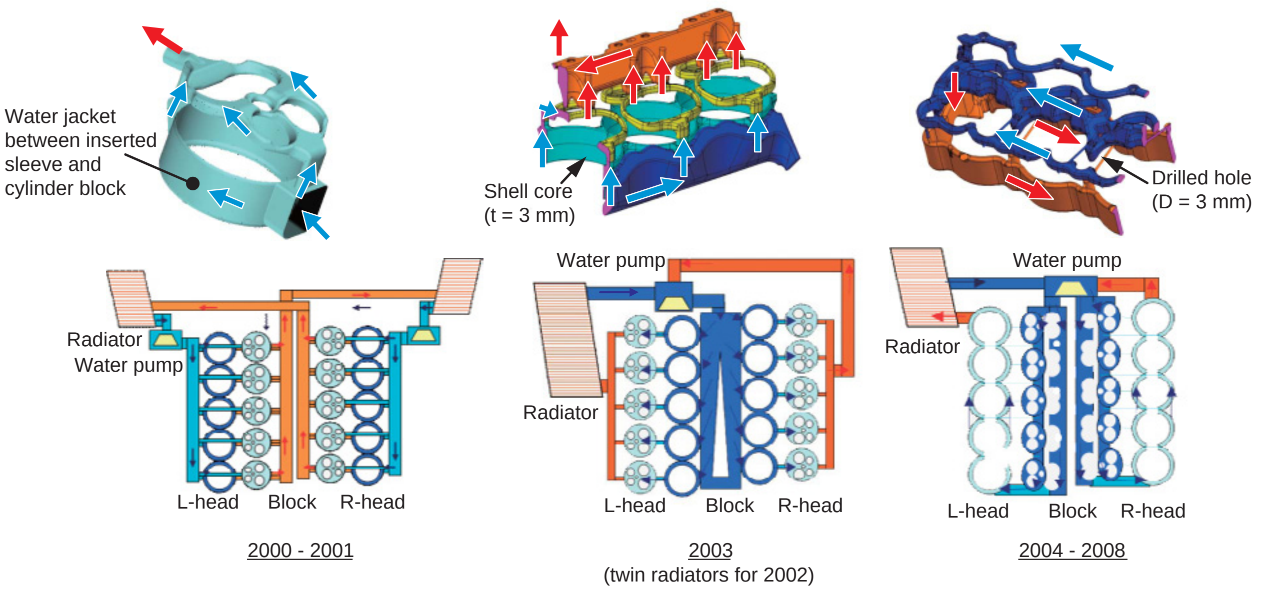


Fig. 4 Schematics of water cooling systems

Figure 7 shows a cross-section of the cylinder heads with an integrated pneumatic valve return system (PVRS) used from 2003, and a sand mold used in their casting. Until 2002, the cylinder heads were divided into cam cases and heads in order to create PVRS air channels. In 2003, a casting technology using complex head shell cores was developed, helping to enable the PVRS to be integrated with the heads. This resulted in the simple cylinder head structure and a weight saving of 6.2 kg. (This figure should be considered as a reference value because the other factors also contributed to this weight saving.)

Laser-clad intake and exhaust valve seats were developed to achieve longer mileage durability (as demanded by the regulations) and higher intake potential, as well as weight saving. Laser-clad valve seat has higher heat resistance and wear resistance than the previous press fit seats. The new valve seats were adopted from 2004, and contributed to a weight saving of 0.4 kg per engine, in addition to an increase in the valve diameter (in the case of the intake valves, from ϕ40.6 to ϕ41.6, and in the case of the exhaust valves, from ϕ31.8 to ϕ32.6), a greater degree of freedom in valve layout, and increased cooling of the combustion chamber from the valve seats. A manufacturing technology enabling a stable, high-quality cladding is an important element in valve seat cladding. In the present case, detailed settings by the Material and Prototype Departments helped to enable the creation of such a technology.

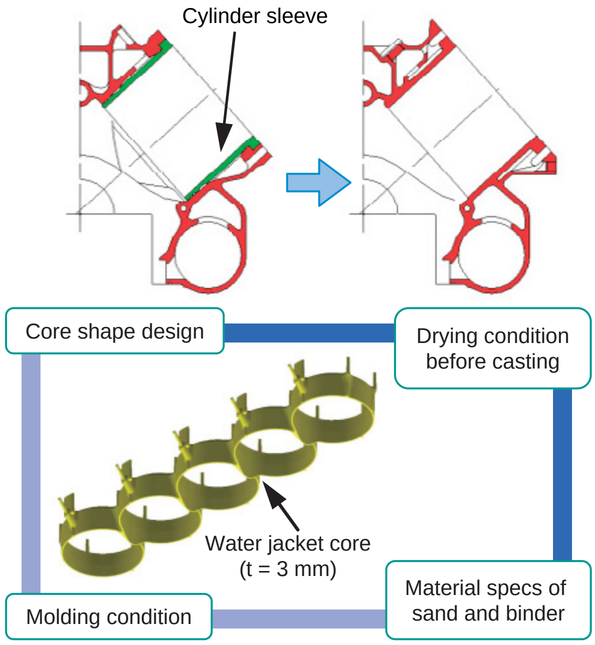


Fig. 6 Closed deck cylinder block and water jacket core

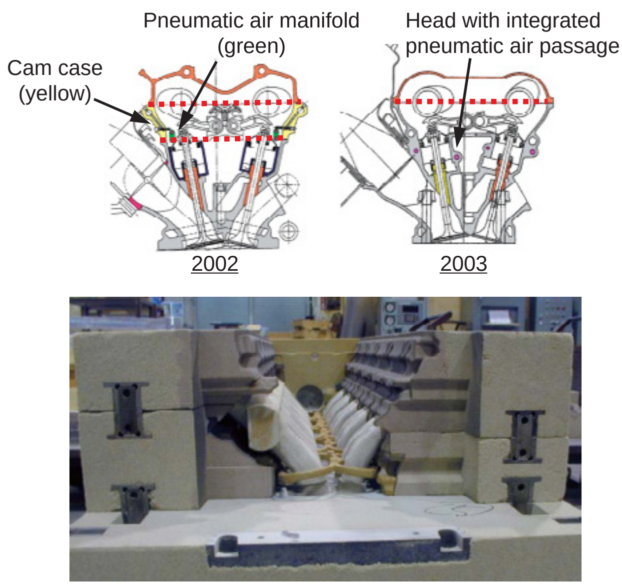


Fig. 7 Cylinder head with integrated PVRS

2.1.3. Reduction of COG

The V bank angle and the height of the crankshaft have a significant effect on the COG. Aiming at equal interval combustion, a V bank angle of 72° was used in the 2000 Mugen Honda engine. In the case of the 2000 Honda engine, the emphasis was on reducing the COG in order to contribute to total vehicle performance, and a V bank angle of 80° was employed. In 2002, an angle of 94° was used in order to increase power by reducing inlet pressure wave interference, in addition to further reducing COG height. From 2003, facing a greater demand to increase engine speed in order to increase power at the same time as reducing the COG, the demands were balanced through the use of an angle of 90°, which theoretically enables secondary vibration to be cancelled.

In the case of the height of the crankshaft, Table 1 shows that it has steadily been lowered with each year. The baseline for the COG height (0 mm) is the mating faces of the skid plate fitted to the bottom of the vehicle and the bottom of the engine, as specified in the Formula One regulations.

The achievement of a compact conrod locus (the envelope surface of conrod oscillation) is an important technology in lowering the crankshaft while preventing an increase in friction. The gap between the rotating parts and the inner wall of the crankcase has a significant effect on oil agitation resistance in the crankcase. 2005 test results showed that a rapid increase in friction occurred at a gap of 3 mm or less (Fig. 8).

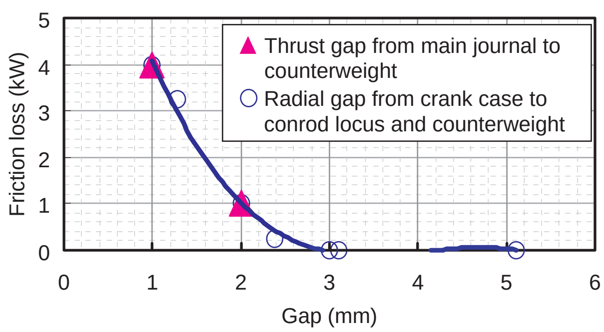


Fig. 8 Friction loss against closest gap

The point at which the gap between the rotating parts and the inner wall of the sump case becomes the narrowest is the lowest position of the conrod locus, determined by the locus of the conrod bolt head. CAE was therefore employed to study the conrod big end, helping to make the big end compact gradually (Fig. 9, Fig. 10).

With regard to the reduction of the radius of the crankshaft counterweight, it was necessary to balance the reduction of the radius and the achievement of reduced weight while maintaining a crankshaft balance rate set on the basis of friction and reliability considerations. Up to the Italian Grand Prix (Round 15) in the latter half of 2004, a counterweight manufactured from a tungsten alloy with a specific gravity of 18 was press-fitted to the crankshaft, following which it was covered with a plug and welded in place (Fig. 11). However, in order to further reduce the radius, from Round 16, the Chinese Grand Prix, a configuration was employed in which a separate tungsten alloy counterweight was directly bolted to the crankshaft. This helped to enable the radius of the counterweight to be reduced to 58.55 mm. In 2005, the radius of the counterweight was reduced to 50 mm, and the crankshaft was lowered by 4.5 mm against the previous year's engine.

2.2. Engine Stiffness

When the dynamic performance as a whole vehicle is taken into consideration, increasing lateral bending stiffness is known to have an effect on the stability of the vehicle. Figure 12 shows an example of the deformation of monocoque, engine, and gearbox under a bending load calculated by the Chassis Development Division. As indicated above, the engine was directly connected to the monocoque and the gearbox, and was thus an element in the stiffness of the vehicle. Increasing the lateral stiffness of the engine would therefore increase the stiffness of the vehicle as a whole, thereby contributing to greater stability on the track.

Figure 13 shows the example of the methods of increasing engine stiffness besides the use of the closed deck cylinder block discussed in Section 2.1. Auxiliary

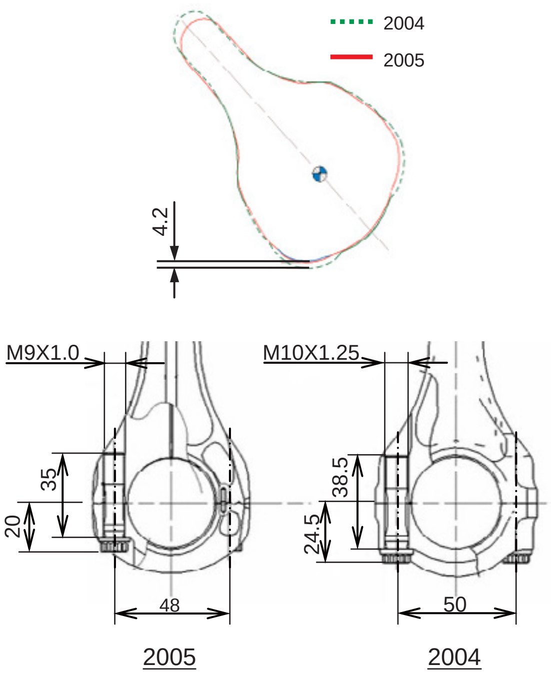


Fig. 9 Reduction of conrod locus

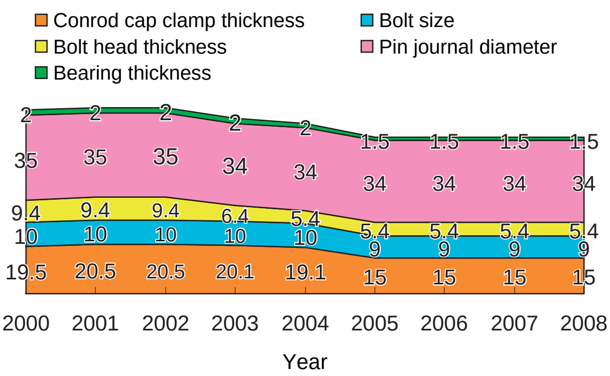


Fig. 10 Progress of conrod configuration

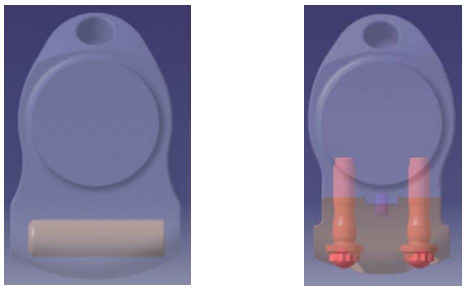


Fig. 11 Schematics of crankshaft counterweights

Fig. 12 Lateral bend calculation

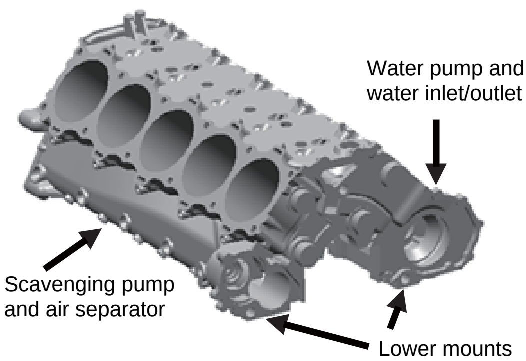


Fig. 13 Cylinder block with integrated oil/water pump

equipments including scavenging pumps and water pump were integrated in the 2002 cylinder block. The scavenging pumps and air separators were positioned at the side of the crankcase, which extended for almost the total length of the engine, to enable oil to be collected from each crankcase throw chamber and blow-by gas to be centrifugally separated at the front of the engine. The water pumps and water inlets and outlets were positioned at the front of the engine due to the demands of the radiator layout. In order to use these auxiliary devices as elements in engine stiffness, from 2002, the scavenging pumps and water pumps were integrated with the cylinder block or sump case during casting. In addition to increasing stiffness around the front lower mounts, this also increased the cross-sectional secondary moment throughout the engine, resulting in greater stiffness in the engine as a whole.

As indicated above, in 2006, major changes were made in the regulations that have effects on engine frameworks, with stipulations demanding the use of V8 engines and setting figures for minimum weight and minimum center of gravity. An example of a technology to increase engine stiffness that made optimal use of these regulation changes will be discussed below.

Figure 14 shows features of the 2006 engine related to stiffness. There should be some weight margin for changes to meet the 2006 minimum weight of 95 kg as the weight of the 2005 engine had already been reduced to 88.6 kg. In addition, the center of gravity of the 2005 engine was lower, at 163.5 mm, than the 2006 regulation figure of 165 mm, meaning that the mass had to be added to the vehicle at a higher position than the center of gravity. Based on these conditions, 2006 engine layout concept was to employ the extra weight margin partly in increasing the joint stiffness of the upper mounts of the engine and the monocoque. At the same time, the span of the joints was increased in order to reduce the degree of local deformation by reducing the input force around the mount joints.

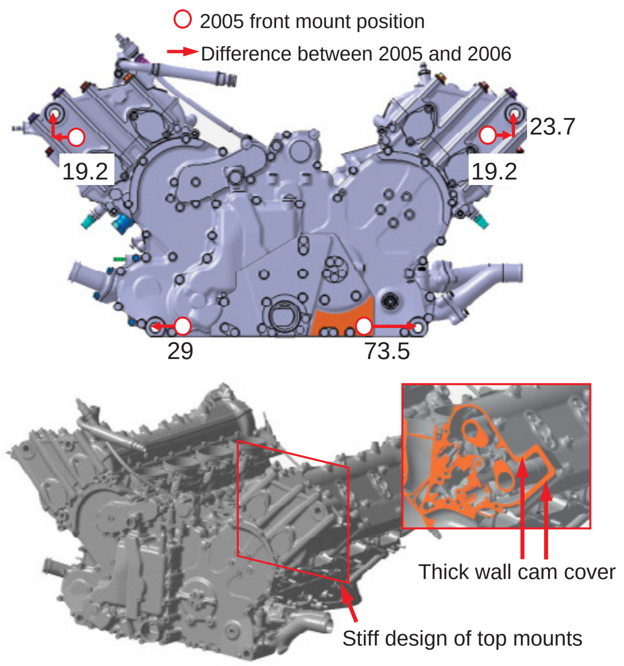


Fig. 14 2006 engine characteristics for increased stiffness

The front upper mounts were extended to the limit of the dimensions specified in the regulations; the form of the cam cover close to the front upper mounts was modified to increase the section modulus, and the thickness of the vertical wall of the cam cover was increased to form stiffening ribs. The gear train layout was modified to help enable the positioning of the lower mounts as far as possible to the left and right, and the shape of the lower mounts was designed to increase the stiffness of the sump case near the mounts.

Table 2 Relative rigidity of 2006 engine (normalized for 2005 rigidity = 1.00)

	Lateral	Vertical	Torsional
Relative engine rigidity (per unit length)	1.21	1.10	1.34
Total vehicle rigidity -Engine & gearbox = '06 -Monocoque = '05	1.50	1.23	1.16

Table 2 shows the stiffness of the 2006 Formula One engine and the 2006 vehicle as a whole based on simulation results. The stiffness values are normalized so that 2005 engine stiffness should be 1.00. The engine stiffness is expressed per unit length because the change from a V10 engine to a V8 engine stipulated in the regulations reduced the total engine length from 581.5 mm to 490 mm, and direct comparisons therefore cannot be made. As for the comparisons of total vehicle stiffness calculation, the 2005 monocoque was modified to fit the 2006 engine and gearbox. Figures for lateral, vertical, and torsional stiffness are all higher for the 2006 model. The achievement of increased lateral stiffness was one of the aims of the development, and this parameter was increased by 50% in terms of total vehicle stiffness.

2.3. Development of Oil Pump

Because Formula One engines use dry-sump lubrication systems, two types of oil pump are employed: A feed pump to supply oil and a scavenging pump ("SCAV pump" below) to collect oil. Torocoid pump rotors with four inner teeth and five outer teeth were employed for both the feed and the SCAV pumps for their excellent volumetric efficiency. The sections below will discuss the functions of the pumps and details of their development.

2.3.1. Development of Feed Pump

The necessary oil volume and oil pressure for each part of the engine varies with engine speed. The sliding parts of the valvetrain need a high-volume oil supply from low engine speeds, but the volume of oil demanded does not increase beyond a specific level as engine speed increases. By contrast, the crank pins, which use a center oil supply that is affected by centrifugal force, need higher volumes of oil at higher pressures as engine speed increases. Figure 15 shows the estimated oil supply to each part of the engine for the final engine specifications.

The set oil supply pressure in Honda's initial Formula One engine specifications was 700 kPa, but this figure increased to 900 kPa in response to annual increases in engine speed. There were two main reasons for this. The first of these was to increase the volume of oil that might contact to the piston ceiling close to top dead center by increasing the injection speed of the piston cooling oil jet. The second was that the supply of oil to the crank pins using the center supply method against centrifugal force necessitated higher oil pressures in order to achieve a stable oil supply at high engine speeds. However, it is known that if the oil supply pressure is too high, the oil pressure pulse will increase, and there will be an effect from negative pressure waves.

The following two points were focused on in order to increase the efficiency of the feed pump:

(1) In order to prevent loss from pressure relief, the minimum pump capacity should be set at a figure that enabled the necessary oil pressure at low engine speeds to be maintained.

(2) At high engine speeds, a reduction in oil volume and oil pressure due to pump suction cavitation, and a consequent loss of torque, should be prevented.

In the case of (1), the appropriate figure could be estimated to a certain extent based on experience and theory, and the aim could be relatively easily achieved by modifying the specifications. In the case of (2), however, not only was the parameter challenging to predict, but the achievement of the aim would necessitate major changes that were contrary to the concept of a race engine, i.e., reducing the speed and increasing the size of the rotor. The 2005 feed pump rotor, of 54 mm in external diameter, was operated at a high speed (12800 rpm), resulting in a severe decline in oil pressure. The following measures were employed against the two main factors in feed pump intake issues in the high-speed range:

(1) An insufficient volume of oil was flying into the rotor suction chamber during rotational transfer at high speeds: A thick rotor presented a disadvantage in this case, and it was therefore replaced with two thin rotors.

(2) The oil drawn into the rotor suction chamber leaked out of the chamber due to the effect of centrifugal force: In this case, a modified suction port shape was adopted in which the opening to the exterior closed as the chamber filled with oil (Fig. 16).

These measures increased the flow rate by 30% with no changes to the basic specifications (Fig. 17).

The main issue of concern in the development of the feed pump was potential fractures of the driveshaft due to torsional resonance. The ϕ12 feed pump driveshaft in the original pump specifications was manufactured from special carburized steel, and possessed a safety factor approximately 100 greater than that of the average drive torque. Nevertheless, fractures occurred frequently. A simulation of shaft behavior showed that tertiary torsional resonance at 17000 rpm and above was the main factor in these fractures. The increase in resonance was reduced by reducing the diameter of one section of the driveshaft to ϕ8 in order to reduce the shaft spring. Fractures did occur following this when the rotor drive method or the rotor itself was modified, but the knowledge gained from the analysis discussed here enabled the situations to be rapidly responded to by altering the torsional stiffness of the driveshaft or the rotor inertia.

2.3.2. Development of SCAV pump

The SCAV pump, by rapidly collecting oil that has been used in lubricating or cooling the engine, enables a sufficient quantity of oil to be maintained in the oil tank and helps to ensure sufficient oil pressure when the vehicle is cornering, in addition to helping to prevent an increase in friction due to oil agitation by the moving parts in the engine. This is particularly important inside the crankcase, where oil agitation tends to generate increased friction. Each of the crankcase throw chambers

Fig. 16 Comparison of suction port shape (shown as pink area)

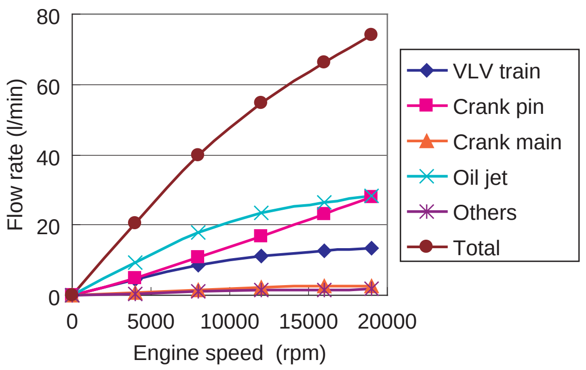


Fig. 15 Oil distribution

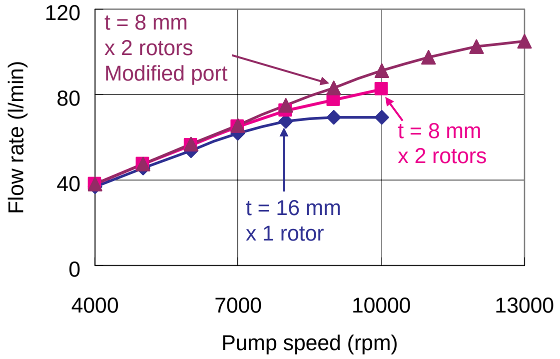


Fig. 17 Effect of twin rotor and modified port shape

is a sealed structure closed off from the others. This configuration helps to prevent pumping loss due to the volumetric transfer of blow-by gas by the alternate actions of the pistons; the gas exhausted by the falling pistons in the V bank would be sucked up by the rising pistons. For the same reason, it is desirable to prevent any contact between the throw chambers in the SCAV pump that collects the oil. The initial SCAV pump design used one SCAV pump for multiple throw chambers, but later, independent SCAV pumps were used for each throw chamber in order to increase energetic efficiency. In addition, to help ensure stable collection of oil in the case in which the oil was unevenly distributed to the front or rear, pumps were placed at the front and rear of each throw chamber.

Figure 18 shows the oil system based on the final specifications.

The main function of the SCAV pumps is the collection of oil, but blow-by gas also plays an important role in this. Just as a vacuum cleaner would not be able to suck up dust in a vacuum, the SCAV pumps would be unable to collect oil in the absence of blow-by gas. For this reason, engine breathing supplies an optimum volume of blow-by gas from the upper section of the oil tank to the heads and the gear housings, where blow-by gas is not normally present. However, oil mixed with large quantities of blow-by gas can have a variety of negative effects on the feed pump, including interfering with filling and producing bearing damage. For this reason, after the oil is collected by the SCAV pumps, the oil and the blow-by gas are separated in an oil-air separator that uses centrifugal separation, and are then sent to the oil tank by means of separate channels.

When a person drinks water through a straw, the water could not be drawn up if the end of the straw was not completely immersed in the water. In the same way, the SCAV pump inlets must be completely immersed in oil in order for the pumps to collect oil. In the initial SCAV pump design, oil struck by the crankshaft and the conrod flew into the SCAV pump under the force of its own inertia (Fig. 19). This design was not optimal for stable collection of oil, because the oil did not collect in the SCAV pump inlet when cornering G force was produced in the opposite direction. The use of an oil trap in the SCAV pump inlets from 2004 increased the efficiency of oil collection (Fig. 20).

An internal compression SCAV pump was introduced in 2003 in order to increase pump efficiency. Formula One SCAV pumps collect a mixture of oil and blow-by gas from inside the crankcase at an absolute pressure of 20 to 40 kPa, and send the mixture through exhaust channels pressurized to levels of between 150 and 250 kPa. This represents a compression ratio of between 4 and 12. Used at such high compression ratios, a compression pump is more efficient than a conventional displacement pump (Fig. 21). Torocoid pumps are able to be used for internal compression, and a torocoid pump could be modified for the purpose simply by changing the port shape for delayed opening. Pump damage when oil alone was absorbed would become an issue, and this was responded to by positioning a relief valve on the rotor side. A compression ratio of 2 was employed to help ensure performance and reliability. This measure reduced SCAV pump drive resistance by 30% and engine friction by 3 kW.

2.3.3. Reduction of weight of pump rotors

The rotors were initially formed from a sintered aluminum powder material, but later a magnesium alloy was employed for the inner rotor and a plastic material for the outer rotor(1) in order to achieve weight savings.

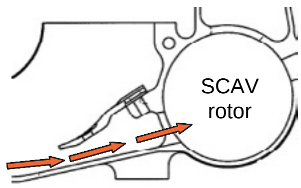


Fig. 19 Straight suction

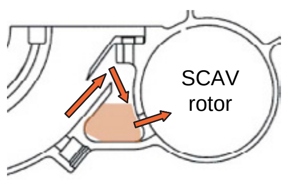


Fig. 20 Trap-type suction

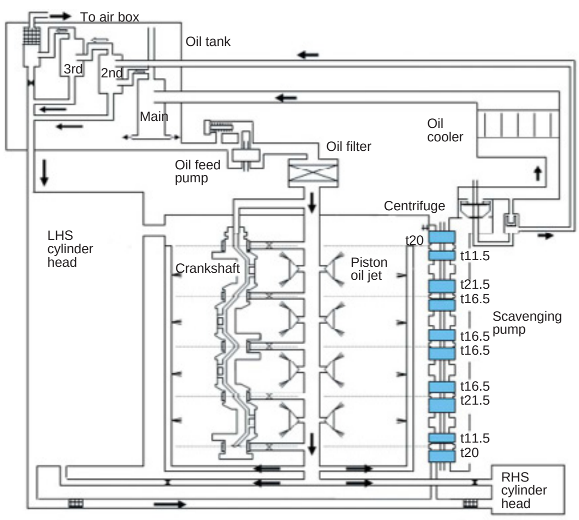


Fig. 18 Oil system

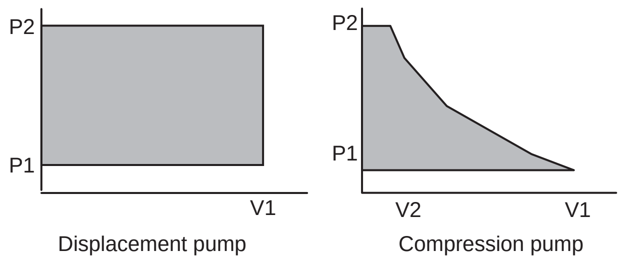


Fig. 21 Comparison of pump work

This reduced the weight of the rotors by half. To help prevent friction due to the use of soft materials, a unique, low contact face pressure tooth shape was employed. In another year, a sintered steel powder material was employed for the feed pump rotor to help prevent torsional resonance, an issue which was discussed above.

2.4. Cooling System

Cooling system development in Honda's third Formula One era commenced with the aim of reducing the size of the radiator. The size of the radiator has a significant effect on the vehicle's aerodynamic characteristics, i.e., is directly related to lap times. Reducing heat rejection and enhancing heat transfer coefficient are essential factors in reducing the size of the radiator, and the development was focused on them.

The water jacket was made as small as possible, in order to help reduce heat rejection by enabling only the areas in which cooling was necessary to be cooled and to help prevent unnecessary heat transfer from the exhaust port. Figure 22 shows the design concept of the water jacket. In addition, individual cylinder type water jackets (Fig. 4) were applied to the cylinder heads in order to reduce the difference in temperature between cylinders. These measures reduced heat rejection by 25% against a conventional engine, and reduced the size of the radiator by 25%.

Increasing the temperature of the coolant, i.e., operating the engine at a high water temperature is an effective method of enhancing heat transfer coefficient. The entire water system was pressurized to 365 kPa in order to help prevent boiling, and the flow rate was increased to help ensure sufficient reduction of the temperature of the walls of the combustion chamber despite the increase in water temperature. In addition, the ability to employ high-temperature water helped to enable a greater degree of closure in the cooling option (the degree of opening of the radiator exhaust port), which is more advantageous from the perspective of aerodynamic performance. While conventional engines operated at a water temperature of 90°C, during Honda's third Formula One era, a temperature of 120°C was achieved in 2000, and the maximum temperature of 130°C was achieved in 2008. At the same time, the power loss in accordance with high water temperature operation was restrained at the minimum.

Figure 23 shows the influence of water temperature on engine power.

Until 2002, the radiator and the oil cooler were each divided and positioned on the right and left of the chassis. In order to simplify the system, including the pipe layout, from 2003, the radiator was installed on the left side and the oil cooler on the right side of the chassis. In this configuration, the cooling water on the right side of the engine bank was recirculated into the water pump suction without passing the radiator, and mixed with the water returned from the radiator on the left side of the chassis. The configuration did not affect performance and reliability.

The introduction of a Siamese block from 2004 in order to reduce the total length of the engine had a significant effect in changing the flow of cooling water. The water sent from the water pump was allocated to each bank of cylinder heads, and flowed from the front to the rear, following which it flowed from the rear of block to the front. This increased the flow rate in the water jackets, thus increasing the rate of heat transfer. This configuration also reduced the volume of cooling water inside the engine from 3.2 L (2003) to 2.1 L (2004), contributing to the achievement of weight savings (Fig. 4). Figure 24 shows temperature measurements taken inside the engine for each water jacket configuration. The results show that increasing the flow rate reduced the temperature of the combustion chamber walls and the area close to the surface of the deck. A water channel was also machined between the

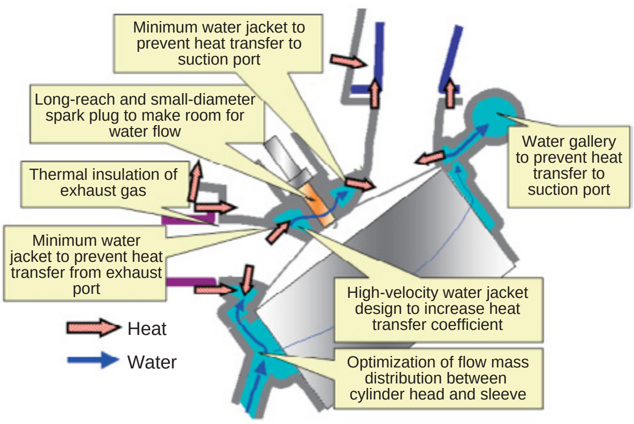


Fig. 22 Design concept of water jacket

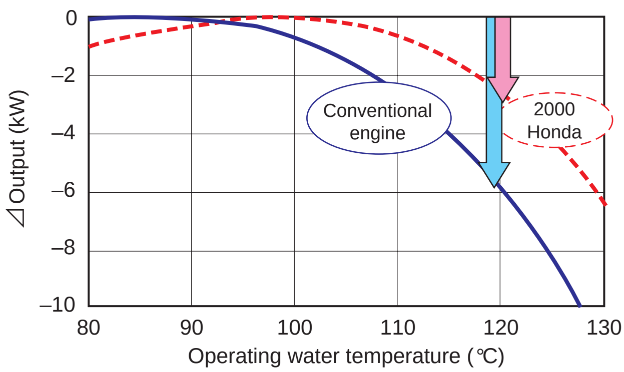


Fig. 23 Effect of water temperature on engine performance

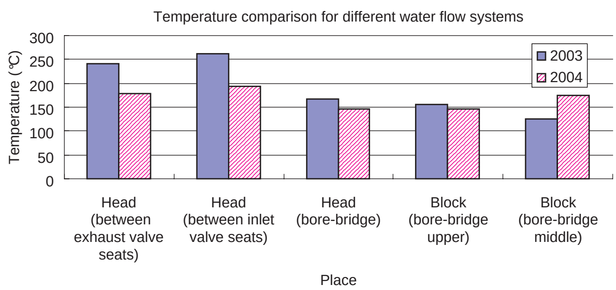


Fig. 24 Temperature comparison between 2003 and 2004

block bore bridges to help prevent an increase in temperature between the bridges.

Cooling system development from 2008 aimed to enhance the aerodynamic performance of the vehicle, and therefore sought to reduce the size of the radiator. To help enable this, it was necessary to increase the head of the water pump and the water flow rate.

3. Conclusion

A compact, lightweight, low center of gravity, high-stiffness and high-power Formula One engine framework with an excellent vehicle fit was progressively developed to reduce lap times on the circuit by enhancing total vehicle performance. The following results were obtained:

(1) The reduction of engine size and weight progressed through a fusion of material technologies, manufacturing technologies, and framework design, including the design of the cooling and lubrication systems. As a result, in 2005 (the final year in which 3L V10 engines were used in Formula One) Honda reduced the weight of its engine to 88.6 kg, producing one of the lightest and the most compact engine among the Formula One teams.

(2) Reciprocating system parts and a high-efficiency scavenging pump were developed in order to lower the center of gravity, helping to enable the height of the crankshaft to be reduced to 58.5 mm with a minimum increase in friction.

(3) Making maximal use of the stipulations of significantly revised engine regulations in 2006, the stiffness of the engine was increased for the sake of the dynamic performance of the vehicle. 2006 engine contributed to an increase of 50% in the vehicle's lateral bending stiffness against that of the previous year's.

(4) Feed pump technologies were developed to accept high speed operation and hence enable more precise response to oil demands in each part of the engine, a lightweight and compact design, and a reduction in drive loss.

(5) The use of independent SCAV pumps provided with oil traps at their suction ports increased oil collection efficiency. The use of an internal compression technology reduced drive loss by 30%, and the use of substitute materials in the rotors halved their weight.

(6) The optimization of the water jacket configuration reduced heat rejection and helped to ensure reliability with operation at high water temperatures while controlling performance loss, thus contributing to enhancing the aerodynamic performance of the vehicle.

■ Author ■

Kiyoshi KONDO

Hajime ENDO

Tetsuo GOTOU

Development of Reciprocating Parts and Crankshaft in Honda's Third Formula One Era

Tetsuo GOTOU* Nagao YANAGISAWA*

ABSTRACT

Reciprocating parts and crankshafts for race use must display reliability and achieve low friction levels. Increasing engine speed is an effective method of increasing the power of natural aspiration engines, and Honda increased engine speed to 19600 rpm in 2006, prior to the introduction of upper limits for engine speed. In 2005, Formula One regulations were changed to increase the distance for which engines would be used from one race event (400 km) to continuous use for two race events (1500 km). In order to balance performance under severe use conditions with the achievement of reduced friction, the shapes, materials, and manufacturing methods used for the reciprocating parts and crankshaft were modified to reduce weight and increase strength and stiffness. In FY2008, the weight of the reciprocating parts was reduced to 358 g, representing a weight saving of 41 g against the FY2000 figure of 399 g. This paper will discuss the development process, focusing on each component part.

1. Introduction

Like many other engine parts, the reciprocating parts used in Formula One engines must reconcile multiple conflicting performance demands. It is advantageous for reciprocating parts to be heavy and robust, in terms of resistance to high combustion pressures for long periods, but the achievement of increased engine speed necessitates lightweight parts.

In addition, reducing the weight and the contact area of the reciprocating parts represents an advantage in terms of the reduction of sliding friction, but going too far in this direction can increase friction and result in wear.

As in the case of previous engine development programs, the development program for reciprocating parts in Honda's third Formula One era developed a large number of new technologies that were later employed in races.

The introduction of CAE resulted in significant changes in the approach to technological development. In particular, from 2003 it became established procedure for designers themselves to construct 3D models and conduct CAE analyses, and this helped to enable the optimal design of structural strength, resonance, flow rates, temperature distribution, heat stress and other parameters within short time periods.

The analysis of relevant phenomena using dyno tests and measurement technologies was also effective in increasing the accuracy of CAE.

Performance demands were fulfilled and durability and reliability and reduced friction were realized through the development of new manufacturing methods, new materials, and surface treatment technologies such as diamond-like carbon (DLC) coatings.

This paper will discuss the history of the development process with a focus on individual parts.

2. Changes in Specifications of Reciprocating Parts and Crankshaft

Table 1 shows annual changes in the use conditions, specifications, and materials of the main reciprocating parts, and Fig. 1 shows changes in the weight of the parts.

The weight of the reciprocating parts increases in some years. This is due, in 2002, to the change in the cylinder bore diameter, in 2005 to responses to regulations stipulating increased use distances, and in 2006 to the introduction of regulations prohibiting the use of aluminum metal matrix composites (MMC) for the pistons.

In 2004, despite the extension of use distances by the regulations, the use of CAE analysis in MMC piston design helped to enable the achievement of weight savings in the pistons. The upshift engine speed was basically increased each year, but regulations limited engine speed to 19000 rpm in 2007.

* Automobile R&D Center

Table 1 Reciprocating parts and crankshaft specification history

Year	'2000	'2001	'2002	'2003	'2004	'2005	'2006	'2007 & 08
Miledge (km)	400			420	800	1500		1350
Engine revolution (rpm)	17500	17800	18600	18800	19200	19200	19600	19000
BorexStroke-Type	ϕ95x42.24-V10		ϕ 97x40.52-V10				ϕ97x40.52-V8	
Piston	Radialy rib	# rib	Bridge #rib					
Piston ring	Rectanglar		Mono "R"	"WR"			Titanium "WR"	
Piston pin	ϕ18.7-MAS1C			ϕ17-MAS1C		ϕ17TiAl	ϕ17-HD200	
Connecting rod	I-section			Hollow			I-section	
Crankshaft (Pin/Main)	(ϕ35/52) NT100		(ϕ36/48) NT100	(ϕ34/46) GKHYW	(ϕ33/45) GKHYW	(ϕ34/46) GKHYW	(ϕ34/45) GKHYW	

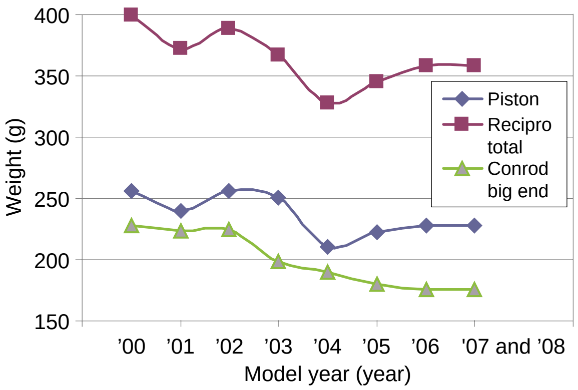


Fig. 1 Reciprocating parts weight history

3. Piston Development

The pistons are directly affected by the energy of combustion gases, and must withstand maximum combustion pressures of 7 t and inertial forces exceeding 2 t while being exposed to temperatures in excess of 2000°C.

The general orientation of the development program was to reconcile high strength with low weight. CAE was introduced at an early stage to assist in the realization of optimal designs. In the initial stages of development, Honda CAE specialists were requested to perform simulations, but by degrees designers themselves became able to construct 3D models, and made judgments regarding strength and stiffness from the results of CAE analyses incorporating combustion pressure and inertial forces. CAE also helped to enable designers to study temperature distribution, thermal deformation, and thermal stress, and the ability to predict lifespans and allowable operating conditions for parts from the results of such simulations increased the accuracy with which the shapes of parts were evaluated (Fig. 2).

The use of these analytic techniques led to a variety of changes being made in the shapes of pistons. The initial external shape of the pistons featured a skirt extending around the circumference, and ribs radiating for equal lengths in eight directions. To reduce the weight of the pistons, their shape was modified to use curb ribs (# ribs). This helped to enable the realization of a weight saving of 15 g. However, both these piston shapes resulted in the conrod small ends being covered by the piston walls, increasing the challenge of providing an oil supply around the conrod small ends and the piston pins. To resolve this issue, and to achieve further weight savings, the shape of the pistons was further modified to use curb bridge ribs (# bridge ribs) with no wall on the piston ceiling side. Changes continued to be made, but this basic piston shape was adopted until 2008 (Fig. 3).

The weight of the pistons, which stood at 255 g in 2000, had been reduced to 210 g by 2004. The use of MMC, which increased high temperature strength by 30%, contributed to this reduction in weight.

The reduction of the weight of the pistons did not only increase their toughness against inertial forces, but also reduced friction. However, it was not sufficient for the pistons to merely be lightweight. It was also essential for the sliding of the piston skirts to be smooth and stable.

The sliding resistance of the piston skirts, which slide at high speeds on an oil film, is determined to a greater

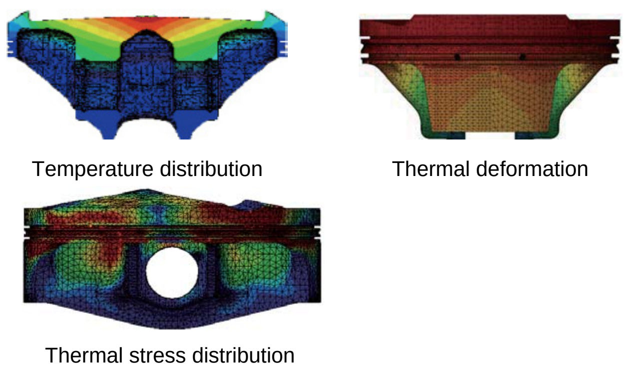


Fig. 2 Examples of piston CAE results

Fig. 3 Piston structure history

extent by the shape of the skirt surface than the specific friction coefficient of the material forming the skirt surface. Rough and deep streaks on the piston skirt surface increase oil retention, but sliding resistance also increases. Sliding resistance is low when shallow striations are employed, but they encourage scuffing of the skirts due to the breakage of oil film. Scuffing was a concern when designers halved the depth of the streaks in order to reduce friction, and the issue was responded to through the use of steel plating and a plastic coating containing metallic pigments, and by modifying the skirt forms.

The initial method of designing the piston skirt form involved a burner method, in which the piston ceiling is exposed to the flame of a burner to measure thermal deformation. A basic form would be established on the basis of the measurements and damage to the skirts following durability tests would then be studied to make further minute adjustments. The increased accuracy of CAE design technology, however, helped to enable the use of a surface pressure method, by means of which the form of the skirts was designed on the basis of CAE analyses of thermal distortion and skirt stiffness values. This method helped to enable the realization of skirt marks largely corresponding to development targets through the use of simulations alone.

The detachment of the pin clips was a piston issue that remained unresolved until the end of Honda's third Formula One era. As engine speed increased annually, engine blow occurred frequently as a result of the detachment of the pin clips. The clips would fracture and detach from their grooves, and the issue was therefore responded to by increasing the strength of the clips through shot peening and nitriding, but the issue reoccurred as engine speed increased.

It was hypothesized that the detachment of the clips resulted from the collapse of the clip grooves. As countermeasures, the diameter of the clips was increased in order to reduce surface pressure and increase strength, and an oil jet was introduced to cool the clips and provide lubrication. It was thought that the issue of detachment of the pin clips had been resolved with the introduction of these measures, but the issue reoccurred with an increase in combustion pressure.

Due to the regulations stipulating periods in which design changes were prohibited, further countermeasures could not be effected. In endurance tests on a test bed conducted in 2008, one-third of the tested engines stopped as a result of detachment of the pin clips. However, this only occurred after a distance of 1000 km had been exceeded, and therefore did not represent an issue during actual driving on the track, including race operation.

As test verification proceeded, it was theorized that the collapse of the clip grooves that was resulting in detachment of the clips was not the result of buckling due to the load from the pins, but of wear resulting from repeated minute deformations of the clip groove due to combustion pressure and inertial forces. Because it was not possible to prevent the deformation of the clip grooves, countermeasures focused on increasing slidability between the clips and the clip grooves, and increasing the wear resistance of the clip grooves. However, the effectiveness of these measures was not verified.

4. Development of Oil Jet

For aluminum pistons, cooling is an important factor in realizing strength and dimensional stability in the piston material and reducing thermal stress.

In the initial stages of development, four or six holes were used per cylinder in the oil jets employed to cool the pistons (Fig. 6). The flow rate oil injected was slightly higher than 10 l/min.

As development proceeded, increasing engine speeds and power output increased the thermal load on the pistons. The flow rate of oil injected was steadily increased by means of increasing the diameter of the oil jet holes, but the use of MMC, a material with good high-temperature strength, a low linear expansion coefficient, and a high level of resistance to thermal stress, to manufacture the pistons during this period meant that fundamental modifications were not sought.

In 2006, the use of MMC was prohibited. Honda returned to the conventional 2000 aluminum material for the manufacture of Formula One pistons, but cracks occurred frequently in the aluminum pistons when attempts to increase engine speed commenced. The mechanism of the cracks occurring in the thin areas of the ribs (Fig. 4) and the centers of the bridge ribs (Fig. 5) could not be explained by CAE analyses of combustion pressure and inertial forces, but the results of thermal analysis showed that the heating of the piston ceilings and increased temperatures added a tensile thermal stress exceeding 50 MPa to the stress that had

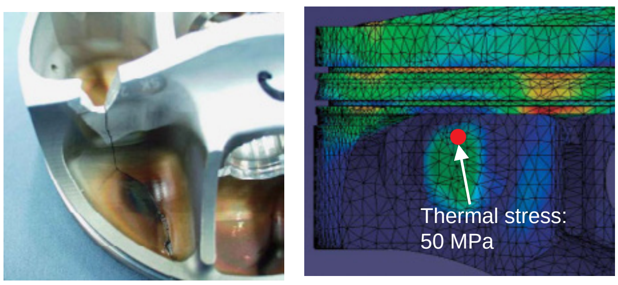


Fig. 4 Crack due to thermal stress in thin area of rib

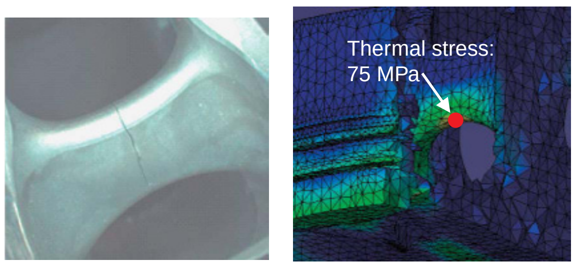


Fig. 5 Crack due to thermal stress in bridge rib

already been studied, and this resulted in the cracking.

Figures 4 and 5 show the results of thermal analyses of the thin area of a rib and the center of a bridge rib respectively.

Because the centers of the piston ceilings bore combustion pressure, strong tensile stresses were produced on them. In addition, their temperatures were increased by longer exposure to flame. This resulted in cracking due to a decline in the strength of the materials. The ignition timing for each cylinder was retarded by an average of 6.5° as a countermeasure. However, the results of endurance tests on a test bed showed that retarding the timing resulted in a power loss of more than 7 kW against optimum ignition timing, and also resulted in a decline in fuel efficiency.

This made the cooling of the piston ceilings an urgent issue. However, simply increasing the volume of oil used in the oil jets would also increase friction, and was therefore not an effective countermeasure.

The thin areas of Formula One pistons are only 2 mm or less thick, and therefore even if the volume of oil was increased in one area, the cooling effect would tend not to extend to surrounding areas. An even spread of cooling oil over a wide area was therefore desirable. The six holes used per cylinder for the oil jets were increased first to 12 holes and finally to 24 holes (Fig. 6). Because the centers of the piston ceilings, where the temperature increase was greatest, were shielded by the conrod small ends, it was challenging to directly introduce oil to this section, but this issue was resolved through the use of an oil jet in which the point of oil discharge had been raised to a level at which it almost made contact with the piston ceilings. In addition, the up-down motion of the pistons extended the distance of travel of the points at which the oil made direct contact, and this expanded the range of direct contact of the oil jets (Fig. 7).

However, it had been necessary to increase the size of the oil jet in order to raise the point of oil discharge, and the incidence of cracks due to engine vibration then became a concern. While the oil jet could be used on the inlet (In) side, the issue could not be resolved on the exhaust (Ex) side by the deadline for engine submission. The 2007 homologation engine finally included an oil jet using 19 holes per cylinder.

The use of an equivalent volume of oil per cylinder to the previous oil jet with six holes per cylinder reduced the maximum temperature of the pistons by 25 °C. The achievement of strength increases in the pistons helped to enable the previously necessary 6.5° retardation of ignition timing to be reduced to 1.5°, and power loss from the retardation no longer represented an issue.

Direct measurements were initially relied upon for adjusting the flow rate of the multi-hole oil jet, but increases in the accuracy of CFD meant that the adjustments could eventually be made using CAE alone. Because it was also possible to predict the piston temperature distribution resulting from the use of the oil jet, it became possible to design the strength of the pistons in tandem with the oil jet.

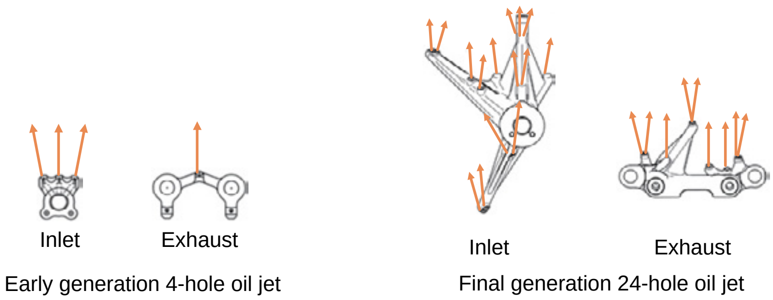


Fig. 6 Evolution of oil jets

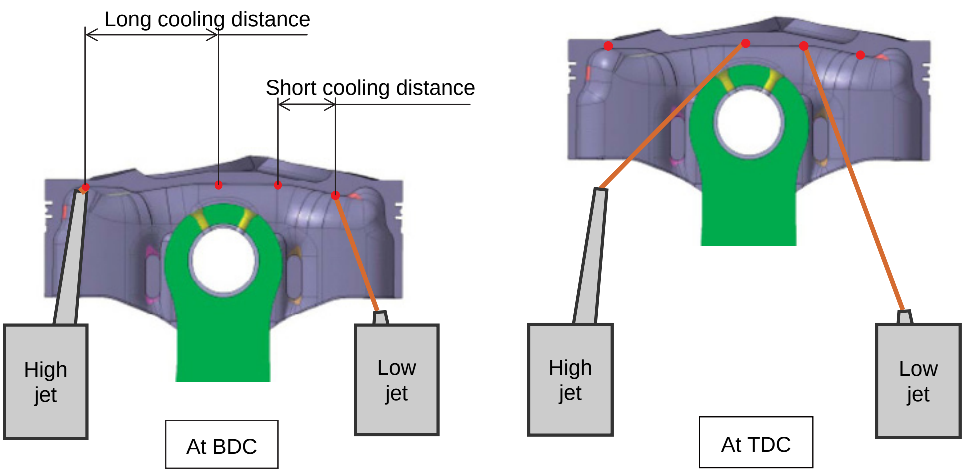


Fig. 7 Efficiency of high oil jet

5. Development of Piston Rings

In order to achieve weight savings and reduce sliding friction, Formula One engine piston rings are formed from top rings and oil rings.

The initial piston ring configuration was identical to that used in mass production vehicles. The rings were formed from a top ring with a rectangular cross-section and an oil ring with a spacer expander inserted between rails.

In 2002, expanded piston rings were developed in order to reduce friction and the decline in oil efficiency and blow-by gas leaks produced by fluttering. These were ultra-fine rings of 0.9 mm in width and 1.4 mm in thickness, with a rear expander to provide tensile force (Fig. 8).

In the initial year of development, a single ring configuration without an oil ring was tested. This configuration boosted power by more than 10 kW, but increased oil consumption to the 30 km/l range. It therefore could not be used in races, and was only used in the Grand Prix qualifying round.

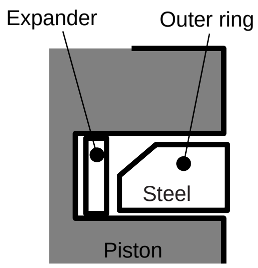


Fig. 8 Expanded piston ring

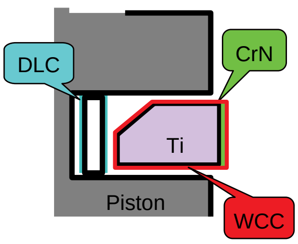


Fig. 9 Titanium expanded piston ring

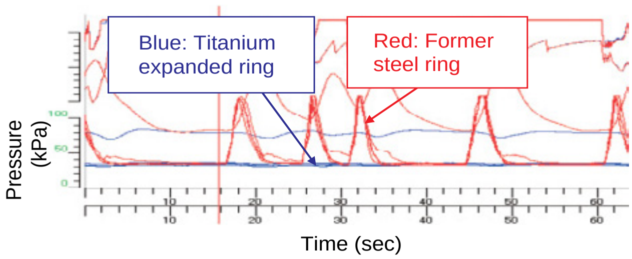


Fig. 10 Rise of crank case pressure due to piston ring fluttering

In 2003, regulations prohibited any change of engine parts between the qualifying round and the race, meaning that the ring could no longer be used in the qualifying round. From 2003, a twin-ring configuration was employed in which an identical expanded ring was incorporated in the oil ring groove. The use of the new rings increased the engine speed at which fluttering occurred by more than 1000 rpm against the previous rings, and returned oil efficiency to the 100 km/l range.

In 2006, the steel expanded rings were replaced by titanium in order to increase fluttering toughness by achieving further weight savings. Titanium displays a high level of aggressivity, making damage to the piston groove side or the expander a concern. To respond to this concern, a tungsten carbide coating (WCC) was applied to the rings and a DLC coating was applied to the expanders (Fig. 9). The effect of the reduction in weight achieved through the use of titanium was tremendous, preventing the increase in pressure in the crankcase due to fluttering not only at wide-open throttle but also when the throttle was off, and helping to enable oil consumption to reach 150 km/l. Figure 10 compares the status of pressure inside the crankcase for the steel and titanium rings.

In mass production engines, one important function of the piston rings is to enable heat to escape from the pistons to the cylinders. In Formula One engines, by contrast, the pistons are cooled by powerful oil jets. In addition, the heat transferring surface area of the expanded rings was also extremely small. The contribution of the piston rings to cooling was therefore minimal, and no issues arose as a result of using titanium, a material with a low rate of thermal conductivity.

Maintaining the benefits provided by the expanded rings necessitated strict control of the taper angle, leading to the development of a dedicated measurement jig. This jig incorporated a ring groove of the same dimensions as the pistons and an external cylinder of the same dimensions as the sleeves, and was able to accurately reproduce the state of the rings when fitted. When a ring was fitted on the jig, it was scanned by the probe of a form measurement device through two measurement windows, enabling measurements to be taken for the entire ring (Fig. 11). Only rings within a tolerance range of a mere 6' were employed in race engines.

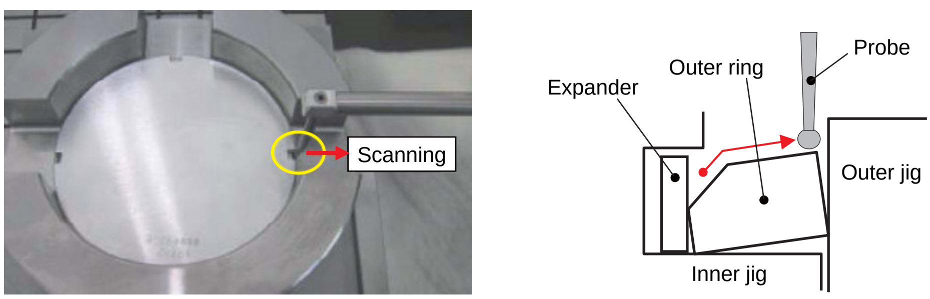


Fig. 11 Expanded ring taper angle measurement jig

6. Conrod Development

The conrods transmit the up-down motion of the pistons to the crankshaft and convert it into torque. They are therefore subjected to powerful compressive and tensile forces. Theoretically, thrust should not be generated in the conrods, but thrust is actually generated due to the deflection of the crankshaft and the inclination of the pistons. This thrust must therefore be limited by position adjustments.

Normally, the side faces of the conrod big ends contact the crankshaft to limit thrust, but if the motion is stable and the thrust load is sufficiently small, thrust can be limited by the side faces of the conrod small ends making contact with the pistons. In this latter configuration, the relative sliding speed of the thrust faces is low, and friction is therefore reduced. In the initial year of development the conrod big ends were used to limit thrust, but later the small ends were used in order to achieve the effect described above. Because this would increase wear on the sliding faces, a DLC coating was applied to the sides of the conrod small end and oil grooves were engineered in the sides of the piston bosses and a fluorine plastic coating applied. These measures helped to enable the achievement of a 3 kW increase in power.

In addition, measurements taken using the link method showed that the motion of the crankshaft generates a torsional resonance in the conrod shafts, the amplitude of which reaches 5°. The fact that torsional resonance was generated was known before these measurements from circumstantial evidence, such as the facts that if the piston recesses were not elliptical, they would strike the valves, and that the actual strength of the conrods was much lower than their unit tensile and compressive strength. For this reason, limits were placed on the reduction of the section area of the conrod shafts in the quest to achieve weight savings by the need to maintain torsional stiffness, rather than by simple tensile and compressive strength.

A normal conrod shaft has an I-section or an H-section (Fig. 12). No matter how large the vertical and horizontal dimensions of these sections, ultimately they are assemblages of three thin sheets. The torsional stiffness of these sections is determined by the cube of the thickness of the sheets, and does not reach a high figure. By contrast, the torsional stiffness of structures with hollow sections such as circular pipes and square pipes is proportional to the cube of the horizontal and vertical dimensions of the structure, and a high figure can be achieved even if the structure is formed from thin sheets.

Fig. 12 Former I-section conrod

Based on the theory that the ideal cross-section for the conrod shafts was not an I- or H-section but a hollow section, development efforts commenced to produce a hollow conrod (this was termed a “box conrod” due to the fact that the hollow section was rectangular). The structure was realized through the use of diffusion bonding (Fig. 13).

Evaluation of the form of the conrods using CAE analyses and the use of technologies to assist in hollowing the rods helped to reduce the weight of the conrods by 30 g, enhance the touch of the valve recesses in the direction of the circumference, and increase the degree of freedom in setting the compression ratio. However, regulations established to reduce costs prohibited the use of this distinguished technology from 2005, and it was therefore only used for a two-year period, from 2003 to 2004.

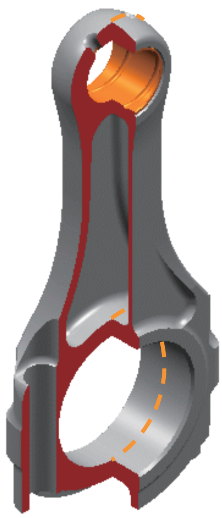


Fig. 13 Hollow conrod

7. Bearings Development

Issues involving main bearings were rare during Honda’s third Formula One era, and the bearings development program therefore chiefly focused on the development of conrod bearings. This was the case because of the severity of the use environment for the conrod bearings in Formula One engines. The diameter and width of the conrod bearings used in Formula One engines are similar to those used in light mass production engines with up to 660 cc displacement, and their dimensions are almost identical.

However, the loads on the conrod bearings during four cycles at wide-open throttle are constantly several times higher than those on the production car conrod bearings (Fig. 14). Their sliding speed is 2.6 times higher, and sliding resistance generates up to approximately 15 times as much heat.

In addition, cavitation attacks are induced by the inertial force of the reciprocating parts, which exceeds 5 t.

For these reasons, the attempt to ensure durability was a prime concern from the initial stage of development. A change in any condition, such as increased engine speed, increased frequency of use of wide-open throttle, or increased oil aeration, would frequently result in damage to bearings, leading to the seizure of the conrods. The development program was particularly challenging in years in which the regulations were altered to double the use distance demanded from Formula One engines.

The 2004 increase in the use distance of the engines from 400 to 800 km was successfully responded to by reducing the weight of the reciprocating parts, and by a variety of other initiatives including modifying the shape of the relief and adjusting the bearing clearances.

With the advent of the 2005 regulation stipulating that an engine must have a life of 1500 km, the materials of the linings and overlays were modified and their thickness adjusted, in addition to which the shape of the relief was modified, the clearances were altered, and high-viscosity oil was employed. Despite these efforts, however, conrod bearing seizure occurred frequently in endurance tests on a test bed.

Finally, the tests showed that a copper alloy with added silicon (silicon bronze) displayed a life of 1500 km when the contact pressure was increased two-fold or more. In addition, no serious damage occurred and no issues were caused by bearing seizure even when low-viscosity oil was used.

The design concept of silicon bronze differs from that of the conventional conrod bearings. The conventional bearings were bi-metal, bringing together the strength of steel and the heat dissipation and slidability of copper, and were produced by sinter bonding. However, because the strength of the surface was determined by the copper, and heat transfer to the conrod was determined by the steel, the bearings also combined the undesirable properties of both materials (Fig. 15).

While costly, silicon bronze is stronger than steel, and displays a level of heat dissipation and slidability close to those of copper. The back metal was produced from this material with the aim of achieving a dramatic increase in toughness (Fig. 16). The Young's modulus of silicon bronze is lower than that of the conventional conrod bearings, and it displays a greater degree of crushing at identical contact pressures, giving it a good ability to follow the deformation of the conrod big ends. The ability to maintain the contact pressure also represented an advantage from the perspective of heat dissipation (Table 2).

The contact pressure was initially set low in order to protect the conrod bolts, but insufficient contact with the conrods prevented the material from displaying its full heat dissipation performance. Increasing the contact pressure helped to enable the expected heat dissipation performance.

Following this, a copper alloy with added silicon and nickel (Corson alloy) with good strength, thermal

Table 2 Characteristics of bearing backmetal materials

		Si-Cu	Former metal
Thermal conductivity	(W/mK)	159	50
σ 0.2	(MPa)	570	420
Young's modulus	(GPa)	148	210

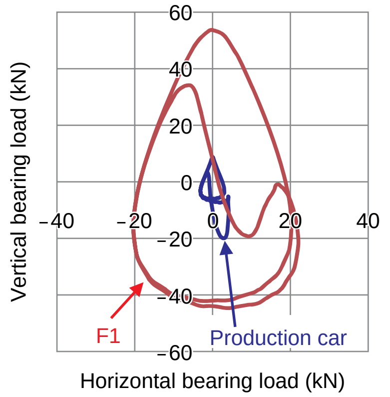


Fig. 14 Conrod bearing loads at WOT, Ps peak revolution

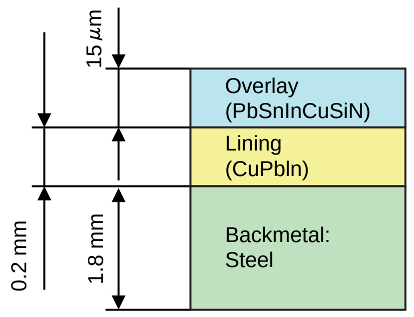


Fig. 15 Former bearing layer

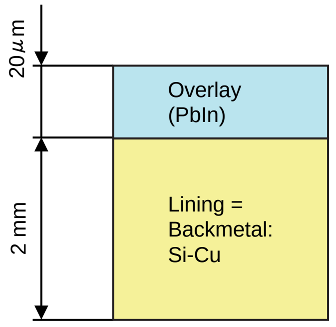


Fig. 16 Si-Cu bearing layer

conductivity, and sliding performance was employed in order to respond to further increases in engine speed.

No active development programs were conducted in the area of main bearings, but in 2005 roller bearings were introduced as a measure to reduce sliding friction. To achieve resistance to high loads, the outer wheels and the rollers were manufactured from tool steel, and the optimum roller diameter to disperse surface pressure was selected by studying diameters in 2 μm increments.

However, despite these elaborated measures, pitting of the sliding surfaces and fractures of the retainers remained issues, and completing the minimum distance of 1500 km set that year represented a challenge. In addition, issues in the seals of the throw chambers for the journal bearings resulted in communication between the chambers, and the resulting pumping loss was judged as having largely canceled out a 4 kW increase in power achieved through the reduction of friction. The use of the roller bearings was therefore ceased.

8. Crankshaft Development

The crankshaft converts the combustion pressure acting on the pistons into torque and outputs this torque. Because combustion pressure and inertial forces mutually act on the crankshaft, the achievement of bending and torsional strength and the adjustment of the specific frequency of vibration of the crankshaft are important factors in the strength of the crankshaft. Oil supply lines also fulfill an important function, helping to ensure a stable supply of oil to the bearings and the counterweight employed to reduce the vibration and friction imparted to the engine by the reciprocating parts.

Preventing fracture of the crankshaft due to torsional resonance and reducing friction and vibration are not easy matters in high-speed and high-power engines such as Formula One engines. For this reason, from the engine concept stage, studies were conducted using CAE and past data with a focus on the vee angle (limited to 90° from 2006) and order of ignition in order to determine the diameter, length, and weight of the pins and journals as well as the mounting method of the clutch. When the actual engine was produced, measurements of the speed at which the crankshaft resonated, the torsional angle of the crankshaft when resonating, and crankshaft friction were taken, and counterweight specifications providing a good overall balance were determined.

The important points were to attempt to ensure that no resonance peak existed in the standard speed range of 15000 to 19000 rpm, and that the torsional angle of the crankshaft when resonating was sufficiently small.

In the initial configuration, the clutch was connected to the crankshaft by a fine input shaft, but from 2003, the clutch was directly connected to the crankshaft in order to prevent crankshaft resonance from occurring in the standard engine speed range, by reducing the speed at which the crankshaft resonated (Fig. 17). The transition to V8 engines in 2006 necessitated an increase in the speed at which the crankshaft resonated, and in this year a method of attaching the clutch via a short input shaft was adopted (Fig. 18).

The use of the high-strength material GKHYW (corresponding to AFNOR 32CDV13) and application of a technology helping to enable the removal of the brittle layer of the oil holes also proved effective, and from 2003 there were no instances of fractures of the crankshaft due to torsion.

In order to reduce the counterweight radius, from 2004 a separate weight was produced from a tungsten alloy (“heavy metal” below) with a specific gravity of 17.5 g/cm^3. This counterweight was then bolted to the crankshaft. The production time for this part, with its different rate of balance, was reduced from three months to three weeks, and it could also be rapidly adjusted (Fig. 19).

Crankshaft fractures due to torsion had been prevented, but cracks in the pin fillet R due to bending, while rare, continued to occur. To respond to this issue, a stress relief groove with a shape designed using CAE

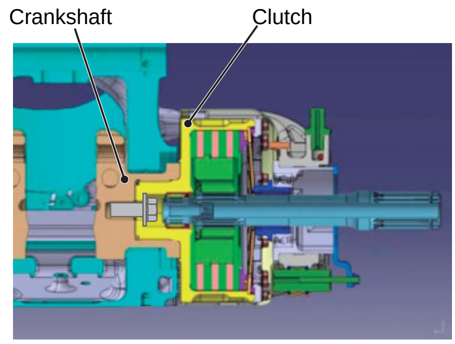


Fig. 17 Fastened clutch

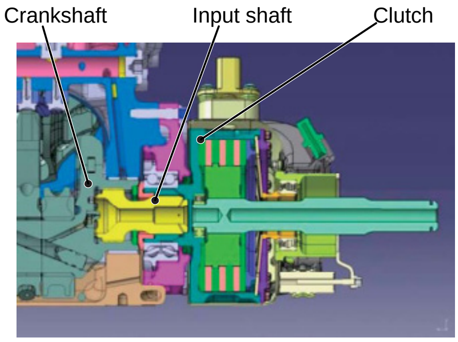


Fig. 18 Transmission with input shaft clutch

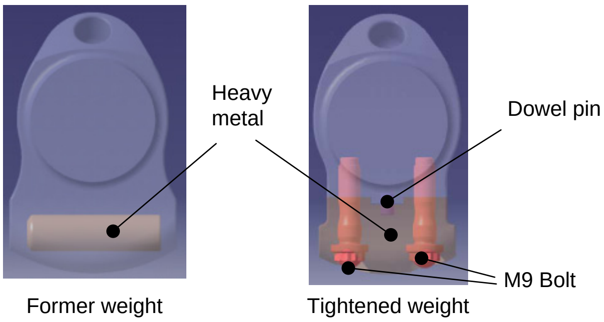


Fig. 19 Crankshaft counterweight history

was machined in the lower section of the fillet R. This prevented the occurrence of cracks in the fillet R, resulting in no further incidence of fracture in the crankshaft unit (Fig. 20).

The stability of the volume and pressure of the oil supply from the crankshaft pin journals has an effect on the lifespan of the conrod bearings.

In Formula One engines, oil is not supplied to the crankshaft from the main journals, as in mass production vehicles. A center oil supply method is employed, in which oil is supplied from the front end of the crankshaft (Fig. 21).

The supply of oil from the main journals as in mass production vehicles would supply oil against centrifugal force. In the case of a high-speed crankshaft, this results in significant loss due to pressure loss and leaks. By contrast, center oil supply, in which oil is supplied from the center of the front end of the crankshaft, is a highly efficient method, using centrifugal force to pressurize the oil. However, even when using a center oil supply method, it is only the first throw to which oil can be supplied without resistance. The oil sent to the later throws will to a certain extent be supplied against centrifugal force, necessitating a layout in which the oil channels deviate as little as possible from the center of the crankshaft. In addition, the use of a center oil supply method does not necessarily resolve all oil supply issues.

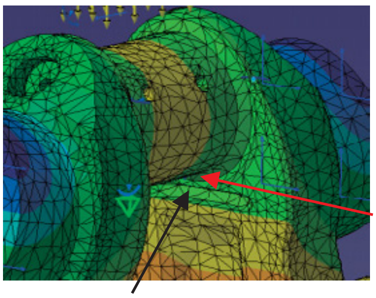


Fig. 20 Crankshaft stress relief groove

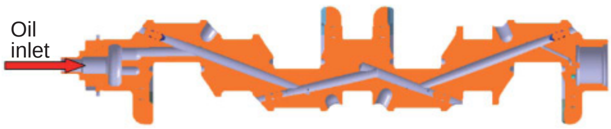


Fig. 21 Crankshaft center oil supply passage

Pin metal seizures also continued to occur as a result of factors including momentary oil supply abnormalities, due to instability of supply from the oil tanks, and oil pressure pulsations, due to the pin oil supply holes being struck by the bearings.

Development of a hollow crankshaft was commenced in order to achieve a weight reduction of 1 kg or more and comprehensively resolve these issues. Friction welding was employed to help enable the main journal of the crankshaft to be made hollow, after which it was able to function as a combined oil channel, oil tank, and damping chamber (Fig. 22).

The hollow crankshaft exceeded initial predictions in helping to increase oil supply. The instantaneous minimum oil pressure that had previously been negative due to oil pressure pulsation returned to 500 kPa, and prospects for durability were also good. However, in 2006 the regulations prohibited the use of the welded crankshaft, and the new crankshaft was never employed in races.

9. Conclusion

(1) In the engine development process conducted during Honda's third Formula One era, development efforts related to the reciprocating parts and crankshafts, which can be considered the core of the engine, produced technologies that helped to enable the engine to satisfy the stipulation for a lifespan of 1500 km at an upshift engine speed of 19600 rpm, without having to protect the parts through retardation of the ignition timing.

(2) The achievement of weight savings, reduction of sliding resistance, and modification of the lubrication method helped to reduce friction and realize a high level of durability and reliability.

(3) During the third Formula One era development process, designers became able to formulate 3D models and conduct CAE analyses, helping to enable the realization of greater accuracy in initial designs and more rapid response to issues. The pistons and bearings are crucial to the achievement of durability and reliability, and the ability to reflect the results of analyses of cooling and heat dissipation performance in designs in particular helped to enable responses to be effected to demands for increased lifespans.

(4) While the individual technologies developed for Formula One engines and the development methods employed

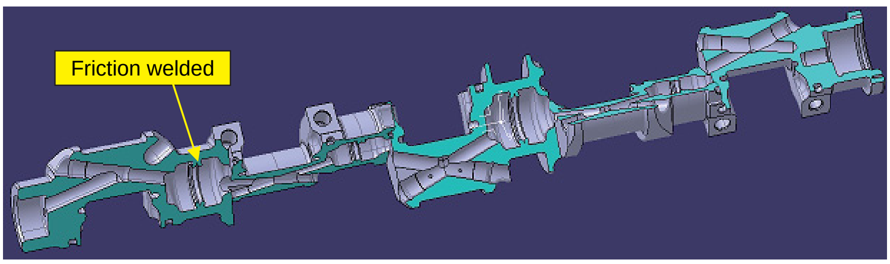


Fig. 22 Hollow crankshaft

will not necessarily be used in mass production vehicles, there are numerous points of similarity in terms of the realization of high levels of efficiency and reliability.

The authors hope that this paper will function as a reference that provides support for future development projects.

■ Author ■

Tetsuo GOTOU

Nagao YANAGISAWA

Development of Valvetrain for Formula One Engine

Shuichi HAYAKAWA* Kazushi OGIYAMA* Masanori TATE*

ABSTRACT

From 2000, the development teams involved in Honda's Formula One engine development program worked as one in development efforts focused on the achievement of increased power through increased engine speed. As part of these efforts, the configuration of the valvetrain was reexamined from the bottom up, and new mechanisms were developed.

To enable the realization of high speed and high lift in the valvetrain, a finger follower system was employed as the valve drive method in place of a bucket tappet system. In addition, the application of a measurement method applying Bezier curves to the acceleration of the valve lift curves and the optimization of the gear train reduced fluctuations in the angular velocity of the camshafts and enhanced valve motion.

Furthermore, it was necessary to reduce the excessive friction due to increased engine speed. In response to this, the pneumatic valve return system was advanced by reexamining the configuration of the cylinder heads. The development of electronically-controlled regulator system also contributed to reducing friction.

1. Introduction

Competition in technological development is particularly intense in Formula One, the pinnacle of the world's auto races. Merely maintaining competitiveness represents technological stagnation or regression – constant advances are demanded.

In the development of valvetrains for Formula One engines to respond to these demands, increased speed and higher lift are sought in order to increase power, but this generates the issue of controlling increases in friction.

With increased power and higher lift as the targets, the valve drive mechanism was redesigned, new materials were employed, and surface treatments such as diamond-like carbon (DLC) were applied, in addition to the achievement of maximal weight savings in the moving parts. Moreover, efforts including reexamination of the valve lift characteristic and the gear train layout reduced lift load at the same time as enhancing the motion of the valves in the high-speed range, thus increasing the allowable speed for the valvetrain.

Changes in Formula One regulations placed restrictions on the materials able to be used, set limits on engine speed, and stipulated that components must be capable of long mileage. This made the reduction of frictional losses in the engines an important issue, and resulted in a significant change in the direction of development efforts.

In response to this trend, a jet-type air control system (J-valve mechanism) was newly developed to modify the pneumatic valve return system (PVRS). This system enabled oil agitation resistance in the PVRS cylinder to be reduced, resulting in reduced motored friction of the valvetrain.

Looking back on development efforts from 2000 onwards, this paper will provide an overview of technologies developed to increase engine speed and reduce friction in Formula One valvetrains with the aim of achieving increased power, and will discuss the performance of these technologies.

2. Overview of Valvetrain

Table 1 shows the main specifications of the valvetrain used in Honda's 2008 Formula One engine, and Fig. 1 shows an external view of the components of the valvetrain.

A variety of studies were conducted of the valve, including those on the angle of the intake and exhaust valves from the perspectives of increased volumetric efficiency and enhanced combustion. For the intake side, a compound valve layout that also displays an angle of inclination in the longitudinal direction of the engine was adopted. The intake and exhaust valves were the components in which the maximum weight savings were sought, and development proceeded on materials for valves so that weight could be minimized within the ratio

* Automobile R&D Center

Table 1 Valvetrain main specifications

Year			2008
Engine code			Honda RA808E
Head			J-Valve
PVRS		Control	P2/P3EAR control
		Air bottle	Carbon bottle 570 cm³
Gear train			AVRS-G
Cam	IN	Lift , valve timing	13.5 mm 19/64
		Material	SCM435 nitriding
		Surface treatment	DLC
	EX	Lift , valve timing	13.0 mm 19/64
		Material	SCM435 nitriding
		Surface treatment	DLC
Valve	IN	Stem diameter	ϕ5.8 (hollow stem ϕ3.5)
		Valve diameter	ϕ41.6
		Material	Ti6246
		Surface treatment	DLC
	EX	Stem diameter	ϕ5.8
		Valve diameter	ϕ32.4
		Material	KS64411Ta
		Surface treatment	DLC
Finger follower	IN/EX	Material	SNCM815 carburizing
		Surface treatment	DLC
Air spring seal		IN/EX	PTFE
Reciprocating mass (g)		IN	50.0
		EX	45.6
Maximum engine speed for valvetrain (rpm)			20300
Upshift engine speed (rpm)			19000

range of specific weight to stiffness as stipulated in the Formula One regulations.

A camshaft and finger followers was used for the opening and closure of the valves, and a DLC coating was applied to the sliding components. In addition, the application of a PVRS using compressed air in place of normal metal springs reduced reciprocating mass, and prevented the surging at high speeds that originated in the metal springs. Figure 2 shows a view of the PVRS as fitted in the vehicle. The system incorporated an air bottle filled with high-pressure air and two regulators,

Fig. 1 View of valvetrain

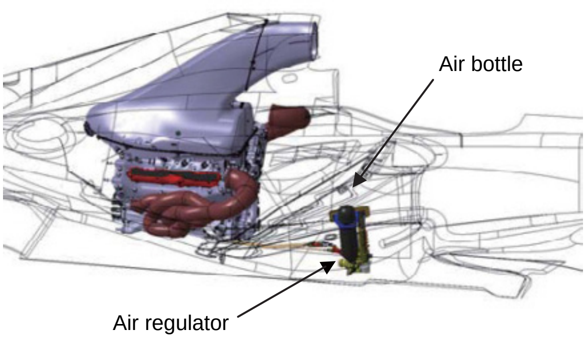


Fig. 2 View of air spring system

for supply pressure and discharge pressure.

Torque was transmitted by the gears from the crank to the camshaft. This helped prevent delay in the valve timing due to high drive torque and resonance generated in the chain or the belt. The gear train was designed to reduce fluctuations in the angular velocity of the camshaft caused by vibration, thus enhancing valve motion and reducing friction.

3. Technologies for Increased Engine Speed

3.1. Changes in Speed Targets

Figure 3 shows changes in the maximum allowable engine speed for valvetrain performance in the Honda Formula One engine development process from 1990 to 2008. In the valvetrain, the seating load increases rapidly due to valve bounce at a higher engine speed. This might reduce durability and reliability of each component of the valvetrain, resulting in a failure. In response to this, valve motion should be enhanced so that valve bounce would not occur even at the maximum engine speed. The maximum allowable engine speed for valvetrain performance can be defined as the limit speed up to which valve bounce would not occur, taking the over-revving projected to occur in the circuit driving environment into consideration. At the commencement of development efforts, increases of approximately 400 rpm per year were achieved, and this contributed to increased power. From 2004, regulations specified long mileage, and the mileage for each engine consequently increased. From 2007, maximum engine speed was limited to 19000 rpm, increasing the frequency of wide open operation of the throttle valves. These changes made increased valvetrain durability and reliability necessary, and development therefore proceeded in order to achieve a maximum engine speed for valvetrain performance of 20000 rpm or more.

3.2. Reduction of Reciprocating Mass

Figure 4 shows a cutaway view of the valvetrain and the reciprocating mass of each component of the valvetrain. Weight savings were achieved by the application of methods including the use of a finger

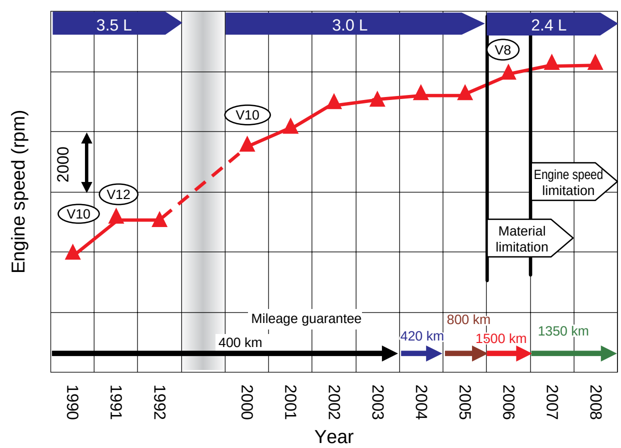


Fig. 3 Changes in maximum allowable engine speed for valvetrain

follower mechanism to drive the valves, instead of a bucker tappet system, and the reduction of the diameter of the PVRS cylinder. In addition, efforts in materials development and the use of CAE enabled the durability and reliability of the valves to be ensured with the minimum weight increase, despite restrictions on the materials that could be employed and the requirement for long mileage. These efforts reduced the reciprocating mass of the intake-side valvetrain by 21.7 g (30%) between 2000 and 2008.

3.2.1. Development of finger followers

The drive system for the valves in a Formula One engine must use lightweight parts in order to achieve increased power, and at the same time must enable increased valve lift in order to increase the volume of intake air. Using the previous bucket tappet system, it was necessary to increase the diameter of the bucket tappet to achieve higher lift, and it was challenging to balance this necessity with the achievement of weight savings. Because of this, finger followers were employed from 2002 onwards, enabling weight to be reduced by 13.5 g and valve lift to be increased by 1.0 mm or more. Figure 5 shows a comparison of the valve drive systems used in Honda Formula One cars.

The shape of the finger followers was determined through studies using 3D models and CAE based on a variety of considerations, including frictional losses at high speeds and high-hertz stress levels as well as the local stiffness and strength depending on inertial loads and the direction of rotation of the cam.

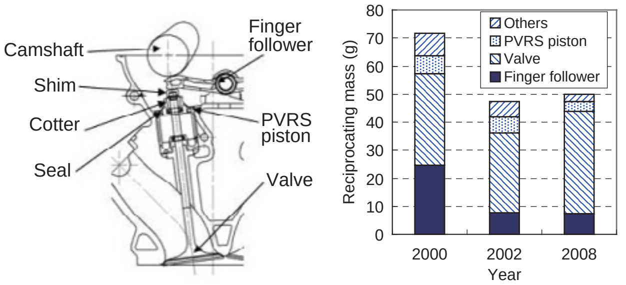


Fig. 4　Reciprocating mass of parts

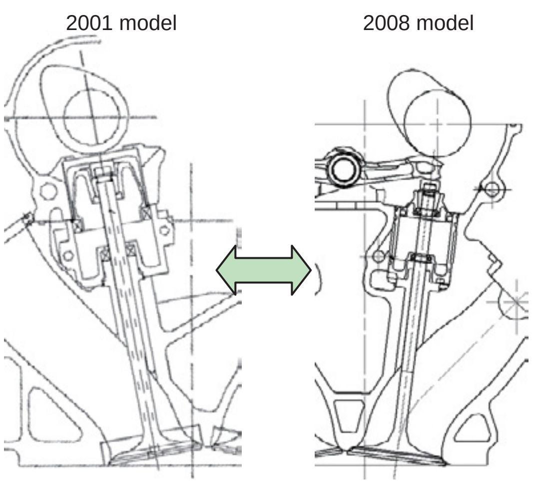


Fig. 5　Comparison of valvetrain layouts

From the perspective of sliding performance, a DLC coating was applied to the cam lobe and the finger follower top pads in order to help ensure durability and reliability at high speeds and high hertz stresse levels, and to reduce friction. In the development of the DLC coating, factors, including the balance of film hardness between the parts and the formation of the films, were optimized. By comparison with the initial stage of DLC development, an increase in surface pressure limit of 17% in high-load operating environments was achieved, and friction was reduced by approximately 2.0 kW per year.

From the perspective of finger follower geometry, positioning the camshafts almost exactly on the stem of the intake and exhaust valves brought the lever ratio close to 1 and reduced surface pressure, in addition to enhancing valve motion by increasing stiffness. Furthermore, positioning the pivot of the finger follower on the same axis as that of the intake and exhaust valves enabled the finger followers to be lengthened and helped to ensure the formation of an oil film on the sliding surfaces.

The oscillation of the finger followers and the inertial load on the arms generated a large moment on the valve stems, raising concerns over bending of the valve stems or fracture of the valve heads. Minimization of the bending moment on the valve stems was an effective means of responding to these concerns. The curved shape for the bottom pads was therefore designed to enable the amount of movement of the contact points with the valve stems to be reduced and their positions to be located in the center of the valve stems.

3.2.2. Development of intake and exhaust valves

In order to increase volumetric efficiency in the Formula One engine, it was necessary to increase the diameter of the head sections of the intake and exhaust valves while reducing the diameter of the valve stems. It was also necessary for the valves to be capable of withstanding the inertial forces and seating loads generated by acceleration of more than 6000 G under combustion gas temperatures of 700 °C or more. Considering these demands, materials possessing high heat resistance strength were developed, and CAE was employed to enable selection of appropriate valve shapes (Fig. 6). Combustion gases and intake air generated a

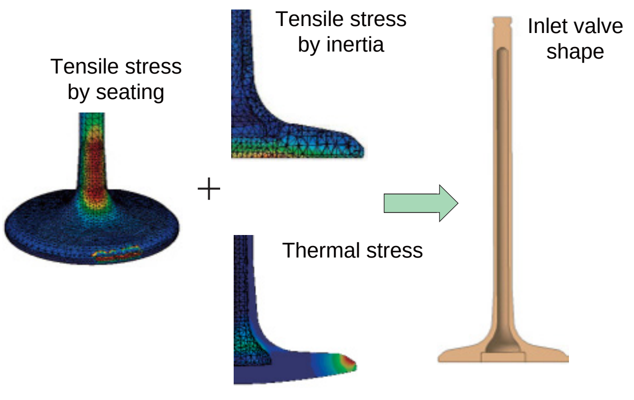


Fig. 6　CAE for inlet valve

significant difference in ambient temperature between the fireface and the back of the intake valves, and there were concerns that high levels of heat stress might be generated on the exteriors of the heads, resulting in seat chipping. Analyses were therefore conducted of stresses generated by heat and by inertial forces, and an umbrella shape was developed that would prevent the level of stress from exceeding the fatigue strength of the material during operation. The swaying of the valves at high speeds also generated partial contacts when the valves were seated on the valve seats. This caused stress to be concentrated at the points of connection between the valve heads and valve stems. As a countermeasure, the valve stems were tapered in order to reduce stress.

Honda began using titanium aluminum (TiAl) materials in 2002, and employed them in race vehicles until 2005. The use of these materials enabled the diameter of the valve stems to be reduced from ϕ 6.6 to ϕ 4.5 and the reciprocating mass of the valves to be reduced by 5.0 g against conventional titanium (Ti) materials. TiAl alloys are intermetallic compounds, and the particular machining employed for these materials produces cracks around which residual compressive stress exists. Because of this, the valves frequently broke during their first period of use. In order to resolve this quality issue, quality control was implemented by subjecting all valves to eddy current testing rather than fluorescent penetrant inspection in order to detect cracks, enabling the realization of a level of durability and reliability high enough to meet the standards of the long mileage regulations.

From 2006, restrictions were placed on materials in Formula One, and Honda had no option but to change its materials. It was necessary to hollow the valve umbrellas and valve stems in order to achieve an identical or lower weight using Ti as had been possible using TiAl. However, in the initial stages, insufficient strength and stiffness resulted in frequent breakages of the valve heads. Measurements of the stress on the valve heads and the temperature of the heads were therefore taken, and analyses were conducted of factors including the motion of the valves during firing. The optimal umbrella shape was selected by feeding the results back into CAE, enabling the realization of a low weight and a high level of stiffness, equivalent to those available from TiAl.

3.3. Optimization of Cam Profile

The cam profile has a wide range of effects on performance. The achievement of increased volumetric efficiency by increasing valve lift and the optimization of the duration angle and the opening and closing timing of the valves enables low- and medium-speed torque to be increased. The stabilization of combustion increases drivability, and the realization of a higher compression ratio increases fuel efficiency. Figure 7 shows the lift curve for the intake valves employed in Honda's 2008 Formula One engine. Study of the valve lift curves of Formula One engines using 3D ηv simulations showed that the positive acceleration of the lift curves reached 50 mm/rad^2 or more.

In the design of the valvetrain, considerations of reliability and durability and the realization of increased speed made it essential to set a level of valve lift acceleration that considered the achievement of a balance between the lift load for the PVRS and inertial forces. A Bezier curve was therefore applied to the lift curve. This type of curve enables designers to freely set valve lift acceleration, and at the same time, vary the lift curve in real time, simplifying the realization of the desired lift curve. By this means, the negative acceleration of valve lift was optimized with consideration of the PVRS pressure characteristic, enabling the valve lift load and inertial forces to be balanced to the limit point and the maximum allowable speed for the valvetrain to be increased by 1000 rpm. In addition, in order to balance increased lift with a high compression ratio, the positive acceleration of the valve lift was designed to display two stages, with the level of lift being reduced only when the clearance between the pistons and the valves was low. This enabled the compression ratio to be increased by 0.3. Figure 8 shows the two-stage acceleration designed using the Bezier curve.

3.4. Reduction of Fluctuation in Cam Angular Velocity

Torsional vibration was one of the factors that had a negative impact on valve motion. This generated a nonlinear resonance phenomenon in the camshafts. Fluctuation in the angular velocity of the camshafts is thought to occur due to fluctuations in the motion of the camshafts as a whole resulting from resonance with the

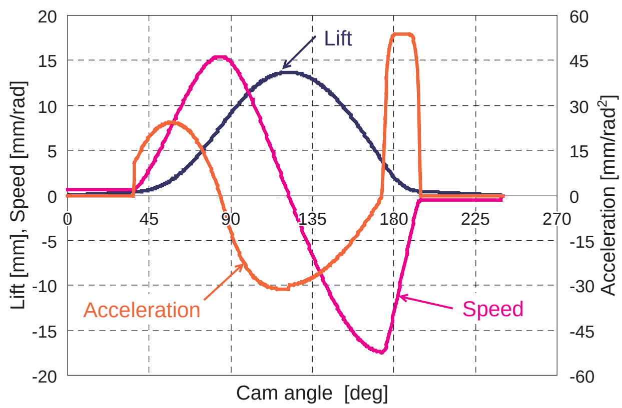


Fig. 7 Inlet lift curve of 2008 model

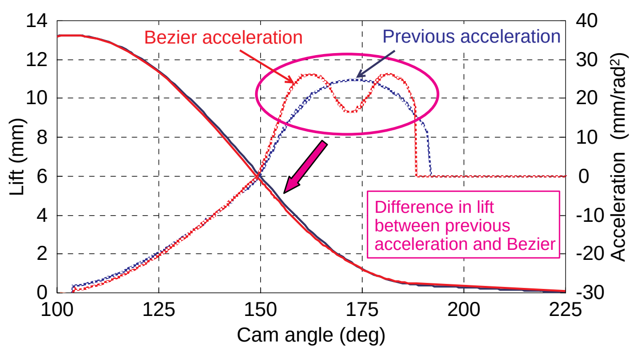


Fig. 8 Bezier lift curve

gear train, and the twisting of the camshafts between cam lobes due to the reaction force of the valve lift. It instantaneously increases or decreases the speed of rotation of the cams, reducing the motion of the valves. This makes it necessary to reduce the fluctuation of the angular velocity of the camshafts at high speeds. The following factors can be indicated as reducing fluctuations in the angular velocity of the cam:

(1) The drivetrain moment of inertia
(2) The drivetrain spring constant
(3) Gear backlash
(4) Cam drive torque
(5) The damping coefficient of the dampers, etc.

Given that the cam drive torque is determined prioritizing vehicle dynamics performance in a Formula One engine, the remaining four factors were focused on. In addition, it was essential to reduce variation in the angular velocity of the cams without increasing the weight of the engine or raising its center of gravity, two factors that have a significant effect on vehicle body performance.

3.4.1. Angular Velocity Reduction System (AVRS)

The AVRS was a mechanism that reduced fluctuation in the angular velocity of the cams by attaching weights to the rear ends of the camshafts (on the side opposite the gear train) in order to increase the moment of inertia and shift the resonant frequency of rotational vibration to the low-speed range. Figure 9 shows the configuration of the AVRS and Fig. 10 shows its effect in reducing fluctuation in the angular velocity of the cams.

The use of weights to increase the moment of inertia of the camshafts and the adjustment of gear backlash in order to control fluctuation of the angular velocity of the camshafts in the high-speed range increased the valve motion speed by 1500 rpm, and reduced friction by 6 kW.

3.4.2. Angular Velocity Reduction System with Gears (AVRS-G)

Figure 11 shows the configuration of the AVRS-G. Figure 12 shows fluctuations in the angular velocity of the camshafts. When the AVRS was employed, the fluctuation in the angular velocity of the rear ends of the camshafts became higher than that of the front ends, and there was therefore a margin for further reduction of fluctuations. The issue was considered to originate in the fact that the rear ends of the camshafts to which the weights were affixed were the free ends of the shafts. The AVRS-G was therefore developed. This mechanism employed gears to connect both the front and rear ends of the intake and exhaust camshafts, and increased the spring constant of the camshafts in their entirety. The use of the AVRS-G reduced the difference in twisting between the front and rear ends of the camshafts at the same time as reducing fluctuations in angular velocity, thus assisting in reducing the stiffness of the camshafts. This enabled the high specific gravity tungsten alloy used for the rear weights in the AVRS mechanism to be replaced by steel rear gears and the hollow diameter of the camshaft to be increased, resulting in a weight saving of 2 kg around the camshaft while maintaining identical valve motion.

3.4.3. Cam damper

While never employed in racing due to Honda's withdrawal from Formula One, the achievement of even

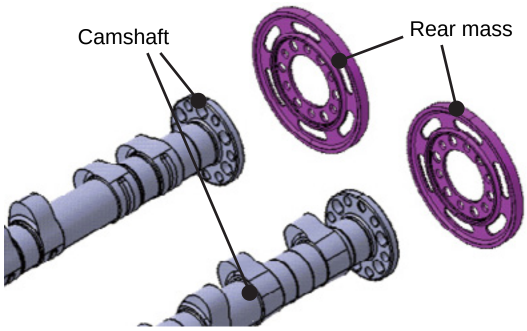


Fig. 9 View of AVRS

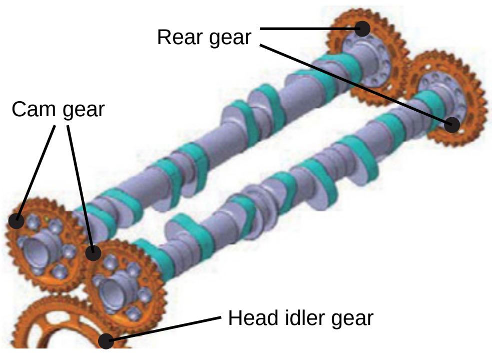


Fig. 11 View of AVRS-G

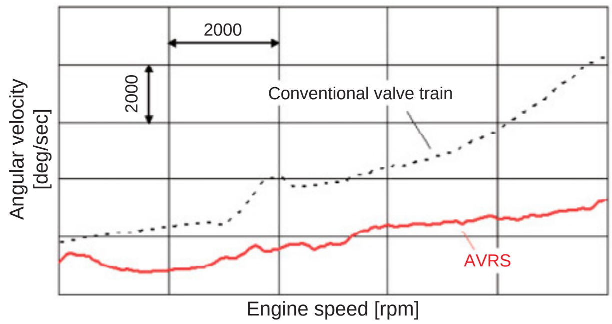


Fig. 10 Angular velocity of AVRS (2002 engine)

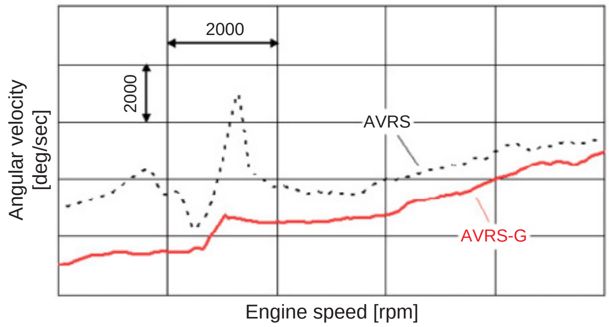


Fig. 12 Angular velocity of AVRS-G (2003 engine)

higher lift was studied in order to further increase power, making it necessary to correct a resulting decline in valve motion. As a means of achieving this goal, cam quills with a damping effect were developed to be fitted between the cam gears and the camshafts, and viscous dampers were attached to the rear ends of the camshafts. Figure 13 shows the configuration of the cam damper system, and Fig. 14 shows its effect in reducing fluctuations in angular velocity.

The stiffness of the quill shafts connecting the cam gears and camshafts and the viscosity of the silicon encased in the viscous damper were adjusted to achieve the desired damping coefficient. Cam drive torque and resonant frequency were calculated for each specification, and specifications that did not produce extreme peaks of fluctuation in angular velocity in simulations were employed.

The combined use of the quills and the viscous dampers reduced fluctuations in angular velocity, and increased the valve motion speed by 1100 rpm against the AVRS-G.

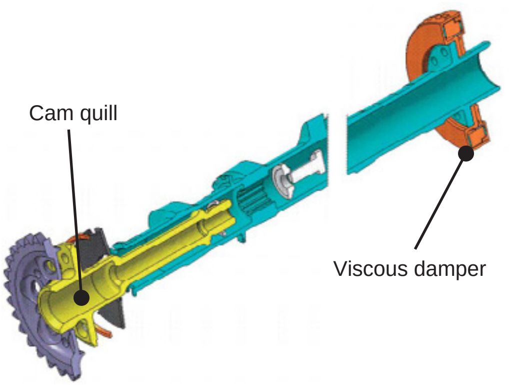


Fig. 13 Cam damper system

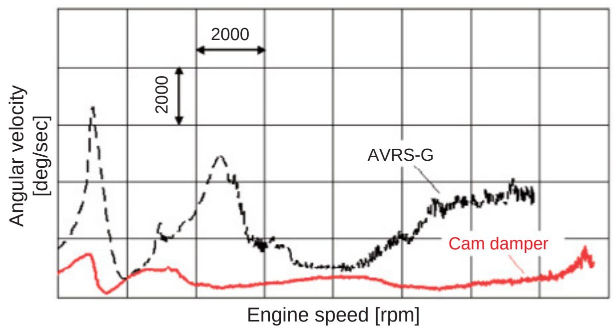


Fig. 14 Angular velocity of cam damper (2008 engine)

4. Friction Reduction Technologies

4.1. Progress in Friction Reduction

Figure 15 shows the progression of reduction of the motored friction of valvetrains in Honda Formula One engine developments from 2000 to 2008.

In the initial stage of development for Formula One, the reduction of reciprocating mass and the control of fluctuations in the angular velocity of the cam enhanced valve motion and reduced lift load. In addition, the reduction of the level of oil in the PVRS cylinders was studied, and in 2005 the employment of a J-valve mechanism in an evolution of the PVRS enabled the oil level to be reduced to zero. These advances reduced friction by 20 kW in the 2005 V10 engine.

4.2. J-valve Mechanism

4.2.1. Aim of development of J-valve mechanism

PVRS is generally employed in the valvetrains of Formula One engines, which are high-speed engines, and Honda also employed the system from 1992.

Conventional PVRS used a check/relief mechanism, and supplied and discharged air with one-way valves fitted at the inlet and outlet of the PVRS cylinder to control pressure. However, the volume of air that could be used on a race was determined by the capacity of the bottle fitted in the vehicle, and it was necessary to control the volume of air consumption from the outlet. To enable this, a system was designed in which oil accumulated in the PVRS cylinder, and the discharge of the oil was prioritized over the discharge of the air. However, the oil generated agitation resistance during valve lift, and became a factor in increased friction.

Figure 16 shows the differences between the check/relief and J-valve mechanisms. Figure 17 shows the correlation between the level of oil in the cylinder, friction, and air consumption. The newly developed J-valve mechanism controlled air consumption by using a single orifice for supply and discharge of air. By separating air consumption from the production of

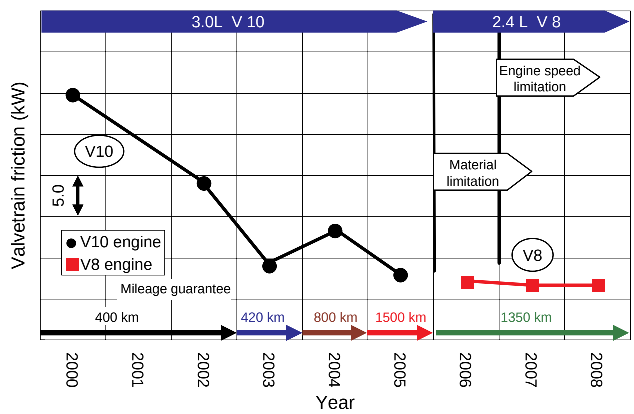


Fig. 15 Valvetrain friction performance

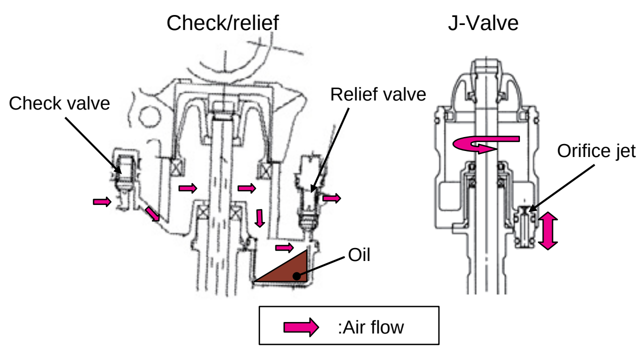


Fig. 16 Comparison of PVRS layout

friction, this mechanism obviated the necessity for oil in the PVRS cylinder, and reduced friction by 3 kW. In addition, doing away with the one-way valves and the air channels on the cylinder heads resulted in a weight saving of 1 kg.

4.2.2. Configuration of J-valve mechanism

Figure 18 shows the system configuration of the J-valve mechanism. One section of the system incorporates an air bottle, a primary decompression mechanism, and an electric air regulator designated P2 (P2 EAR), which regulates the supply pressure via an air injector. Air is supplied to the engine from this section. The outlet side is provided with an electric air regulator designated P3 (P3 EAR), which similarly regulates discharge pressure using an air injector. The development of these two EAR units enabled variable control of pressure.

Figure 19 shows a section view of the PVRS cylinder. Two concerns in the development of the J-valve mechanism were: 1) the feasibility of a configuration of the seal rings that helps prevent oil from flowing into the cylinder; 2) the durability and reliability of the stem seals in the absence of oil accumulation. To respond to these concerns, the seal rings were formed from PTFE jackets with a ripple shape and rubber rings with an X shape. This produced a stable tensile force and enhanced oil discharge performance. The durability and reliability of the stem seals was increased by reexamining the configuration of the cylinder heads and employing a layout that enabled forced oil supply only to the seals.

4.2.3. Control

The J-valve mechanism was fundamentally designed as a configuration in which there was no oil inflow to the PVRS cylinder. However, the seals did allow tiny amounts of oil into the cylinder, which would increase the internal pressure of the cylinder if not dealt with. An increase in the internal pressure of the cylinder would lead to increased friction, and in addition seal ruptures and sliding abnormalities could occur and lead to engine troubles. In addition, the volume of oil inflow to each PVRS cylinder was unknown. To respond to this issue, a system of air line purge control was developed enabling forced discharge of the oil that had accumulated in the PVRS cylinder. Air line purge control enabled variable control of the pressure in the PVRS cylinder by means of electronic control of each EAR unit, and forced air flow through the air channels. Figure 20 shows a diagram of the P2 EAR unit.

The forced discharge of oil in the air line purge was conducted by increasing the pressure on the inlet side

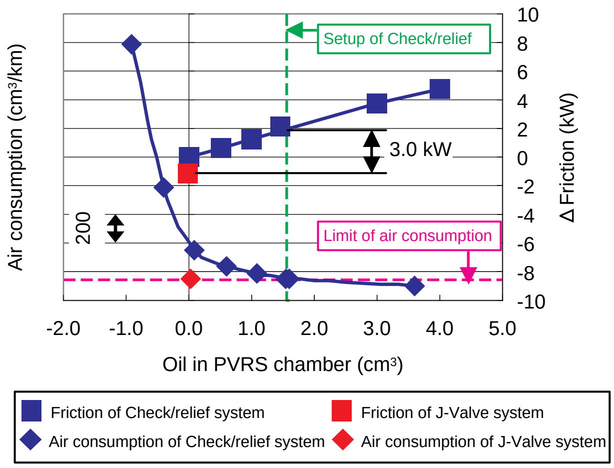


Fig. 17 Relation between air consumption and friction

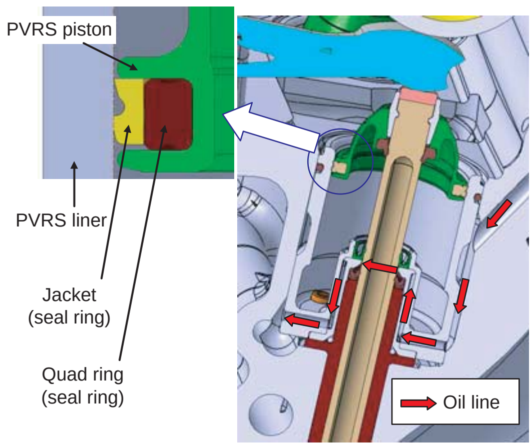


Fig. 19 Section view of PVRS cylinder

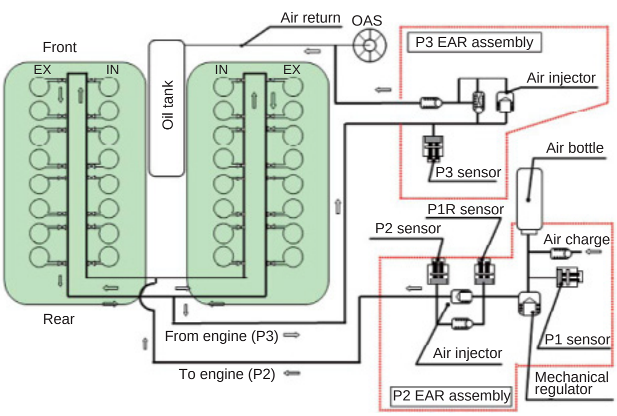


Fig. 18 J-Valve system configuration

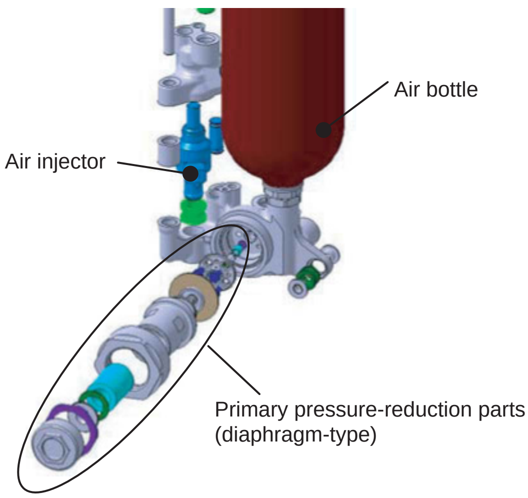


Fig. 20 View of P2 EAR

and creating a pressure differential with the outlet side. In order to achieve complete oil discharge, it was necessary to create a pressure differential that was able to generate an air flow speed of 25±5 L/min, and to use a volume of 8 L or more of air. The air line purge was automatically conducted when a preset number of laps (distance) had been covered, and the oil was periodically discharged from the PVRS cylinder throughout the race. The air line purge could also be conducted manually in the event of abnormal functioning or during pit work.

When long mileage regulations for Formula One engines came into effect, the amount of use of the idling range (5500 rpm and below) increased. Because the inertial load in the valvetrain is low in the idling range and the lift load becomes dominant, cam surface pressure is high. This raised durability and reliability concerns over the sliding of the cam. To address this issue, a control method that helped prevent excessive lift loads in the idling range was developed which reduced the surface pressure of the cams and helped to ensure reliability and durability. Lift loads during idling were reduced by reducing the air channel pressure to a level lower than that of during race driving, using the P3 EAR unit. Figure 21 shows an overview of lift load switching control during idling.

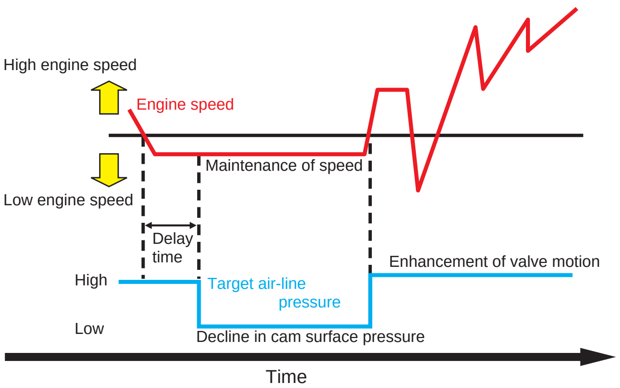


Fig. 21 Pressure control during idling

5. Conclusion

During Honda's Formula One engine development program between 2000 and 2008, the company developed a valvetrain reconciling the achievement of high speed with low friction. The following results were achieved:

(1) Finger followers were employed as the valve drive method, and materials development was conducted and measurement technologies and CAE introduced to respond to materials and long mileage regulations, resulting in a 21.7 g (30%) reduction of the reciprocating mass of the valvetrain.

(2) To respond to the necessity for increased power in order to maintain competitiveness, valve lift acceleration and the gear train layout were reexamined, enabling the allowable speed for the valvetrain to be increased by 2700 rpm.

(3) The reduction of valve lift load through the enhancement of valve motion, the reduction of frictional losses through the use of a DLC surface treatment, and the reduction of oil agitation resistance through the use of a J-valve mechanism helped to reduce motored friction of valvetrain by 23 kW at an engine speed of 18000 rpm.

■ Author ■

Shuichi HAYAKAWA

Kazushi OGIYAMA

Masanori TATE

Development of Induction and Exhaust Systems for Third-Era Honda Formula One Engines

Ken NISHIMORI* Yasuhiro MOTOHASHI*
Masayoshi TAKAHASHI* Ryuichi FURUKAWA*

ABSTRACT

Induction and exhaust systems determine the amount of air intake supplied to the engine, and as such are critical elements affecting engine output.

In addition, the layout of the induction and exhaust systems affects the vehicle's aerodynamic performance, and so it must be considered together with vehicle development.

At first, there were few CAE software and computer resources available, and induction and exhaust system components were produced by measurement and guesswork so that development was largely performed on a trial and error basis, but in recent years, the 3D-CAD and CAE software has advanced so quickly, and computer resources have expanded so much, that development is done by simulation.

The enhanced phenomenon elucidation and forecast precision have made it possible to shorten the time it takes to determine specifications and reduce development costs.

1. Introduction

The first thing required of a racing engine is power performance, but transient characteristics and vehicle package also impact performance. As shown in Fig. 1, the induction and exhaust systems are placed outside the engine, and have a large impact on the package.

Vehicle weight and inertia influence dynamic performance, but at a vehicle weight of 600 kg including the driver, the engine weight accounts for a large percentage of the total, so when designing engine components, one has to make them light while maintaining their necessary functions.

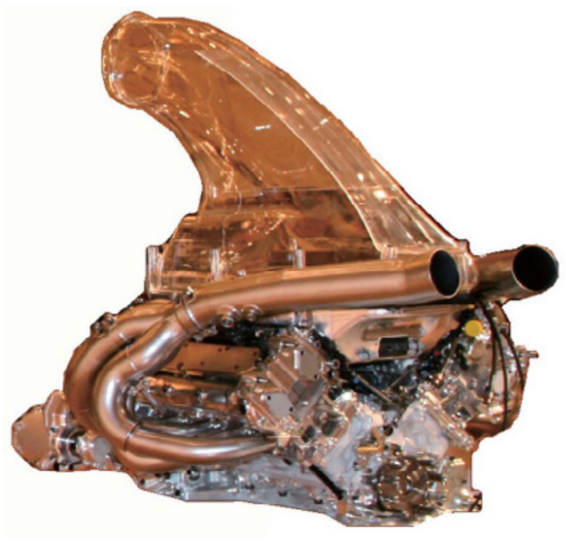


Fig. 1 Engine with complete induction and exhaust

The development of induction and exhaust systems was mainly about lowering flow resistance in 2000, when the initial development was being done, but since 2006, it has been possible to predict dynamic effects during the process of design examination, meaning that one can carry out optimal design, with consideration of vehicle package, in a short amount of time. This paper discusses the content of this development.

2. Induction

2.1. Induction System

2.1.1. Reducing flow resistance

One technique for increasing volumetric charging efficiency is to reduce induction system pressure loss (below, pressure loss). The main factors relating to pressure loss are air filter performance and airbox (below, ABX) form.

Sometimes impurities drift in when the ABX is opened after running, so an air filter is required to avoid having to retire from the race. Because it is located within the air intake passage, however, the filter's flow resistance directly affects intake resistance of the engine. Air filters, therefore, must be designed to reduce flow resistance.

* Automobile R&D Center

Honda's third-era Formula One engines originally used a sponge-type air filter from an overseas manufacturer, but in 2004 Honda began examining a nonwoven fabric-type air filter. In 2004 a Grand Prix event was held in Bahrain, in a dusty area, and the sponge-type filter was not able to trap the fine desert sand and presented the crew with a big challenge.

At first, they used a dry filter that was installed on mass-market vehicles in dusty areas. However, although this increased the scavenging ratio, pressure loss also increased and power dropped by 4 kW, so the use of this filter was limited to certain events.

Subsequently, an investigation was begun on how to achieve enhanced engine durability, reliability and a higher scavenging ratio without diminishing performance.

A wet, nonwoven fabric filter was developed with high scavenging ratio and low pressure loss, which was used starting with the Italian Grand Prix, one of the last events of the season.

Starting in 2006, engines were limited to eight cylinders, which led to frequent induction fires caused by backfiring. Similar phenomena were also seen in CART series V8 engines, along with the phenomenon of induction systems exploding and components scattering in CART races where premixed methanol fuel were used. In Formula One engines, the air filters suffered melting damage (Fig. 2), and simultaneously with this seeping into engines, CFRP components were burnt and vehicles left unable to run.

Engine controls were changed as a countermeasure, while the filter was changed to a highly fire-resistant material.

After that, there were no more troubles in actual driving from filters burning as a result of backfiring.

Because the flow resistance of an air filter is proportional to the square of flow velocity, it is important to evenly distribute and reduce the velocity of the air-flow through the filter in order to limit pressure loss.

Within the scope allowed by the layout and aerodynamic performance, one needs an efficient velocity transformation to the maximum filter area that fits inside the engine cover.

Fig. 2 Damaged air filter

Although it looks like nothing more than a container that simply supplies air to the engine, the ABX is an important component that takes air flowing at a relative speed of 300 km/h and turns it into a homogenous, low-speed flow of air to be sucked into the engine.

Figure 3 illustrates results of ABX CFD examination in 2008.

To reduce flow velocity in a limited package, one has to efficiently expand the air passage in cross-section. The velocity of air taken into the ABX contains a large horizontal constituent.

The layout of the ABX in the vehicle is such that the aperture is high to protect the driver, the ABX is front-mounted on the engine to concentrate the mass, and there are surfaces with much curvature in the front. These front surfaces are subject to separation, which causes variance in the amount of specific intake air volume to each cylinder and a drop in performance. Therefore, the form was optimized using CFD to enhance filter efficiency.

At first, the ABX was developed on the assumption that air is brought in to the ABX inlet homogenously. But a wind-speed simulator (below, WSS)[1] implemented in 2007 was used to simulate actual driving conditions and measured pressure distribution on the inlet duct. This made it clear that the distribution was not homogenous, as illustrated in Fig. 4. The results confirmed that engine power under these circumstances showed a 5 kW power loss as compared to a homogenous condition.

Figure 5 shows CFD results at low vehicle speed. The flow that stagnated at the lower end of the roll hoop rises while containing a Z-direction constituent up to a point close to the air intake and comes flowing into the

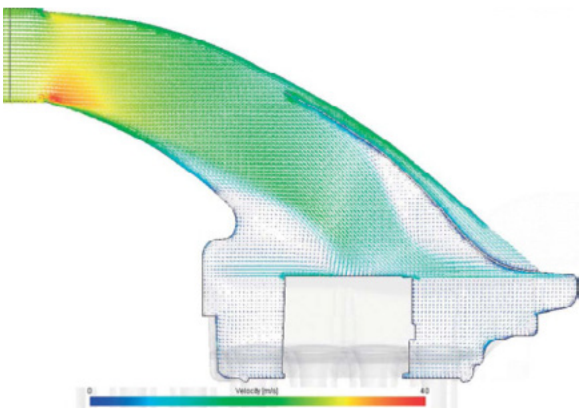


Fig. 3 ABX CFD result

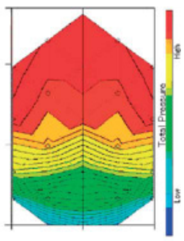


Fig. 4 Total pressure distribution at inlet to ABX

ABX. As a result, separation occurs on the lower surface of the part where the ABX comes in (below, the snorkel).

During development of the 2009 model, the air-flow status of the ABX aperture was considered during examination of vehicle aerodynamics, and development went forward using CFD to make inlet pressure distribution almost uniform.

2.1.2. Using intake pulsation

In an induction system, the intake pulsation caused by the opening and closing of the intake valve interacts dynamically within individual cylinders, among cylinders and between the cylinder banks. By controlling these interactions, one can try to enhance engine performance.

Figure 6 illustrates a variable-length intake system (below, VIS).

A VIS uses the fact that the intake pipe length has a dominant effect on engine speeds at which resonant supercharging effect can be achieved. By using this system, intake pipe lengths are continuously achieved that are appropriate for each engine speed. In addition, corrections are made that correspond to changes in intake temperature.

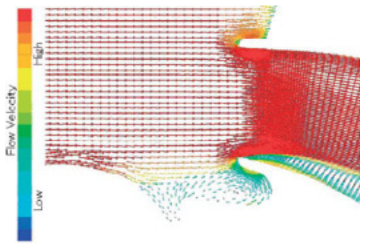


Fig. 5 CFD result of ABX inlet on chassis (150 kph)

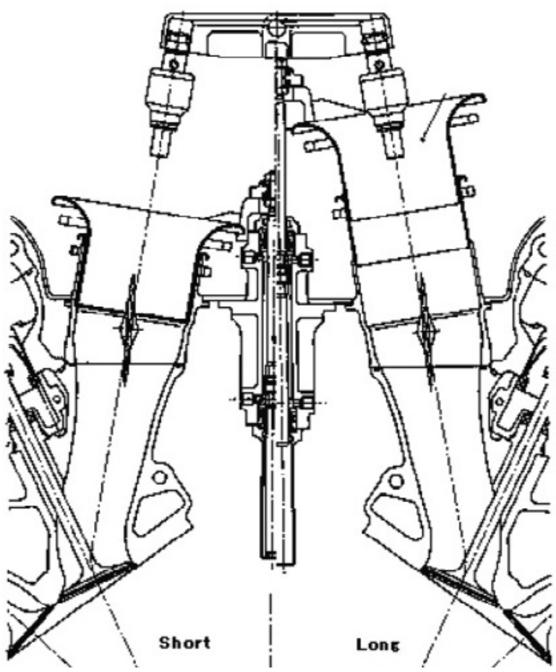


Fig. 6 Schematic view of VIS system

In a V10 engine, intake interference between the banks causes the quadratic standing waves in the ABX to amplify, with the result that the secondary constituent of the pulsation in the port attenuates, interfering with dynamic air intake.

A splitter is an effective way to block interference between the banks. Figure 7(i) shows a splitter on the 2004 (V10) ABX. Adding on the splitter helped enhance power by 7 kW.

The item indicated by the letter “a” in the figure is a component installed to plug the small gap, which boosted power by 2 kW. However, because of the tolerance of this component and the variance that occurred when it was installed, the gap could not be completely plugged, and eliminating the impact the gap had on power was a continuing issue.

Fig. 7(ii) shows the splitter used since 2006.

The changes made to the regulations in 2006 required the use of V8 engines. In addition, the previously described VIS could no longer be used, so the team was required to expand the torque band with a fixed intake pipe length.

The effect of inter-cylinder interference within the induction system in a V8 engine is greater than that in a V10 engine (Fig. 8), and torque characteristics change depending on the form of the induction system. Because of this, inter-bank interference of the four cylinders in the center was controlled, while promoting active interference in the four cylinders in front and back,

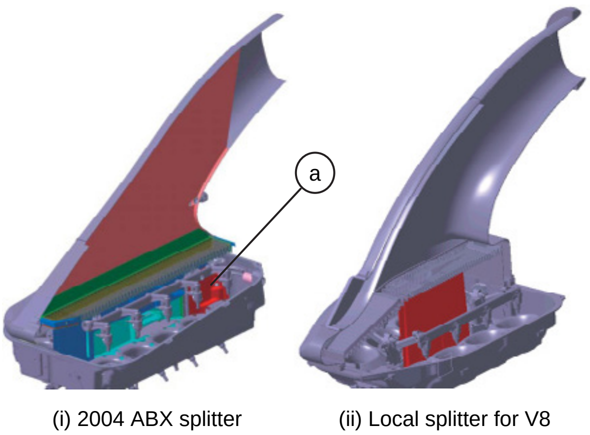


Fig. 7 Schematic view of ABX splitter

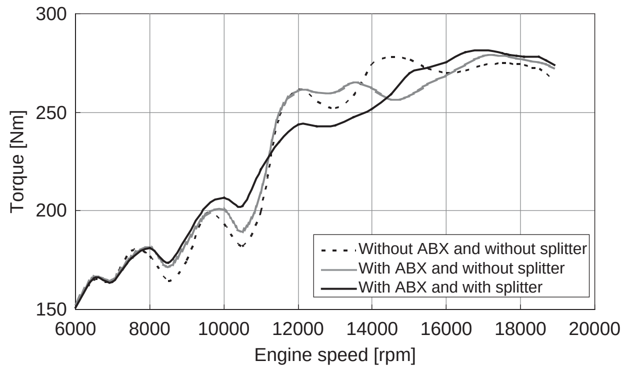


Fig. 8 Effect of splitters

which increases volumetric charging efficiency (below, ηv) over a wide range of engine speeds and flattens torque characteristics.

Figure 9 shows a test ABX for adjusting intake pulsation. Inserts made by a rapid prototyping machine are attached to and used on the interior of an ABX built to a large size. The ABX is made with lightweight CFRP to ensure rigidity in those areas where the inserts are attached.

The form-selection process switched to testing with the above test ABX, which made it possible to find the correlation between ABX wall form and power characteristics in a short amount of time. They were also able to be validated on 3Dηv simulation[2].

Examination of the making of inserts by rapid prototyping machine can be done until just before the test, and furthermore, the replacement with an optional part taking account of test results could be performed quickly.

Also, correlation with 3Dηv simulation was conducted simultaneously, with enough precision to forecast the power-performance benefit at the design stage.

2.1.3. Reduction of intake temperature

Reducing intake temperature is one technique for increasing specific intake air volume.

In a Formula One engine, the injector is located in the upper part of the intake pipe inside the ABX, and the fuel’s latent heat of vaporization is used to reduce intake temperature.

The injector height and angle are adjusted to maximize this effect.

In addition, a great distance from the inlet valve seat face enhances the temperature reduction benefit, but also leads to decreased engine response and drivability, so these considerations must be balanced.

It has been confirmed that intake temperature is higher than ambient temperature, and that the temperature difference between an air intake sensor located in the induction system and one in the nose is approximately 4°C. However, it was found that the actual intake temperature difference can be more than 4°C, depending on the peak brake power revolution as measured from drive-shaft torque in driving status. The reason for this can be speculated to be due to heating of the induction system by the increased temperature inside the engine cover.

As a countermeasure to this, engine control mapping was revised with the aim of enhancing reliability, while using the sensor in the nose as a reference point.

In addition, an air inlet was added to the engine cover to allow fresh-air induction (Fig. 10), and circuit tests were thus conducted.

Because the engine cover has a high flow rate on the surface and a high internal pressure, a reverse flow of hot air from within the engine cover occurs if unmodified. It was not possible to take countermeasures in 2008. In the design of the 2009 vehicle, however, this item was incorporated from the start, and a design was chosen that suppressed this temperature increase inside the engine cover.

2.2. Inlet Port

Points to watch in the design of inlet ports include:

- Charging stroke
- Fuel distribution
- Suctorial dynamic effect

Charging stroke and suctorial dynamic effect impact the absolute air-flow rate, while fuel distribution and flow-velocity distribution impact combustion speed.

2.2.1. Reducing charging stroke

In cylinder head design, the following are done from the time of layout to reduce charging stroke.

- Reducing valve stem diameter
- Adjusting valve form
- Adjusting valve layout
- Adjusting form close to valve seat
- Smoothing-out component surface on inlet port
- Adjusting port form

The valve head shape contributes to reducing flow resistance.

The valve margin thickness and angle R close to the valve seat face are adjusted, and a thin form of valve margin thickness is used as the final form.

As the valve lifts, the effective area expands. If at this time the distance between the cylinder wall and valve face is short, the effective aperture between the valve and seat cannot be used efficiently. Therefore, the valve pitch was decided by taking into account the valve face’s position relative to the cylinder wall.

In addition, the exhaust valve face was set in a

Fig. 9 Picture of test ABX and inserts

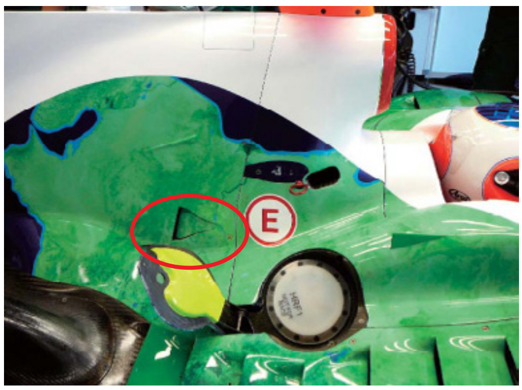


Fig. 10 Additional air inlet on engine cover

position so that it would not interfere with air inducted through the inlet.

The valve angle is one factor determining cylinder head size. For the 2005 Formula One engine, the valve angle was made small, and it was the most lightweight of all Honda Formula One V10 engines.

However, engines with small valve angles cannot get enough effective area relative to valve lift, which makes it difficult to increase power even if one does achieve a compact combustion chamber.

The engine in the first half of 2006 took the design of the 2005 engine into account, and the valve placement emulated that engine's as well, but power could not be increased for the above reasons, so the valve layout was changed mid-season.

Experiments were done with changing to a valve angle equivalent to that of 2004, but the ability to install to the chassis for racing was a necessary condition for making a layout change mid-season, so in fact, the finalized specification was intermediate between 2004 and 2005.

Because of homologation, which froze development of engines themselves after 2007, the valve layout was maintained as is.

A compound valve layout is a technique to achieve the compact combustion chambers needed to produce good combustion.

In a parallel valve layout, because port form and flow direction are arranged in relative conformity to each other, it requires no great effort to set a port form immediately over the seat. During the initial examination of the compound valve layout, the seat's upstream form was mapped together with the valves in parallel, and the effective area relative to the parallel valve port used as a base was reduced. However, because the actual flow showed no direct correlation to the valve stem (Fig. 11), the form was designed with no consideration for the valve stem, thereby minimizing the loss in effective area. As a result, power was increased by 4 - 6 kW, including the valvetrain design.

In designing the port form, one chooses a form that tries to control the separation that occurs on the side towards the lower face inside the port (a). As for the form of the upper face close to the seat, designers are conscious of overall air flow. Since the master stream of air flows in a different direction from that of the valve stem, the part directly over the seat (b) is not expanded (Fig. 12).

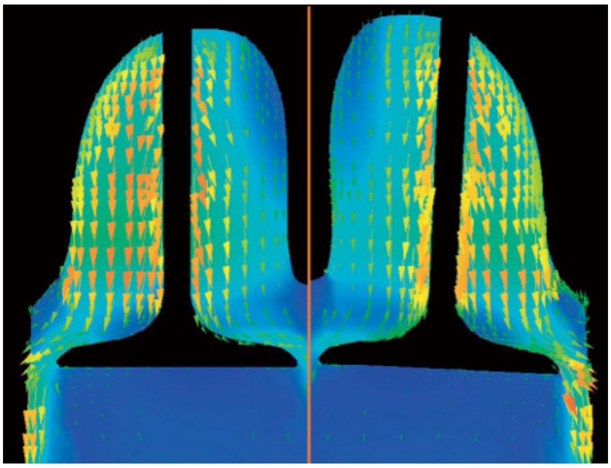


Fig. 11 Influence of shape around valve seat (Left : Parallel Valve Right : Compound valve)

2.2.2. Fuel distribution

In the early stages of engine development in 2000, the main focus was on reducing intake resistance. During development in 2004, a port to control separation from the inner face was examined as a technique to reduce intake resistance. Expansion of port volume was approved and ηv was expected to increase, but power actually went down. The discrepancy expected in the measured ηv could not be identified, and it was considered that perhaps some factor other than mass flow-rate was impacting power performance. It is considered that, although this specification controls separation, there is significant fuel adhering to the internal wall as a consequence. Figure 13 shows results of a check of fuel adhering to the port internal wall. Where the color is darker in the figure, fuel is adhering to the internal wall of the port. It can be confirmed that ports with reduced intake resistance have more fuel adherence.

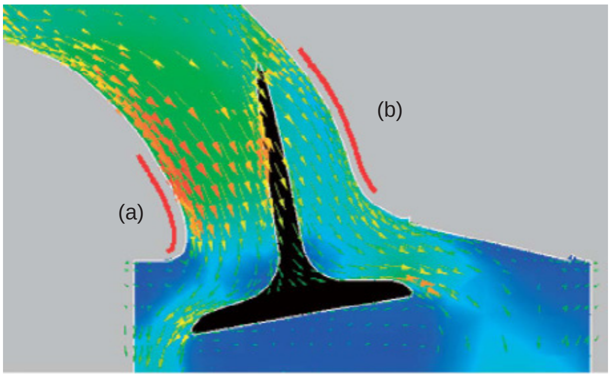


Fig. 12 CFD result of 2004 inlet port

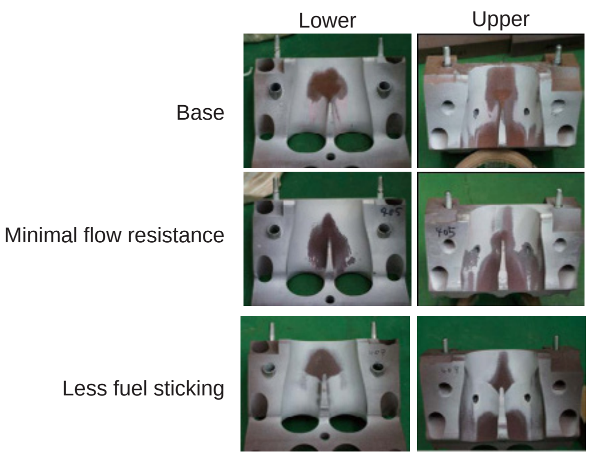


Fig. 13 Fuel sticking area with inlet port when conscious of flow (middle)

By controlling fuel adherence, a power increase of 2 kW was achieved. It was confirmed that besides reducing intake resistance, the volume of the inlet port and volume allocation in sectional area also affect power. It was simultaneously confirmed that the form of the port is also related to the form of gas mixture in the cylinder.

2.2.3. Dynamic effect

Following on the fact that ηv does not increase in enlarged ports with reduced flow resistance, in the middle of the 2004 season an investigation was initiated into forms that would narrow-down the inlet port within the range where flow resistance would not be decreased. This concept accounted for 3 - 6 kW of power gain and was used in the 2004 Japanese Grand Prix. Since then, Honda models up to 2008 were given similar distribution in sectional area.

Analysis testing with a single-cylinder engine was done to accelerate port development. However, in comparisons of Formula One engine port performance, multi-cylinder engines (V8 with intake interference) and single-cylinder engines (with no intake interference) have sometimes shown completely opposite trends (Fig. 14).

The cause of this is intake interference between cylinders in a V8 engine. Two approaches can be taken as countermeasures: controlling intake interference with the ABX; and optimizing ports in places where there is intake interference.

During development of the 2009 engine, the inlet port form was examined with the objective of promoting flow within cylinders (tumbling). Although there was a trend for the effective area to grow smaller, no difference was found in performance, and no great discrepancy was found in ηv at this time. These results could not be explained just from the perspective of reducing charging stroke.

2.2.4. Inlet port development techniques

When designing an engine, the platform is first decided, then the form of the ports included in this platform is designed, and finally the induction system is designed.

However, advances in 3D CAD and CFD have now made it possible to predict induction system performance with high precision at the design stage. It is therefore possible to design an optimized induction system at the earliest stages of development, and overall performance of the induction system as a whole as well as development efficiency have been enhanced.

To increase efficiency of inlet port development, the method of filling ports with adhesives and again applying CNC processing techniques was adopted in 2004. This offered the advantages of being able to produce optional parts in a short time so they could be put into use more quickly, and of being able to minimize the effect of individual differences on engine performance.

3. Exhaust System

3.1. Exhaust Port

During the study of single-cylinder engines, it was learned that flow rates from exhaust ports rose to nearly the speed of sound and caused choking. In steady flow tests, the flow velocity did not come close to the speed of sound due to equipment capacity, and thus choking was not recognized.

While in choke, flow depends on cross-sectional area and the state of the fluid, so performance tests were conducted after expanding the part with the smallest cross-sectional area. Some enhancement of performance was seen with the single-cylinder engine, but none was observed in the V8 engine. In a single-cylinder engine, choking occurs during the period of valve overlap from blowdown, but in the V8 engine, there is a range where pressure within the exhaust pipe is high compared to that of a single-cylinder engine because of the effect of inter-cylinder interference, so it is believed that a high pressure differential such as would cause choking does not exist, so the same effect could not be achieved.

The cause of this is that, at exhaust-pipe diameters that would be realistic on board a vehicle, it is not possible to produce a diffuser effect that would cause choking of the throat of the exhaust port of a V8 engine, as shown in Fig. 15. However, by using expanding pipe that brings flow velocity close to the speed of sound within the exhaust port, staying within the range of equipment that can be mounted on board and without expanding port diameter, performance could be enhanced at high engine speeds.

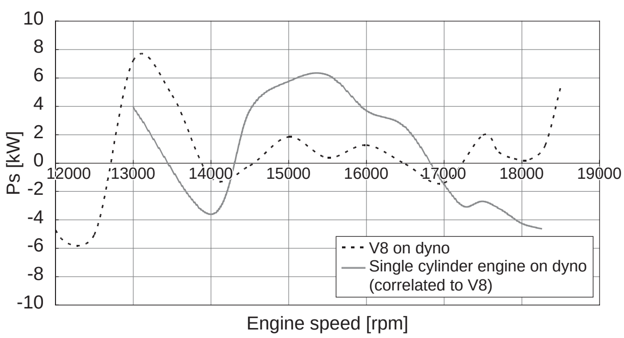


Fig. 14 ⊿Ps of inlet port on V8 and single cylinder engines

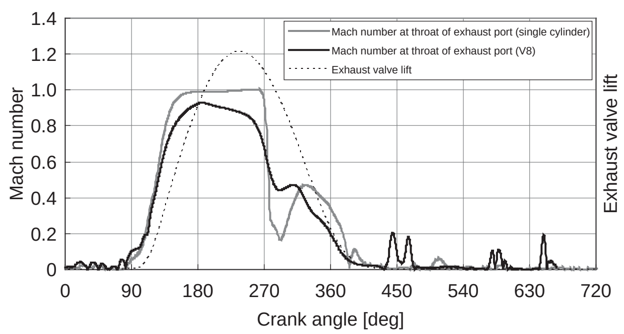


Fig. 15 Calculated mass flow of exhaust port

3.2. Exhaust System

3.2.1. System overview

The exhaust system of a Formula One engine consists of three parts: primaries, a collector and a tail. Figure 16 shows these in assembled form.

In exhaust systems used for testing on a dyno, the three components are each configured separately, which takes account of the need to change components because of damage, and the need for tuning between the collector and tail. In exhausts for actual vehicle testing and racing use, however, the collector and tail are welded together and are used as a single piece. Although making the three components into a single piece would make the exhaust lighter in weight, the primaries are kept separate to enable spark plug maintenance, during which the cylinder head covers are removed. The material used was Inconel 625, with pipe thickness of 0.7 mm.

In the world of Formula One, which is under no emissions regulations, the exhaust system is designed for high power. Broadly speaking, there are two points to focus on: reducing exhaust loss and using the dynamic effect of exhaust pulsation.

To reduce exhaust loss, pressure loss has been consistently reduced by enlarging the pipe bending-angle and bending-radius within the limits allowed for the layout under the vehicle cover (cowl).

On the other hand, the primaries are gathered together to utilize exhaust interference. As a result, within the collector, exhaust pressure causes reflected waves to form from the open edge. These reflected waves have an effect on internal pressure in the cylinders during valve overlap, and the amount of residual gas can be decreased. This leads to an increase in specific intake air volume as a result, making it possible to alter power characteristics in relation to engine speed. This uses the dynamic effect of exhaust pulsation. With the reflected waves in the collector alone, however, the range of engine speeds in which pulsation can be used is limited, so stepped pipes (steps) were implemented in the primaries (Fig. 17).

At 17500 rpm, a gain of 4 - 8 kW was realized as compared to an exhaust without steps.

While Honda was competing in Formula One during the third era, a forward exhaust system was used in 2007 only (Fig. 18), while a backward exhaust system was used in all other years (Fig. 19).

In order to enhance car aerodynamic performance to compensate for the engine power that decreased with the change in regulations that stipulated V8 engine use, it was necessary to increase the degree of freedom of aerodynamics design. A forward exhaust system has less negative impact on engine power than a backward exhaust system, and allows a greater degree of freedom in the aerodynamics design of the rear of the engine.

However, all the high-temperature parts were contained within the engine cowl, and the exhaust outlet was placed in a position that did not promote ventilation from the outlet, so there was frequently heat damage, including to vehicle parts. For that reason, Honda returned to the backward exhaust system in 2008.

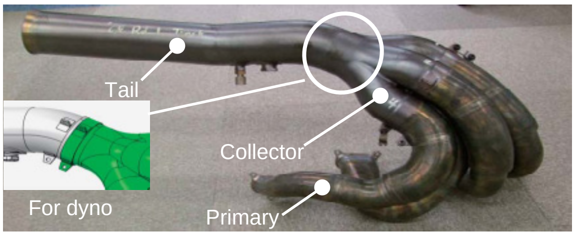


Fig. 16 F1 exhaust system (right-hand side for car)

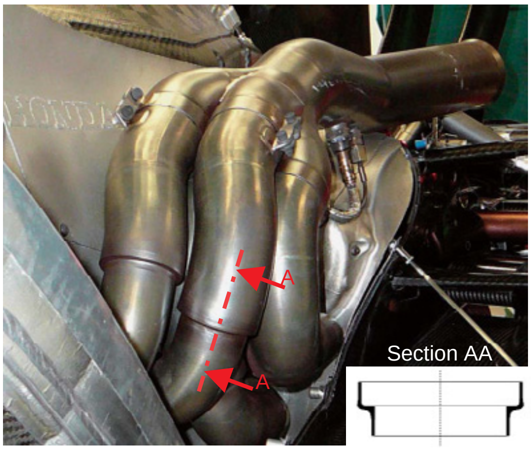


Fig. 17 Stepped primaries

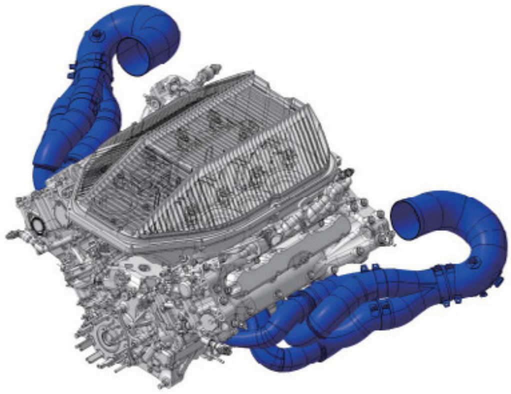


Fig. 18 Forward exhaust system

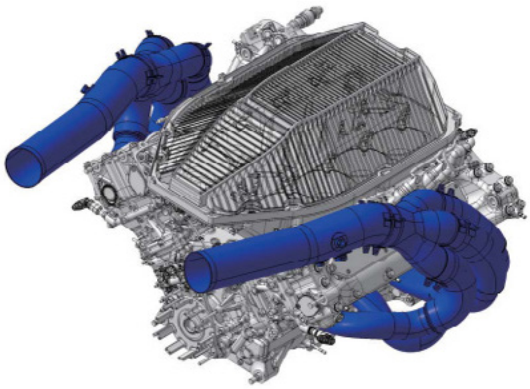


Fig. 19 Backward exhaust system

3.2.2. Compact and lightweight technology

The exhaust system has a major impact on a racing car's vehicle dynamics in terms of component size and weight. Therefore, the technology of packaging the components themselves was always being advanced.

Because an exhaust system is made by bending cylindrical pipes, the primaries and collectors require space within the cowl and have an impact on aerodynamics.

With the exhaust system with non-circular sections (compact exhaust) developed in 2008, the aim was to achieve a space-saving, lightweight exhaust system that retained engine power, durability and reliability, rather than deciding on an exhaust form in a way that depends on the aerodynamics concept based on the cowl.

For the primaries, a non-circular section structure, three-dimensional bend layout and ellipsoidal stepped pipes were used and design was conducted that took account of heat damage to the engine itself. The production method proposal ensured molding accuracy by pressworking that took advantage of the know-how of an exhaust system manufacturer.

To make the collector lighter, it was considered to use a shared-pipe outer wall where the collectors gathered together, and to use a long collector in which a part of the primary would be taken into the collector and sharing of the wall further increased, and components were thus produced.

When sharing the pipe outer wall, if using pressworking, which is the conventional way, more dies are needed and material yield declines, which raises production costs, so a new precision casting technology was used. The result was a form with wall thickness of 0.7 mm (Fig. 20).

To deal with the increased thermal load that comes from sharing the pipe outer wall, René 41 was used, which is suitable for casting and has greater high-temperature fatigue strength than conventional materials. Results showed that in dyno durability tests, component life was extended about 60% over that of ordinary Inconel 625 collectors.

An image of the entire compact exhaust system is shown in Fig. 21.

These specifications allowed the cowl line to draw 50 mm closer to the engine than before. However, in order to make effective use of limited space while minimizing the impact of using a non-circular section, a layout was chosen that extended primaries to a length greater than standard specifications and wrapped them around to the engine front.

Figure 22 shows results of checking engine power under these specifications. This shows the compact exhaust power gain under the condition that the primaries have been extended. This is for a single bank, but there was an increase in power of 4 kW on average from 7500 rpm to 11500 rpm. The factor that increased power is believed to be that, since the volume of the collector gather was approximately 34% less than the conventional specifications, there were changes in exhaust pulsation in a certain engine speed range, which caused engine power characteristics to change.

3.2.3. Torque boosting technique

During races, engine speeds of 16000 rpm and higher are used about 90% of the time. However, torque characteristics at lower engine speeds are critical at the start of the race and when the car is accelerating from the apex of the curve, and such characteristics affect lap times and race results.

For that reason, the exhaust system is developed not only for power at high engine speeds but also for torque characteristics at low engine speeds. Specifically, the form of the internal wall of the collectors has undergone optimization since 2005.

Figure 23 shows a cross-sectional view of collectors with internal walls.

The form of the internal wall in a standard collector is decided by the collection angle of the pipes as well as the pipe diameter, and the wall edge forms a curve. If left unmodified, the cross-sectional area of the pipe would remain unchanged up to the gather as it went to the open edge.

If the wall height were intentionally raised and the cross-sectional area restricted, there would be no

Fig. 20 Casting collector

Fig. 21 Compact exhaust system

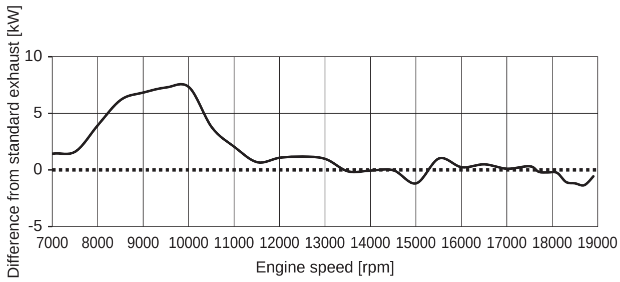


Fig. 22 Performance of compact exhaust system

decrease in power at high engine speeds as a result of increased pressure loss, and the phase of reflected waves would be delayed. This is thought to be caused by the fact that the distance to the open edge was increased and exhaust gases were kept below supersonic speed.

As a result, exhaust pulsation characteristics change at low engine speeds and combustion gas flow inside the cylinders is impacted, causing changes in torque characteristics. Starting in 2005, the optimization of the form of the internal wall of the collectors was constantly pursued as one means of enhancing torque characteristics at low engine speeds.

Since 2008, the regulations have prohibited the use of traction control systems, making it even more necessary to enhance torque characteristics from the point of view of drivability.

However, there are areas where optimization of wall form alone is not sufficient compensation, and there is also the effect of inconsistent combustion, so the issue of drivability was not completely resolved.

An effective way to enhance drivability is to flatten engine torque at partial throttle (i.e., engine torque when the throttle is neither fully closed nor fully open) at low engine speeds.

To increase development efficiency, a test exhaust system with variable pipe diameters and lengths was used (Fig. 24) as well as simulation to study form, thereby verifying the drivability enhancement effect. This part discusses two results that demonstrate the effectiveness of development.

The first was an exhaust system with connecting pipes (i.e., balance pipes) (Fig. 25). The goal was to cause changes in exhaust pulsation characteristics with these connecting pipes. Simulation was used to select connecting pipe width, length and connecting points, and specifications were selected that enabled residual gas to be reduced.

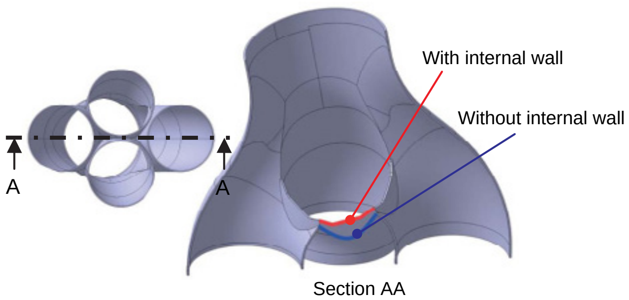


Fig. 23 Cross-sectional view of collector

Fig. 24 Test exhaust system

The second was the 4-2-1 exhaust system. The basic stance was to alter exhaust pulsation characteristics just as for the balance pipes mentioned above. Specifications were selected while confirming the power with the test exhaust system. Ultimately, the primary length was made 50 mm longer than in a 4-1 exhaust system, with 360° assemblies of #1-#4, #2-#3, #5-#8 and #6-#7 (i.e., connecting cylinders with the ignition phase offset 360°).

Power check results are shown in Fig. 26.

The balance pipes yielded a power gain of 12 kW at engine speeds of 8500 rpm and 10500 rpm, but a drop of 2.5 kW at 17000 rpm.

The 4-2-1 exhaust system increased power by an average of 8 kW at engine speeds from 8500 rpm to 10500 rpm and yielded about the same results as the base exhaust (4-1 exhaust system) at 17000 rpm and up.

Figure 27 compares results of engine torque at partial throttle.

Both the balance pipes and 4-2-1 exhaust system showed torque flattening and superior performance as compared to the base exhaust.

Fig. 25 Balance pipes

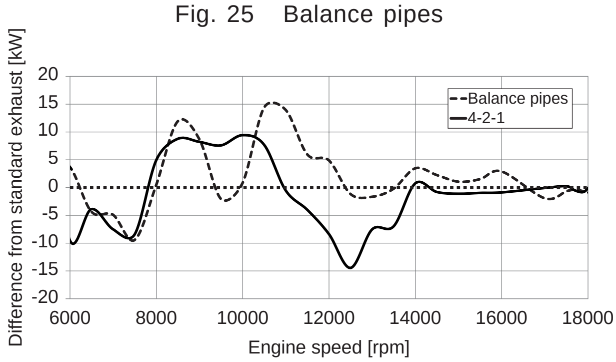


Fig. 26 Performance of balance pipes and 4-2-1

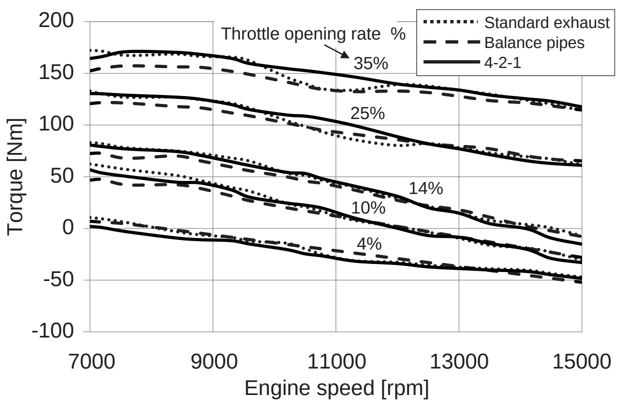


Fig. 27 Torque characteristics at partial throttle

However, drivers were not able to experience the superiority of the balanced pipes in circuit tests. Also, the connecting pipes caused the weight to increase, and there were still issues with durability and reliability because of scattered cracks in the weld to the base pipe.

It was estimated that the weight increase would be approximately 220 g over that of the base exhaust.

4. Conclusion

Through engine development during Honda's third-era Formula One activities, we have learned the following about induction and exhaust systems.

(1) The importance of induction and exhaust system design that is mindful of the vehicle package became apparent once again, and development techniques were created that extract maximum performance from Formula One cars.

(2) Even with racing engines, it is necessary to be aware of torque characteristics at low and medium speeds, and is important to design exhaust systems as a technique for their enhancement.

(3) Induction system development techniques were created that use CFD and can predict dynamic characteristics from the design stage. If it is possible to predict the dynamic characteristics of exhaust systems in the future, this will enhance development efficiency even more.

(4) Test part-production techniques were implemented to suit a short development cycle, and the time required to optimize power characteristics and determine specifications was shortened.

References

(1) Nakamura, S., Motohashi, Y., Hayakawa, S.: Development of Wind Simulator Equipment for Analysis of Intake Phenomena in Formula One Engines, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), P. 89-94

(2) Hanada, N., Hiraide, A., Takahashi, M.: CFD Technology for Formula One Engine, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), P. 82-88

■ Author ■


Ken NISHIMORI


Yasuhiro MOTOHASHI


Masayoshi TAKAHASHI


Ryuichi FURUKAWA

CFD Technology for Formula One Engine

Naoki HANADA*　Atsushi HIRAIDE*　Masayoshi TAKAHASHI*

ABSTRACT

Simulation technology has advanced markedly in recent years, and various types of CFD models have come into use for Formula One engine development.

However, to use such simulation modeling it is necessary to establish simulation technology for the unique conditions for Formula One engines, such as that for high engine speed.

The pressure of each engine part was measured and in-cylinder gas motion and fuel spraying behavior were also measured using a single-cylinder optical engine, enabling CFD technology that can be applied to Formula One engines to be created.

1. Introduction

Simulation technology has advanced markedly in recent years. Even in many fields for Formula One, simulation technology has been established and has come into use for development. CFD requires extensive computer resources and has become more practical as hardware and computing technology have advanced.

CFD modeling uses a variety of sub-models such as turbulence models, and it is crucial to validating whether the phenomenon being calculated can be expressed. For example, in the case of automobile reciprocating engines, which use a wide range of operating speeds, the pressure fluctuation within the intake and exhaust pipes will differ at low engine speeds and high engine speeds. A calculation technique validated for just low engine speeds or just high engine speeds would not necessarily be compatible with the other. Dependable validation of simulations helps make it possible to create CFD technology that is effective for development.

CFD technology for commercial engines[(1)-(5)] has been developed by members of a special team. At first, since the Formula One engine is also a reciprocating engine, it was thought that the CFD technology should also be compatible with the Formula One engine as is, and the attempt was made to apply the technology to Formula One engine development. When actual simulation was performed, however, it became clear that the technology could not express Formula One engine phenomena. Since commercial engines are normally used at low engine speeds, low engine speeds of about 2000 rpm are an important analysis condition of simulations too. Since the highest priority of a Formula One engine is output, it is necessary to conduct evaluation when the engine speeds are high, as high as 20000 rpm. At high engine speeds, several factors affect output: increased piston speed and gas flow rate, fuel spray behavior and cylinder interference in the intake and exhaust systems resulting from pressure fluctuations within the intake and exhaust pipes, the motion of gas coming into the cylinders, and so on. We began work on developing a simulation model that could express these impact parameters.

Using data from pressure measurements for each part and in-cylinder measurements taken with an optical engine, a simulation model usable for the development of Formula One engines was validated.

This paper explains CFD technology for the flow of air during intake and exhaust, in-cylinder gas motion, in-cylinder fuel behavior, and combustion.

2. Intake and Exhaust Systems

The intake and exhaust systems of a reciprocating engine impact volumetric efficiency and as such are important components determining engine output.

Being able to analyze pressure and gas flow of intake and exhaust systems and predict performance in advance provide effective ways of determining specifications in a short period of time when developing Formula One engines, where enhancements are expected on a daily basis. To achieve such predictions of performance, it is necessary to perform simulation studies in advance, and the results should be cross-checked with the measurement data.

Intake and exhaust system simulations were originally performed using one-dimensional gas dynamics and engine performance simulation software WAVE made by Ricardo Software. Subsequently, for the intake system,

* Automobile R&D Center

an in-house software capable of three-dimensional compressible fluid analysis was used, with the aim of enhancing calculation accuracy. For the exhaust system, usable three-dimensional analysis software could not be found and so we continued to use WAVE.

2.1. Intake System

The current Formula One engine intake system is equipped with an airbox, and the pressure fluctuation (pulsation) that occurs within the intake pipe of each cylinder propagates and reflects inside the airbox and is transmitted to other cylinders. As a result, airbox shape is one factor affecting engine output.

On the other hand, the body's aerodynamic performance greatly affects lap time. Therefore, the shape of the body cowl is determined by aerodynamic characteristics. As for airbox shape, the degree of freedom is restricted by the shape of the cowl.

Using WAVE, at first, performance was evaluated with the air intake form only, but in order to achieve both body aerodynamic performance and engine output, a large amount of time was spent during development on selecting the airbox shape. We tackled this issue so that simulations could be utilized for performance evaluation and phenomenon analysis including also the influence of airbox shape.

The simulation software used was Honda's three-dimensional compressible fluid analysis software[(1)] in which a one-dimensional model is coupled.

The simulation model is shown in Fig. 1. The airbox portion was expressed in three-dimensional form, and the portion from the downstream part of the air intake throttle to the cylinder and exhaust pipe was expressed by one-dimensional and scalar models.

The greatest feature of Honda's software is that it solves compressible fluids by not only one-dimensional but also three-dimensional calculation.

Although it is possible to do similar calculations by combining Ricardo Software's three-dimensional fluid analysis software (VECTIS) with their one-dimensional gas dynamics simulation software (WAVE), these software programs did not satisfy our target standards of three-dimensional compressible fluid calculation accuracy.

Based on the measured intake pipe pressure, boundary conditions, mesh density, flow coefficient, filter model and so on were investigated and a series of enhancements were made until a simulation that could be used for the analysis of intake interference in the airbox of a Formula One engine was created.

Figure 2 shows measurement and calculation results of pressure within intake pipes in each of the cylinders in one bank of a V10 engine. Engine speed was 17000 rpm, and the crank angle of 360 deg is top dead center of intake stroke. Intake interference within the airbox causes the different intake pressure in all the cylinders. While the amplitude and phase of pressure varies with

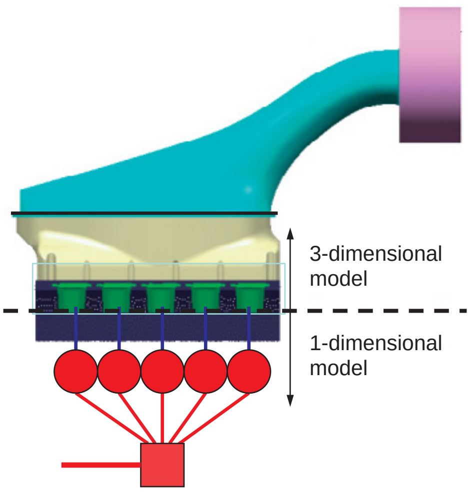


Fig. 1 Simulation model

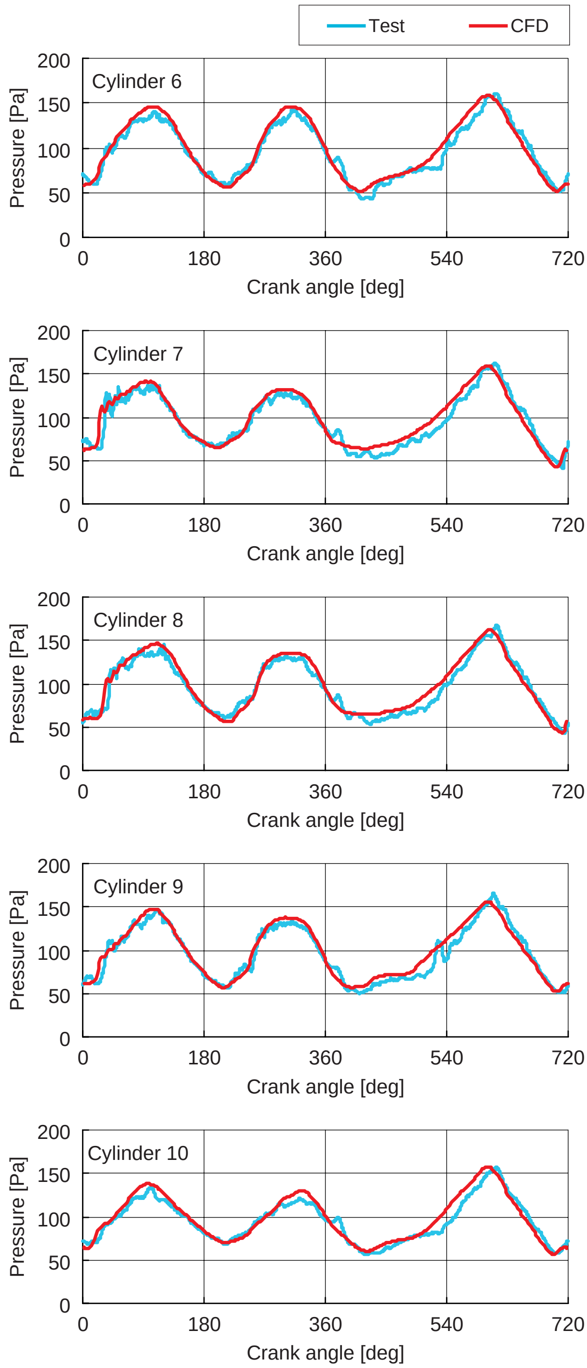


Fig. 2 Intake pressure of V10 engine at 17000 rpm

Table 1 Parameters for WAVE simulation

Parameter	Old value	New value
Flow coefficient at pipe end (inflow)	Auto	0.8
Flow coefficient at pipe end (outflow)	Auto	0.5
Ambient temperature of exhaust side	Atmospheric temperature	400 K
Multiplier of heat transfer of cylinder wall when intake valves are closed	1.0	2.0

each cylinder, there is a particular difference between cylinder 10 and the other cylinders. It can be seen that intake interference causes the volumetric efficiency of each cylinder to change and has a direct impact on engine output.

The calculation results reproduce these characteristics with good accuracy, demonstrating that Honda's software is capable of analyzing intake interference within the Formula One engine's airbox.

The slim and curving airbox shape required for good body aerodynamic performance is due to reduced volume and is characterized by a weak attenuation of pressure fluctuation and lowered volumetric efficiency. However, because practical simulation has been achieved, it is now possible to satisfy both a slim airbox shape which can contribute to body aerodynamics and engine output enhancement by optimizing air flow in the airbox.

2.2. Exhaust System

WAVE was used to analyze the exhaust system and examine its shape.

This explanation uses a V10 engine as an example. A 5-into-1 collector exhaust system, as shown in Fig. 3, was used. The WAVE model is shown in Fig. 4. Because the effect of intake interference in the airbox cannot be expressed in one-dimensional simulation WAVE, a model used for the intake side was designed such that intake pipes are open to the air, with each cylinder being independent. Because the left and right exhaust systems are independent, the WAVE model used one bank with five cylinders.

Figure 5 shows V10 engine validation results.

Exhaust pressure was measured at a downstream position approximately 40 mm from the exhaust valve seat of cylinder 10. Because the calculation model has five cylinders, the data used was of cylinder 5. The initial calculation results (b) differed from the measured results (a).

A parameter study to revise the variables used for simulation shown in Table 1 yielded the results shown

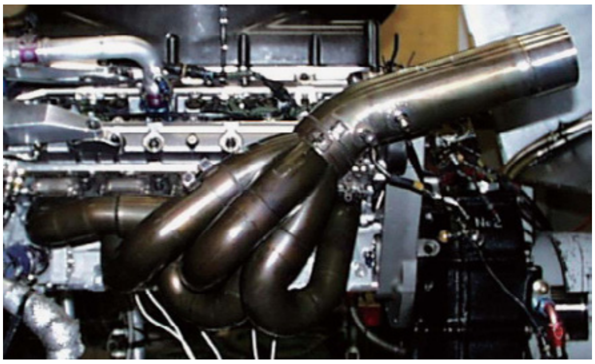


Fig. 3 Exhaust pipe of V10 engine

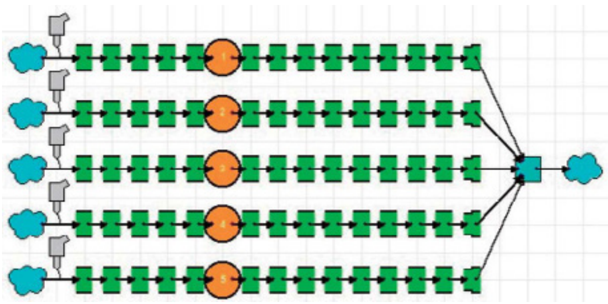


Fig. 4 WAVE model of 5 cylinders for V10 engine

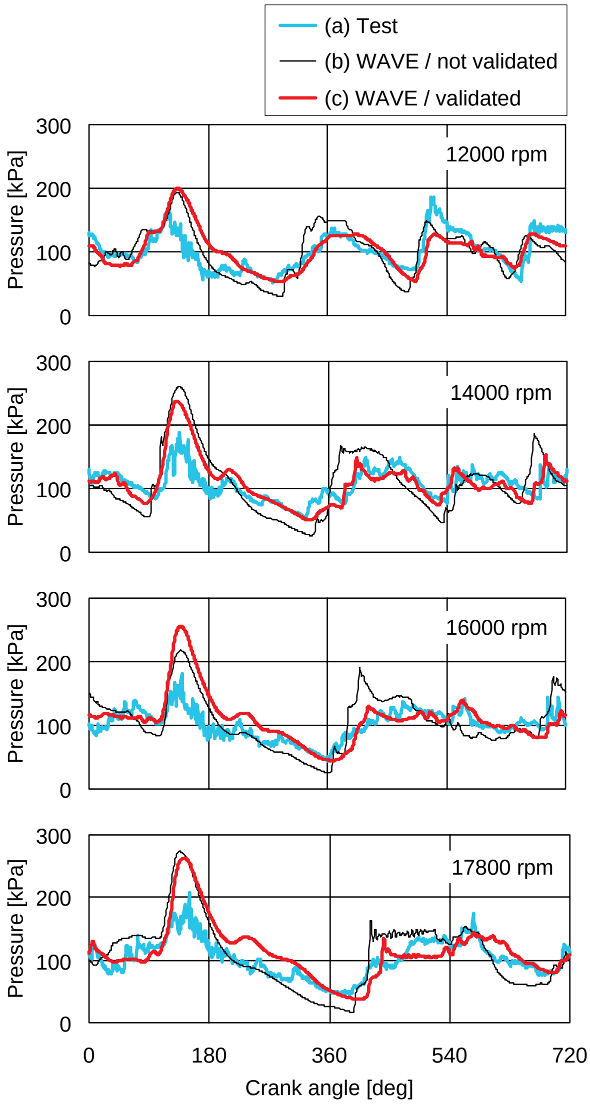


Fig. 5 Exhaust pressure of V10 engine (#10 cylinder)

in (c) in Fig. 5, with enhanced calculation accuracy. The important revised parameter is the flow coefficient for the end of the exhaust pipe. In the WAVE calculation, exhaust pressure changes greatly depending on the value of the flow coefficient, and ultimately volumetric efficiency and output values also change. Estimating the flow coefficient correctly is clearly the most important aspect of one-dimensional simulation.

In addition, this result shows that there are limits to analyzing exhaust systems with one-dimensional simulation. One-dimensional analysis has many subjects to be addressed such as followings. Naturally, it is not possible to estimate pressure loss in each part with one-dimensional analysis. General usefulness of flow coefficient is low for changes in the form of the exhaust pipe or changes in flow velocity. To some extent it is possible to analyze and examine flow within exhaust pipes with one-dimensional simulation, so this can be used in the development of exhaust systems. However, the calculation accuracy is not necessarily sufficient; for example, revision of the flow coefficient is inevitable when engine specifications change. Development of three-dimensional simulation, as used with the intake system, is needed in the future as a way to resolve this issue.

3. In-cylinder Behavior

Enhancing the combustion status is indispensable for achieving engine performance enhancements. Besides that, the current Formula One engine has longer combustion duration than commercial engines because of its high engine speed and big bore. Therefore, shortening the combustion duration is necessary.

To shorten combustion, it was considered necessary to adjust the in-cylinder gas motion and fuel distribution. To do this, it is indispensable to be able to predict in-cylinder gas motion and simulate fuel spray and mixture distribution formation.

3.1. In-cylinder Gas Motion

Validation on in-cylinder gas motion has been performed on commercial engines and an attempt has been made to apply the results to Formula One engines, but no correlation was found to actual engines. Thus we started over again and redid the process from validation of the Formula One engine's in-cylinder motion.

Figure 6 gives an illustration of the system for measuring in-cylinder motion. An optical engine[6] was used. Two high-repetition-rate YAG lasers were used with a YAG 532 nm second harmonic wavelength. The camera used was Vision Research's Phantom V7, and resolution was 656 × 328 pixels. Hollow resin tracers were used, with particle size of ϕ 40 μm and density of 36 kg/m^3. Based on tracer size and density, it is thought that air flow traceability will have a 95% attenuation rate for fluctuations of approximately 300 Hz. The measurement data took the mean for 100 cycles.

The simulation software was three-dimensional fluid analysis software (VECTIS). Figure 7 shows the simulation model. To correctly express intake and exhaust pulsation, the calculation covered the entire single cylinder engine from air inlet chamber to exhaust pipe.

Table 2 Calculation conditions of in-cylinder gas motion

Calculation region	Whole engine
Number of calculation cycle	4 cycles
Mesh density (for combustion chamber)	1st to 3rd cycle: 2.0 mm 4th cycle: 1.5 mm (Half the above size is used near the wall.)

The calculation conditions are shown in Table 2. These conditions were selected to enable calculation results to reproduce PIV measurement results and at the same time cause intake pulsation, volumetric efficiency and in-cylinder turbulence energy to converge. The reason that mesh density is changed with the number of cycles is to ensure both calculation accuracy and reduced calculating time. Except for turbulence intensity, the calculation results showed no differences between 2 mm and 1.5 mm of mesh densities, but for turbulence energy, changes in cycles could not be expressed at mesh density of 2 mm. For that reason, calculating time was saved by using a coarser mesh to calculate the first three cycles, and then a finer mesh was used on the fourth cycle to ensure calculating accuracy.

Figure 8 shows visualization and simulation results in a motored condition at 10000 rpm. The crank angle shown in the figure is the angle after top dead center. Measurements were performed through a cross-section

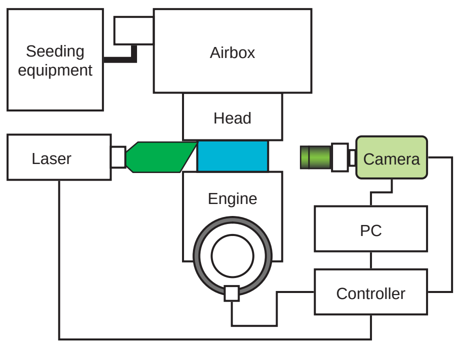


Fig. 6 Schematic of PIV measurement system with optical single cylinder engine

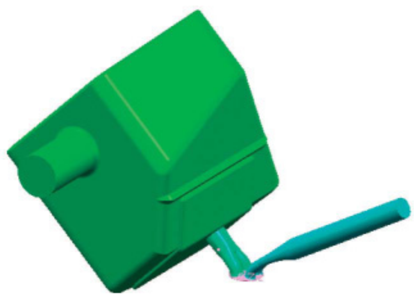


Fig. 7 Simulation model of in-cylinder gas motion

under the intake valve, as shown in the upper right of the figure. Two intake pipe lengths of 200 mm and 150 mm were used to compare differences in gas motions. The left column is from the 200 mm intake pipe, and the right column is from the 150 mm intake pipe.

In-cylinder motion in a Formula One engine consists of two vortexes, as demonstrated by the example of crank angle 216 deg ATDC: a large vortex centered on the bore, and a vortex that forms on the bottom of the intake port. The images show that the twin vortex configuration does not change with the difference in intake pipe length, but the gas inlet velocity, vortex center and vortex speed are affected by intake pipe length. This means, in other words, that intake pulsation has an impact on not only volumetric efficiency but also combustion.

Calculation results showed a flow pattern that closely matches visualization results. This demonstrates that if a model shape and calculation conditions are chosen such that intake and exhaust pulsation can be expressed, it is possible to accurately simulate in-cylinder gas motion at the high speed of 10000 rpm.

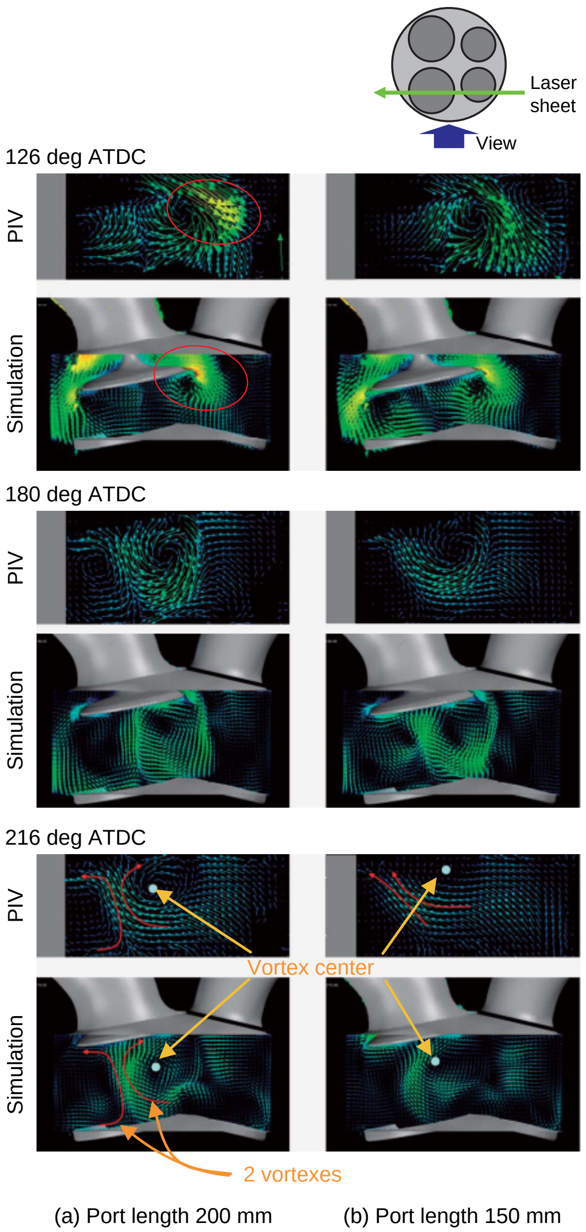


Fig. 8 Comparison of flow vectors between PIV measurement and simulation results at 10000 rpm

These initiatives have made it possible to predict gas motion in a Formula One super-high-speed engine. In addition, this research has renewed the researchers' awareness of the importance of measurement that helps one to grasp phenomena when creating simulation technology. Without PIV measurement results, what to set as the number of cycles and mesh density cannot be determined.

The fact that turbulence intensity is not validated is an issue, and analysis of this, including its impact on combustion, should be performed.

3.2. In-cylinder Fuel Behavior

The distribution of fuel within the cylinder is one factor affecting the quality of combustion, and it is important to predict this in advance with simulation. However, there are issues with measuring fuel distribution within cylinders, and measurements are not easy. For that reason, analysis by simulation is valuable in terms of advancing phenomenon analysis.

Injectors of Formula One engine are typically installed close to the end face of the trumpet. The reason is because volumetric efficiency is increased as a result of charge cooling by evaporation latent heat of the fuel. Because of this installation, fuel injected toward the interior of the intake port is affected by gas motion which fluctuates because of intake pulsation, and then it enters the cylinder. Therefore, to calculate in-cylinder fuel distribution, the necessary calculations must include fuel behavior inside the intake port.

Simulation validation was performed with the behavior of the fuel droplets flowing into the cylinder which was photographed with an optical engine. Using two kinds of injectors, it was evaluated whether the simulation can express the difference produced from the difference in the fuel spray characteristic. One was a pintle form with fuel pressure of 1.2 MPa, and the second was a six-hole plate form with fuel pressure of 10 MPa. These were characterized by respective Sauter's mean diameters of approximately 40 μm and 16 μm, respectively.

Using an optical engine[6], spray droplets were filmed directly with a strobe as a light source. Operating conditions were: motored at 10000 rpm, and wide open throttle.

For simulation, VECTIS was used, the same as for in-cylinder motion calculation. An entire single cylinder engine was modeled and a fuel spray calculation was added. To calculate fuel spray, the following method was used. The state of spraying with the injector alone was matched with measurement results in advance. Then, the fuel spray calculation conditions were input into the single cylinder engine calculation.

The fuel spray form of the injector alone is shown in Fig. 9.

This shows that with both the 1.2 MPa and 10 MPa injectors, the calculations were able to sufficiently express the actual spray form.

It was learned that for in-cylinder mixture forms to converge in calculations of fuel spray in the engine, about five cycles are necessary. Figure 10 shows the cycle history of in-cylinder average A/F and A/F variance when calculating sprays in an engine using a 10 MPa injector. Fuel injected from the trumpet end face begins to enter the cylinder on the second cycle after injection, but the amount is small. This figure can be read as showing in-cylinder fuel behavior converging from the fourth to fifth cycle. To give some margin, it was decided to conduct the analysis with five cycles.

Figure 11 shows the results of photographing in-cylinder spraying and the calculation results.

The measurement results are colored to represent the degree of photographed brightness. In the calculation results, spray droplets are represented by dots, and the broken line shows the approximate area in which the droplets occur.

With the pintle, the fuel spray has large particles and the spray has great inertia. That is why, as the photograph taken at crank angle 120 deg shows, the spray entering the cylinder from the intake valve passes below the exhaust valve and collides with the cylinder wall. In contrast, the high pressure injector spray has little inertia, and after it enters the cylinder, it rides the flow of air down the cylinder toward the piston. In addition, the spray entering from the intake valve enters not only from the exhaust side but also from the port bottom side that is opposite it.

To make the fuel pass through the entire combustion chamber, it would be effective to reduce particle size and inertia.

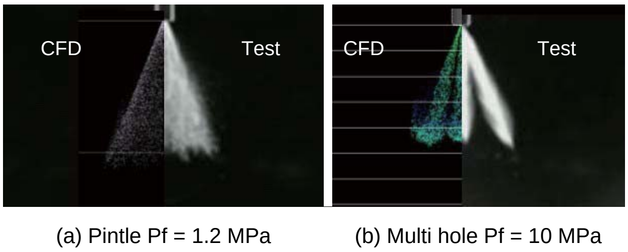


Fig. 9 Fuel spray form

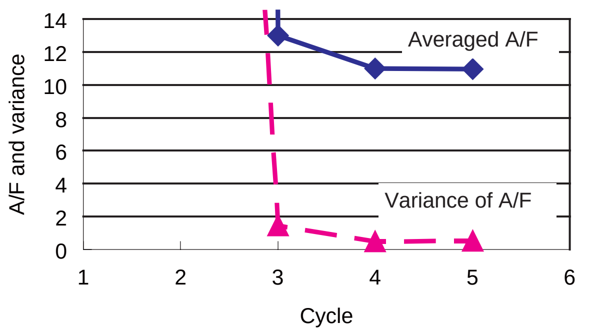


Fig. 10 Simulation convergence of averaged A/F and variance of A/F in cylinder

The simulation largely expressed the measurement results, and by matching the spray with the injector alone beforehand, it was possible to predict in-cylinder fuel behavior. In addition, the fact that gas motion inside both the intake port and the combustion chamber was correctly solved is one factor for accurate calculation.

3.3. Combustion

To directly evaluate the influence of gas motion and fuel behavior on combustion, and to predict amount of residual gas, efforts were focused on combustion simulation as the next step.

The software used was VECTIS, but the combustion simulation module(4) made by Honda was incorporated into VECTIS.

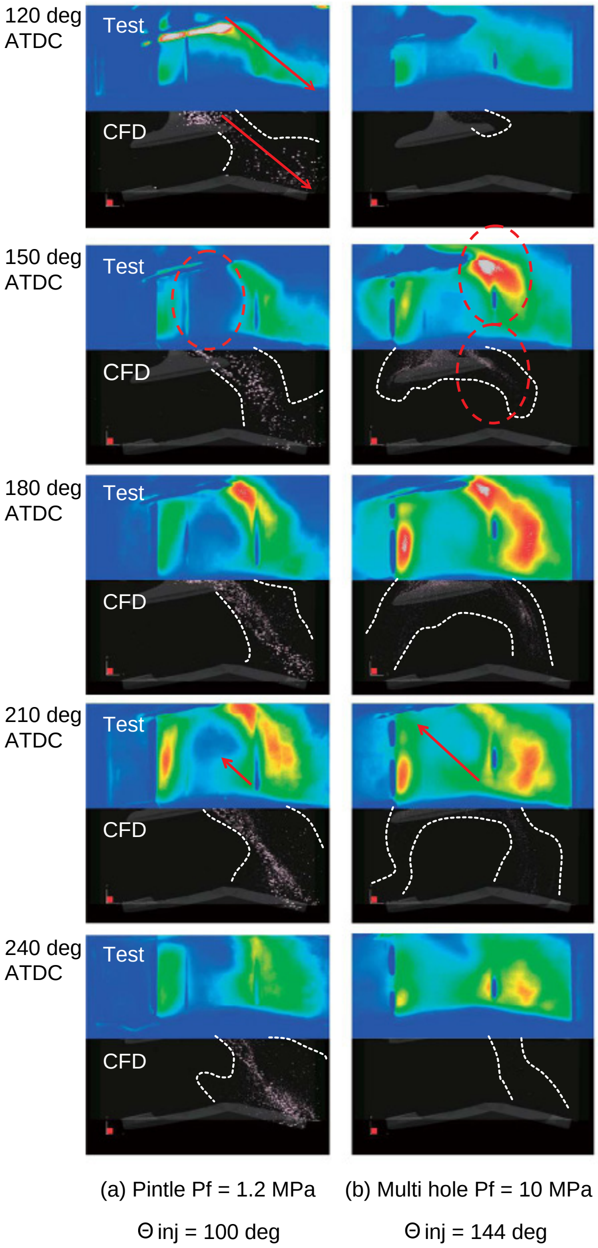


Fig. 11 Comparison of fuel droplets in cylinder between measurement with optical engine and simulation with VECTIS at 10000 rpm

The subject of the simulation evaluation was not the Formula One engine itself, but rather a single cylinder engine with the same specifications as the Formula One engine. The calculation model was the same as the in-cylinder motion and fuel spray model shown in Fig. 7.

As a calculation procedure, at first, in-cylinder motion and fuel behavior in a motored condition were calculated up to the point just before ignition on the fifth cycle for convergences of gas motion and mixture concentration. After that, combustion calculation was performed only by a model of the combustion chamber except intake and exhaust elements.

In-cylinder pressure measurements and calculation results for a single cylinder engine at 15000 rpm and 17750 rpm are shown in Fig. 12. In the combustion simulation used, it is necessary to tune a constant that adjusts combustion speed, but the simulation was able to reproduce in-cylinder pressures at different engine speeds with the same constant. However, this is a different value from the constant used for commercial engines and it was confirmed that it is necessary to tune a unique value for Formula One engines. The outlook is that Honda's own combustion simulation can be applied to Formula One engines as well, and the research is at the stage of confirming whether it has general usefulness for an even greater number of operating conditions.

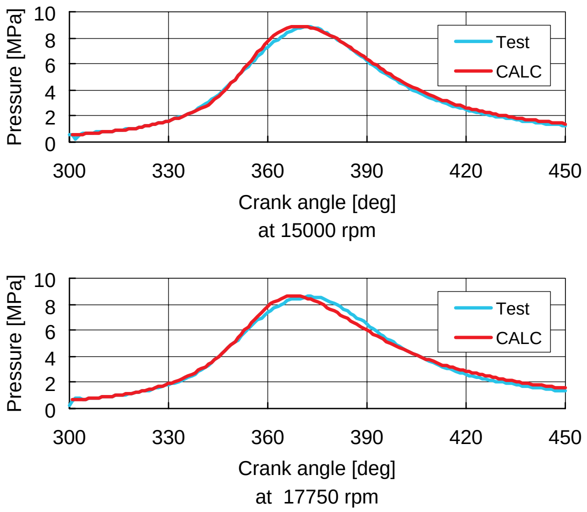


Fig. 12 Comparison of cylinder pressure between measurement and simulation

4. Conclusion

Creating simulation models has been attempted to allow development of Formula One engines with more performance more quickly.

Simulations which use various models cannot demonstrate their ability unless reliable validation is conducted for each subject of calculation.

In the future, we hope to use the model validated and created with Formula One to develop CFD technology for engines that will be friendlier to the global environment.

References

(1) Takabayashi, T., Hotozuka, Y., Tsushima, H.: The Three-dimensional Pulsation Flow Simulation and Its Application for Engine Intake and Exhaust Systems, 2000 FISITA World Automotive Congress, F2000A091 (2000)

(2) Ishikawa, N., Hiraide, A., Takabayashi, T.: Air/Fuel Distribution Simulation in a Port-Injected Gasoline Lean-Burn Engine, SAE Paper, 2001-01-1230 (2001)

(3) Zhu, G., Reitz, R. D., Xin, J., Takabayashi, T.: Characteristics of Vaporizing Continuous Multi-Component Fuel Sprays in a Port Fuel Injection Gasoline Engine, SAE Paper, 2001-01-1231 (2001)

(4) Yang, X., Ohashi, T., Takabayashi, T., Kubota, S., Katsuyama, H., Urata, Y.: Ignition and Combustion Modeling with G-Equation in Spark Ignition Engines, The 13th International Pacific Conference on Automotive Engineering (2005)

(5) Ohashi, T., Yang, X., Takabayashi, T., Urata, Y., Kubota, S., Katsuyama, H.: Ignition and Combustion Simulation in HCCI Engines, SAE Paper, 2006-01-1522 (2006)

(6) Yanagisawa, N., Hanada, N., Hamakawa, T., Ogiyama, k., Goto, T.: Measurement Technologies for Formula One Engines, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 101-108

■ Author ■

Naoki HANADA

Atsushi HIRAIDE

Masayoshi TAKAHASHI

Combustion Diagnosis of Formula One Engine Using Micro-Cassegrain Sensor

Naoki HANADA* Atsushi HIRAIDE* Kohei YAMADA*
Yuji IKEDA** Atsushi NISHIYAMA**

ABSTRACT

Combustion diagnosis of a Formula One engine during wide open throttle (WOT) acceleration and deceleration operations was performed using a micro-Cassegrain system.

The air/fuel mixture (A/F) in each cylinder was measured, which helped in the development of controls to minimize the torque loss due to unstable combustion.

In transient conditions where acceleration and deceleration are performed repeatedly, the air/fuel mixture around the spark plugs becomes too rich or too lean, and results in unstable combustion. In addition, the air/fuel mixture formation is different inside each cylinder.

The fuel distribution to each cylinder needed to be controlled to a high degree of accuracy, and to do this, a highly responsive air/fuel ratio sensor (LAF sensor) is required.

1. Introduction

To increase the engine output, it is important to find a way to efficiently combust the air/fuel mixture. Combustion involves a series of processes, including forming the air/fuel mixture inside the cylinder, performing an effective ignition, promoting combustion with turbulence, and scavenging. Making the initial flame more stable by performing an effective ignition can minimize unstable combustion and lead to increased thermal efficiency.

Although analysis of the air/fuel mixture formation and flame propagation inside all the cylinders was also required, priority was given to analyzing the engine issue as described by the driver. The A/F around the spark plugs, which directly affects the ignition performance, was measured using a micro-Cassegrain sensor.

In Formula One engines, nearly all throttle operations are WOT or in fully closed positions. For this reason, fuel control is required that matches sudden changes in the amount of induction air, and if the fuel correction is inappropriate, unstable combustion will occur and the drivability will be reduced.

The flame will not be able to propagate quickly if the air/fuel mixture is too lean or too rich.

To enhance the drivability, fuel need be supplied to the cylinder so that the appropriate A/F is provided to each cycle.

As shown in Fig. 1, as the indicated mean effective pressure (IMEP) rises, the misfiring occurs in some cycles. This is because the appropriate A/F is not formed around the spark plugs, resulting in unstable combustion. It is important to analyze why the misfiring occurs and to take preventive measures.

Existing combustion diagnosis methods include measuring the induction pressure, in-cylinder pressure, exhaust pressure, and exhaust gas characteristics; and

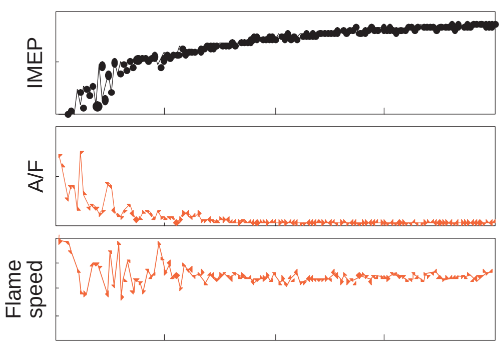


Fig. 1 Combustion characteristics under acceleration

* Automobile R&D Center
** Imagineering, Inc.

measuring the A/F ratio between the cylinders with the LAF sensor. Regarding the measurement of the combustion variation between cylinders, published reports include those that describe pressure[(1), (2)], ion probes[(3)], the IR absorption method[(4), (5)], and the chemiluminescence measurement method[(6), (7)].

As a new combustion diagnosis method, a micro-Cassegrain sensor was used to directly measure the A/F around the spark plugs, and measurement was performed of the corresponding time series variations and combustion variations between cylinders. In this method, time series analysis can be performed up to 20000 rpm.

This document is a report about the results of the unstable combustion analysis performed under the transient operation of a Formula One engine.

2. Measurement Device

A V8 race engine in 2008 that had a displacement of 2400 cm^3 was used. Figure 2 shows the exterior of the engine.

A micro-Cassegrain system (made by Imagineering, Inc.) was used to measure the A/F around the spark plugs.

When the flame passed a certain point (about 0.1 x 0.1 mm), the chemiluminescence intensities of four radicals (OH, CH, CN, C_2) from the flame chemiluminescence was measured using a band-pass filter and a photomultiplier.

The intensity ratios of these radicals have an unambiguous correlation with the equivalence ratio in the flame[(8)–(16)]. This means that an approximate measurement can be made of the flame propagation speed and the A/F in the flame.

The measurement system was comprised of an optical fiber sensor that measures the local chemiluminescence, hardware that processes the chemiluminescence intensity ratio in high-time resolution, control software, and a monitor.

The sensor type used was an M10 spark plug with a built-in Cassegrain optical element. The distance of the focal point from the ignition position was 3 mm.

The effect on combustion by inserting the sensor into the spark plug was checked. Figure 3 shows the results of the evaluation of the combustion variation using the pressure sensor. Normal spark plugs were installed for cylinders 1 to 4, and spark plugs with the built-in micro-Cassegrain sensors were installed for cylinders 5 to 8.

Fig. 2 Honda Formula one engine

The combustion pressure for 500 cycles was measured with an engine speed of 16500 rpm and WOT.

The air excess ratio (set Lambda) was set at 0.95.

The deviation of combustion pressure between cylinders was 16.7%, and so it was shown that the built-in micro-Cassegrain sensors did not greatly affect the combustion variation.

It was also confirmed that negative effects from the rpm, loads and Lambda value and the plasma from early discharge were not detected by the micro-Cassegrain sensors.

Sapphire glass was installed on the end of the micro-Cassegrain sensors as a countermeasure for heat and pressure resistance.

Figure 4 shows an overall view of cylinders 5 to 8 to which the micro-Cassegrain sensors were attached. Figure 5 shows a view of the optical fiber attached to a spark plug in a plug hole.

Figure 6 shows the device configuration. The Cassegrain sensor output value, in-cylinder pressure, induction pipe pressure, exhaust pipe pressure, LAF sensor output value, and crank angle signal were synchronized and imported.

SPB2000 (made by Imagineering, Inc.) and DS2000 (made by Ono Sokki Co., Ltd.) were used as the data processing devices. SPB2000 is equipped with a band-pass filter and interference filter for each radical, and the light signal from each cylinder is converted into an electrical signal by the photomultiplier.

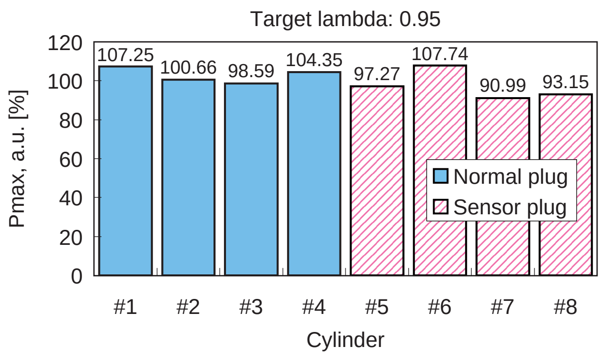


Fig. 3 Comparison of maximum cylinder pressure (Pmax) using normal plug and sensor plug

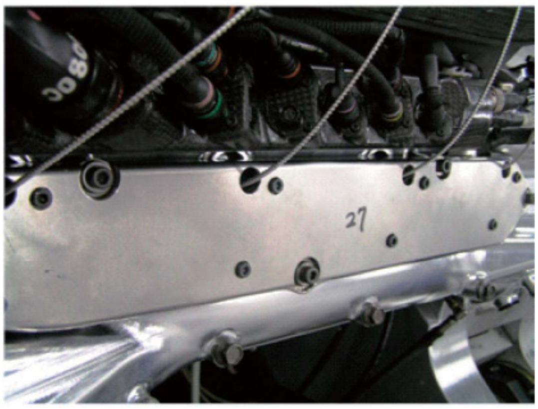


Fig. 4 Photograph of overview of cylinders to which micro-Cassegrain sensors were attached

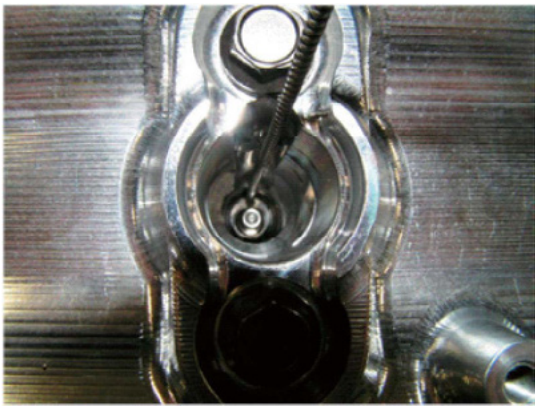


Fig. 5 Photograph of plug hole in which plug with sensor was attached

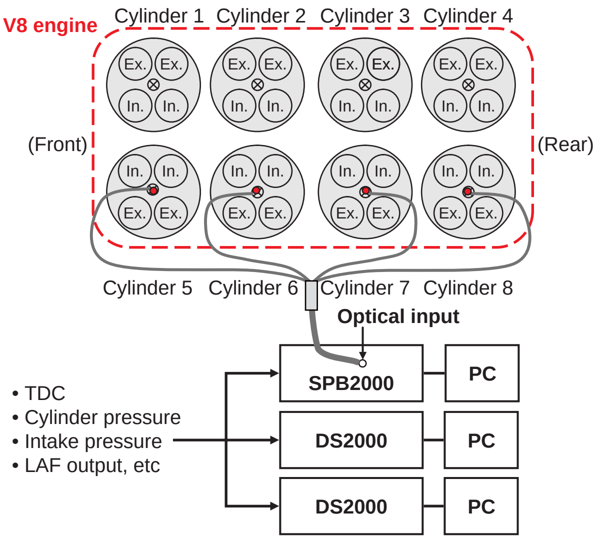


Fig. 6 Schematic diagram of measurement system

The hardware is comprised of an amp, filter, and A/D converter (14-bit). It can import data simultaneously and perform measurement with a resolution of a crank angle of 1° at an engine speed of 18000 rpm. The light emission data and pressure data for each cylinder can be checked on the monitor.

3. Results and Discussion

The ram pressure changes in accordance with the vehicle velocity, then the air flow pattern inside the air box changes. Therefore, the air inflow amount of each cylinder also changes. The air inflow amount also changes in accordance with the shape of the air box. In particular, unstable combustion may be triggered under transient conditions. The combustion analysis of the V8 Formula One engine was performed under such conditions. In the analysis, the Lambda around the spark plug, the flame propagation speed (FPS), reaction zone thickness, IMEP, P max, induction pressure and exhaust pressure were calculated, and the combustion difference and time series combustion variation were compared.

3.1. Combustion Characteristics at 18000 RPM Engine Speed, WOT Load, and Steady Condition

Figure 7 shows a comparison of the results with different vehicle velocities and air box specifications. The horizontal axis is Lambda, and the vertical axis is the average values and the variation range. The combustion difference and the time series combustion variation were compared between cylinders 5 and 6.

The faster the vehicle velocity, the higher the IMEP becomes because of the lam pressure. The highest pressure was reached when the set Lambda was around 0.9.

However, P max and IMEP values varied between cylinders. The combustion pressure variation range also varied according to the Lambda value.

At 120 km/h and 300 km/h in the standard air box, the average A/F and the A/F around the spark plugs were approximately the same, which is a desirable result.

The air box inlet shape was changed and measurement was performed at a vehicle velocity of 300 km/h. The A/F around the spark plug in cylinder 5 became richer by about 0.5 from the set value. The inlet shape affects the air flow pattern, and so it also likely affects the A/F distribution inside the combustion chamber.

No significant change was seen in the FPS within a Lambda range of 0.8 to 0.95. The same trend has been reported in another combustion analysis for Formula One engines[7].

However, it should be noted that the FPS varies greatly regardless of the Lambda.

3.2. Combustion Characteristics at 14000 RPM Engine Speed, Partial Load, and Steady Condition

Figure 8 shows the combustion characteristics of cylinders 5 and 6 with a throttle position of 30%, an

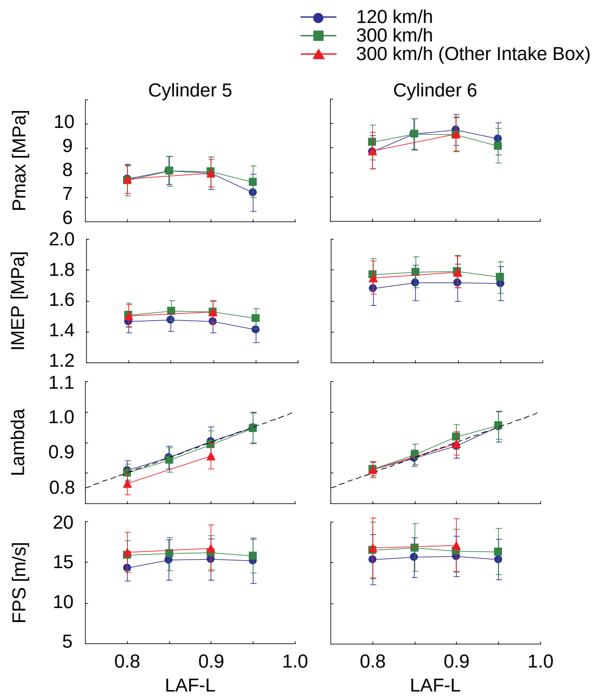


Fig. 7 Combustion characteristics at 18500 rpm and WOT

engine speed of 14000 rpm, and steady condition. It shows that the Lambda value shifts from the set value, and that the flame propagation value and the value of its variation changes significantly.

For cylinder 5, it is shown that the Lambda around the spark plug is lean under all conditions, and that a bias occurs in the A/F distribution around the spark plug under a partial load. For cylinder 6, the Lambda value is closer to the set value than it is for cylinder 5.

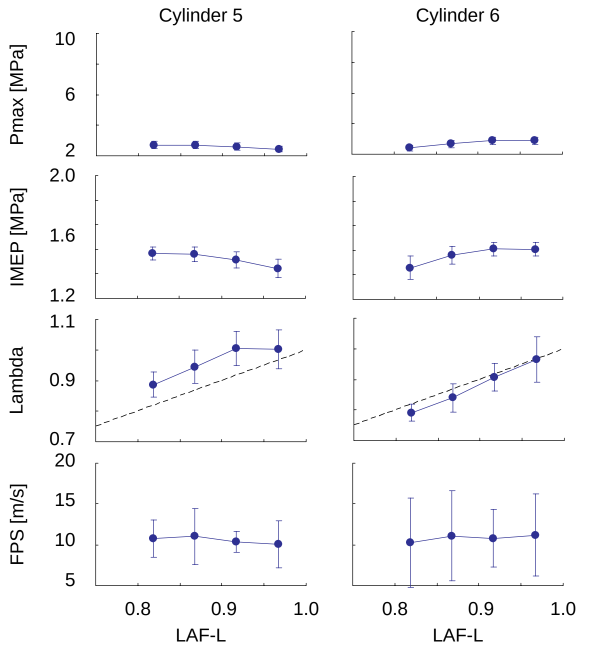


Fig. 8 Combustion characteristics at 14000 rpm and 30 % throttle opening

However, its FPS variation is larger than for cylinder 5, and it can be found that significant combustion instability occurs even though the formation of the air/fuel mixture is even.

In this way, the relationship between the set Lambda and the Lambda around the spark plug is greatly different even in cylinders that adjoin each other, and the corresponding time variation may also vary greatly. This is proof that although the macro-level state of all the cylinders can be ascertained from the pressure and LAF sensors, the detailed state around the spark plugs cannot be ascertained, and so there is a limit to the optimization that can be performed based on these measured values.

3.3. Combustion Characteristics in Transient Conditions

Combustion analysis in transient conditions was performed to confirm for misfiring and torque loss in transient conditions and identify why they occur. Measurement was performed at the set Lambda of 0.8 while accelerating at a ratio of 3000 rpm/sec from the engine speed of 9000 rpm. Figure 9 shows the P max, IMEP, Lambda and FPS for cylinders 5 and 8. The Lambda graph shows three types of data. (a) is the Lambda around the spark plug derived from the Cassegrain sensor, (b) is the value from the LAF sensors installed on the exhaust pipes of each cylinder, and (c) is the set Lambda. The horizontal axis is the number of cycles, and the 4500th cycle is the start of acceleration. Combustion variations that involve misfiring were observed from cycles 4530 to 4540. The IMEP also dropped dramatically.

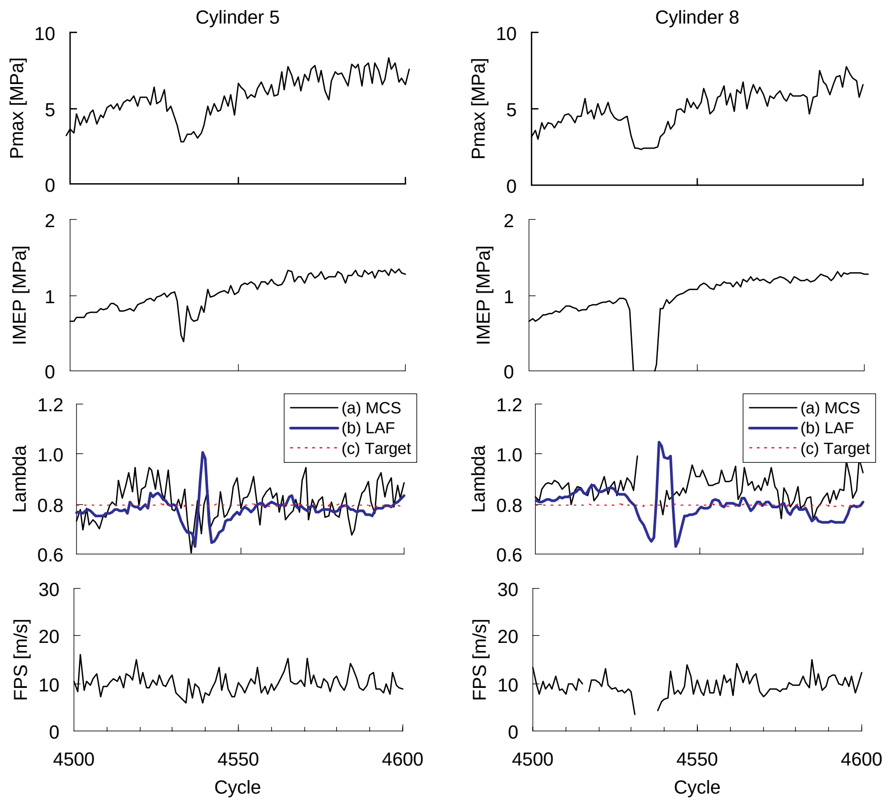


Fig. 9 Cyclic variations of combustion characteristics under transient condition

In cylinder 5, the Lambda value dropped but did not reach misfiring. However, the A/F around the spark plug was too rich, leading to a drop in output.

On the other hand, the misfiring phenomenon can be clearly seen in cylinder 8. In contrast to cylinder 5, it misfires because the A/F is too lean. After the misfiring, combustion does not start again immediately. It starts after four cycles.

This is likely the effect of the rigid vortex that occurs in the air box. A recirculating vortex is formed in the air box by the inertia force of the induction air. This is likely to result in different air fuel ratios between cylinders. This fuel bias state occurs even in steady conditions. A corresponding amount of fuel injection is set. However, it was found that a setting based on a steady condition is insufficient under a transient condition.

Because the vehicle is always driven under transient conditions during a race, engine calibration is required for these conditions.

3.4. Combustion Characteristics in Fuel Cut and Recovery Cycles

In order to perform torque control, the fuel injection control was performed in which the fuel was cut during deceleration and the fuel injection was recovered during acceleration. At this time, sometimes unstable combustion occurred during fuel recovery. In response, the fuel recovery state was simulated on a test bed, and the combustion state was measured. Figure 10 shows the results. It shows the P max, IMEP, MFB 0 to 10%, Lambda and FPS cycle data from the fuel cut to recovery for cylinder 6. The same as in Fig. 9, the Lambda graph

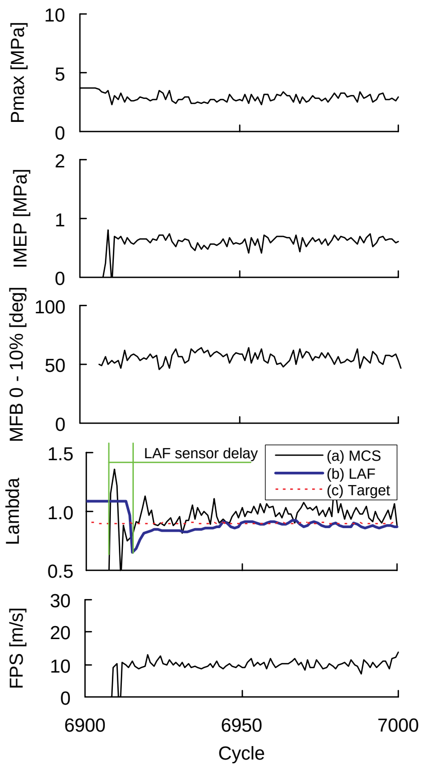


Fig. 10 Cyclic variations of combustion characteristics in fuel cut recovery test

includes the value around the spark plug, the LAF measurement value, and the set value.

Looking at the FPS, IMEP, and Lambda around the spark plug, it can be seen that there is no ignition in the first cycle after the fuel recovery. However, in the next cycle the mixture is rich and there is an ignition, and in the following cycle, the mixture changes to a lean one. It can be clearly observed that the Lambda around the spark plug changes alternately between rich and lean, and there is significant variation between cycles.

It was demonstrated that the micro-Cassegrain sensor can measure the Lambda for each cycle and has a high responsiveness.

On the other hand, because the responsiveness of the LAF sensor is low, it cannot measure the behavior of each cycle. It detects the Lambda value after fuel cut recovery with a delay of eight cycles.

4. Conclusion

The results obtained when combustion diagnosis of a Formula One engine has been performed using a micro-Cassegrain sensor are described below.

(1) Except under certain conditions, in WOT operations with an engine speed of 18500 rpm, average cylinder Lambda and Lambda around the spark plugs are approximately equal and so are in a desirable state.

(2) Under the operating conditions described above, the FPS is not influenced by the Lambda.

(3) The tendencies of the A/F distributions between cylinders differ under WOT and partial throttle conditions.Even if the average A/F in the cylinder is set to an appropriate value using the LAF sensor, the A/F around the spark plug may not be appropriate.

(4) The A/F behavior during transient operation was shown to be different from that during steady operation. When torque loss or unstable combustion occurred during transient operation, the A/F around the spark plug was too rich or too lean. Further, it was shown that the A/F behaviors were different between cylinders even during the same cycle.

(5) In the future, highly accurate fuel control will be required under transient conditions, and to achieve this, highly responsive measurement instruments will be required. The micro-Cassegrain sensor provides this function, and by using this instrument, it may be possible to accelerate future engine development.

References

(1) Neo, G. H., Collings, N.: Pressure Data Analysis of Formula One Racing Engines, SAE Paper, 970061 (1997)

(2) Alten, H., Illien, M.: Demands on Formula One Engines and Subsequent Development Strategies, SAE Paper, 2002-01-3359 (2002)

(3) Shimasaki, Y., Sakaguchi, J., Maki, H., Suzuki, H., Kondo, N., Yamada, T.: Study on Combustion Monitoring System for Formula One Engines Using Ionic Current Measurement, JSAE Technical Paper, No.

20055017 (2005)
(4) Ikeda, Y., Nishiyama, A., Kawahara, N., Tomita, E., Baritaud, T.: Application of IR Absorption Method to a Racing Engine, 2007 JSAE Annual Congress proceedings, No.13-07 (2007)
(5) Ikeda, Y., Nishiyama, A., Bariatud, T.: Flame Speed Measurement of a Racing Engine by IR Method and Chemiluminescence Method, 14th Lisbon International Symposium on Applications of Laser Techniques to Fluid Mechanics (2008)
(6) Ikeda, Y., Nishiyama, A., Kim, S. M., Takeuchi, A., Winklhofer, E., Baritaud, T.: Cyclic Variation of Local A/F and Mixture Quality in SI Engine Using Local Chemiluminescence, Proc. of 7th international symposium on internal combustion diagnostics, p. 277-284 (2006)
(7) Ikeda, Y., Jeong, H., Nishiyama, A., Baritaud, T.: Cylinder to Cylinder A/F Fluctuations Measurement in a Racing Engine by Chemiluminesence, Porc. of 7th International Conference on Modeling and Diagnostics for Advanced Engine System, p. 535-540 (2008)
(8) Lee, J. G., Kim, K., Santavicca, D. A.: Measurement of Equivalence Ratio Fluctuation and Its Effect in Heat Release During Unstable Combustion, Proc. of the Combustion Institute, 28, p. 415-421 (2000)
(9) Ohyama, Y., Ohsuga, M., Kuroiwa, H.: Study on Mixture Formation and Ignition Process in Spark Ignition Engine Using Optical Combustion Sensor, SAE Paper, 901712 (1990)
(10) Ikeda, Y., Kaneko, M., Nakajima, T.: Local A/F Measurement by Chemiluminescence OH*, CH* and C2* in SI Engine, SAE 2001 Transactions, Journal of Engines, Section 3, 110 (2001-01-0919) (2001)
(11) Ikeda, Y., Nishihara, H., Nakajima, T.: Measurement of Flame Front Structure and Its Thickness by Planar and Local Chemiluminescence of OH*, CH* and C2*, SAE 2001 Transactions, Journal of Engines, Section 3, 110 (2001-01-0920) (2001)
(12) Ikeda ,Y., Kojima, J., Hashimoto, H.: Local Chemiluminescence Spectra Measurements in a High-Pressure Laminar Methane/Air Premixed Flame, Proc. of the Combustion Institute, 29, p. 1495-1501(2002)
(13) Kojima, J., Ikeda, Y., Nakajima, T.: Multi-point time-series observation of optical emissions for flame-front motion analysis, measurement Science and Technology, Vol. 14, P. 1714-1724 (2003)
(14) Ikeda, Y., Kawahara, N., Tomita, E.: Time-series A/F Analysis in a SI Engine by Micro-Local Chemiluminescence Technique, Proc. of 6th COMODIA 2004, Paper No.C3-3 (2004)
(15) Kawahara, N., Tomita, E., Takeuchi, A., Arimoto, S., Ikeda, Y., Nishiyama, A.: Measurement of flame propagation characteristics in an SI engine using micro-local chemiluminescence technique, SAE Paper, 2005-01-0645 (2005)
(16) Ikeda, Y., Nishiyama, A., Kawahara, N., Tomita, E., Arimoto, S., Takeuchi, A.: Spark-plug-in sensor for initial flame analysis and its flame structure in an SI engine, SAE Paper, 2005-01-0644 (2005)

■ Author ■


Naoki HANADA


Atsushi HIRAIDE


Kohei YAMADA


Yuji IKEDA


Atsushi NISHIYAMA

Development of Wind Simulator Equipment for Analysis of Intake Phenomena in Formula One Engines

Satoshi NAKAMURA* Yasuhiro MOTOHASHI* Shuichi HAYAKAWA*

ABSTRACT

Formula One vehicles repeatedly accelerate and decelerate, reaching maximum speeds of 300-370 km/h and facing maximum G forces of 4.0 G. Under driving conditions such as these, the vehicle's wind speed (ram pressure) has a significant effect on engine power. However, due to issues involving the fitting of measurement devices onto vehicles, the prioritization of vehicle set-up work, and other factors, time has not been available for measurements to be taken, and analyses of air intake phenomena due to ram pressure during circuit driving have not been conducted.

A wind simulator able to reproduce ram pressure during circuit driving was developed, enabling efficient development of intake systems using bench tests.

In addition, dyno equipment with identical inertia characteristics to those of a vehicle drivetrain and a vehicle simulator system able to simulate body characteristics, throttle and accelerator work, and the conditions of different courses were added to the system. This enabled circuit driving modes to be used in bench tests, making it possible to establish appropriate intake settings for each circuit.

1. Introduction

Recent Formula One regulations have reduced the number of days for running tests and placed limitations on the annual driving distance in order to reduce costs. This has necessitated a thoroughgoing reduction of engine-related running tests via higher accuracy of performance predictions at the diagram stage and the use of bench tests in place of running tests.

In order to realize this goal, items from running tests were not simply substituted in bench tests; a device was developed that would enable phenomena in the engine during circuit driving to be clarified.

The factors that affect dynamic performance in engine-related running tests are changes in the vehicle's wind speed (ram pressure) and atmospheric conditions (temperature and humidity). Conventional bench tests for intake system parts were unable to reproduce the vehicle's wind speed (ram pressure), temperature, and humidity conditions in the high-speed range for all circuits, and there were therefore cases in which specifications determined on the basis of bench test results were ineffective and were not applied.

A wind simulator (ram pressure generator) was developed as a means of resolving this issue. In addition, the use of a vehicle simulation system able to employ circuit driving modes enabled engine phenomena during circuit driving to be reproduced.

The developed equipment, Real Vehicle Dyno bench (RV bench), consisted of a wind simulator, dyno equipment displaying identical inertia characteristics to the drivetrain of an actual vehicle, and a vehicle simulation system.

The vehicle simulation system was based on the Vehicle Simulation Module (VSM) produced by AVL List GmbH.

In addition to controlling engine torque based on engine speed and throttle opening via the dyno equipment, the system reproduced driving conditions through substitution of different vehicles, drivers, and courses, enabling lap times to be evaluated. The system featured a variety of modes, including cornering (deceleration and acceleration), full lap driving, and anti-stall (idling and take-off).

Because the system enabled evaluation of the effect of the parts employed on lap time in addition to power output, it provided race teams with a more informed perspective, and increased the efficiency of development.

This paper will provide an overview of the system, its correlation with circuit driving, and the results of its use in the analysis of engine air intake phenomena.

* Automobile R&D Center

2. Overview of Equipment

2.1. Overview of Wind Simulator

Figure 1 shows a section view of the RV bench, and Fig. 2 shows an overhead view of the test room. Figure 3 shows the bench test equipment with an engine fitted.

The air blowers were designed for transient performance enabling wind speed to be varied to reproduce typical acceleration (1.7 G) and deceleration (4.0 G) conditions for Formula One vehicles. In order to minimize temperature increases through adiabatic compression, the wind speed distribution was rendered uniform by positioning variable-flow cooling fins, the shapes of which had been designed based on CFD, in the air ducts.

To enable the temperature of the air used in the system to be adjusted appropriately in response to temperature-setting commands, a boiler and a refrigeration machine were positioned in the second-floor test room, and the air produced here was sent to the first-floor blower room.

Three blower motors were used to bring the air in the blower room to the target wind speed, after which it was sent to the first-floor test room.

Three air ducts were set up in the first-floor test room, supplying air at a uniform wind speed (reproducing the vehicle wind speed during circuit driving) to the air box inlet, radiator, and oil cooler.

Wind could be blown by the equipment in response to simulated vehicle speed (0-360 km/h) while meteorological conditions (temperature and humidity)

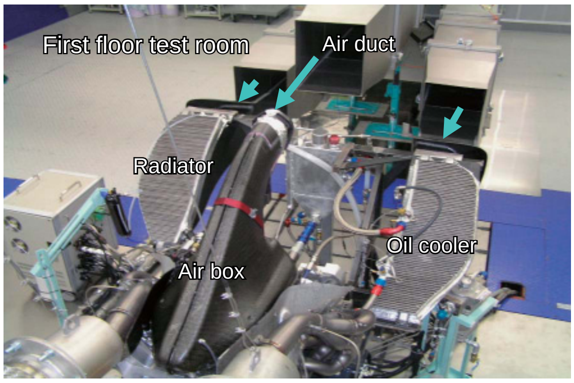


Fig. 3 Overview of test room

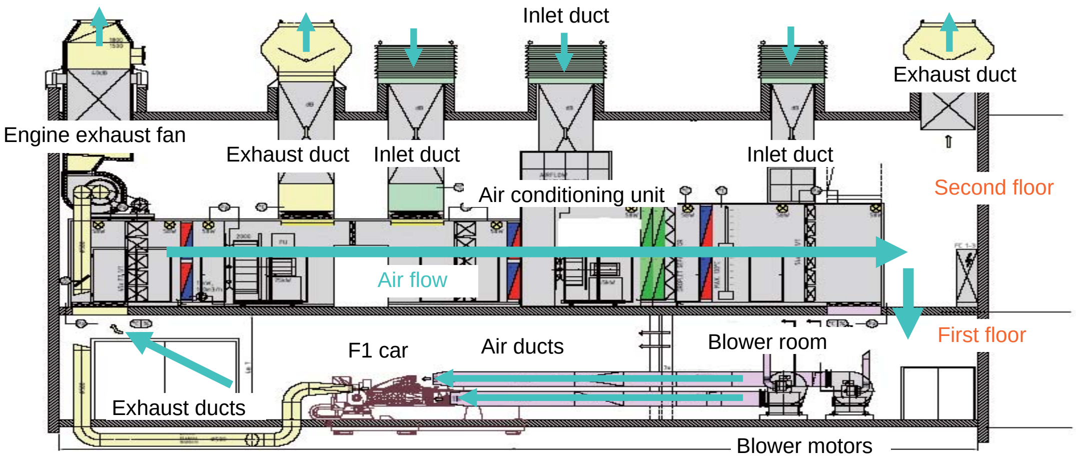


Fig. 1 Section view of RV bench

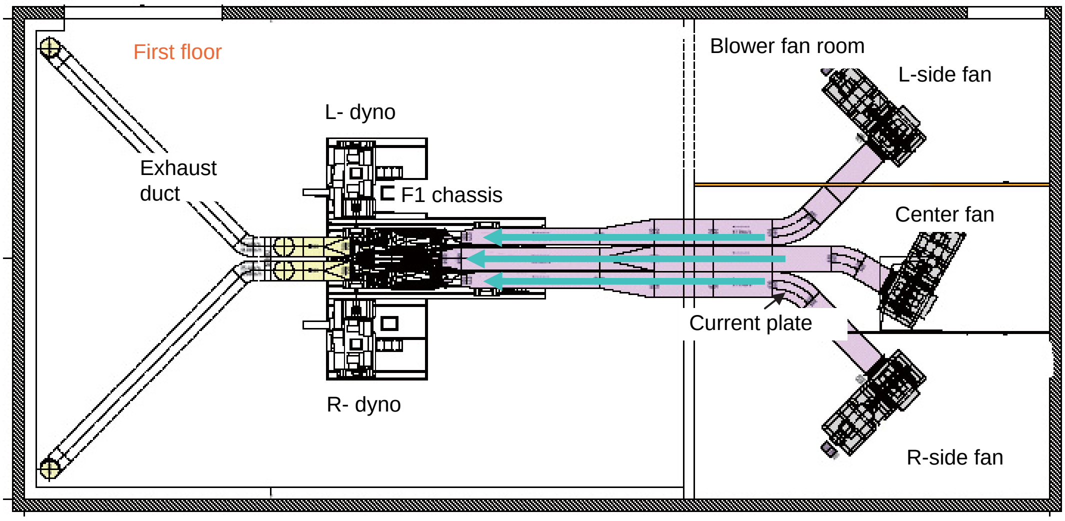


Fig. 2 Top view of 1F test room

Table 1 Control range

Windspeed	0-360 km/h (Max 380 km/h)
Acceleration and deceleration	Max 4.0 G
Temperature	10-40 °C
Humidity	30-90% (Rh)

Table 2 System specifications

<Dyno motor>

AC-motor mass Inertia	0.767 kgm^2
M dnom	2,000 rpm
P mon	400 kW
N max	3700 rpm

<Gear box>

Max input speed	22000 rpm
Transfer power	400 kW × 2 (Max 800 kW)
Gear ratio	9
Inertia	0.015 kgm^2
Acceleration	130000 rpm/sec
Control speed	9 kHz cycle

were controlled. The equipment was centrally controlled from an operating panel and a control panel, enabling wind speed, temperature, humidity, transient speed, and exhaust duct air flow to be controlled collectively.

The equipment was linked to an engine-side dyno controller and a vehicle simulation, creating a system in which a vehicle's wind speed could be varied in response to vehicle speed during a circuit driving simulation. The system was able to reproduce the atmospheric conditions of all the circuits used in Formula One racing. Table 1 shows the control range for the system.

2.2. Ultra-low Inertia Dyno Equipment

If the system was able to operate in circuit driving modes, it would become possible to conduct control system development, previously only possible using running tests, in a more concentrated manner. This would enable a consistent development process, in which control system development would be conducted following the evaluation of hardware and devices.

Ultra-low inertia dyno equipment displaying identical inertia characteristics to the drivetrain of an actual vehicle was introduced to the system to enable circuit driving modes to be realized. Table 2 shows the specifications of the AVL-manufactured equipment.

3. Correlation with Circuit

3.1. Pressure Distribution at Air Box Inlet

The wind simulator produced a uniform distribution of wind speed in the air ducts. However, under actual driving conditions, the pressure distribution at the air box inlet is not uniform, and it was necessary to reproduce this condition in the bench tests. Naturally, the ideal situation is to ultimately render wind speed uniform.

In order to achieve a correlation between the wind speed distribution during circuit driving and in the RV bench, airspeed was measured by taking measurements of the pressure distribution at the air box inlet using a pitot tube sensor (Fig. 4).

It was determined that the wind speed distribution in the RV bench was changed by 1) the vehicle's windscreen, and 2) changes in the air flow produced by the driver's helmet.

The effect of these factors on pressure distribution and engine output was analyzed in order to enable the pressure distribution in the RV bench to be correlated with the pressure distribution during actual driving.

Figure 5 shows the complete RV bench test room, and Fig. 6 shows the total pressure distribution at the air box inlet.

When the total pressure distribution during actual driving at an engine speed of 18500 rpm and vehicle speed of 300 km/h [total pressure distribution (a)] was compared with the total pressure distribution in the initial RV bench under the same conditions [total pressure distribution (b)], it was discovered that while the total pressure distribution at the air box inlet in the RV bench was uniform, part of the lower section of the air box displayed a lower pressure during actual driving.

Fig. 4 Pitot sensor position

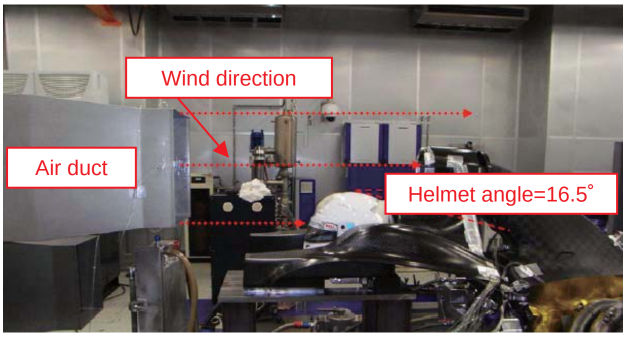


Fig. 5 View of RV bench test room

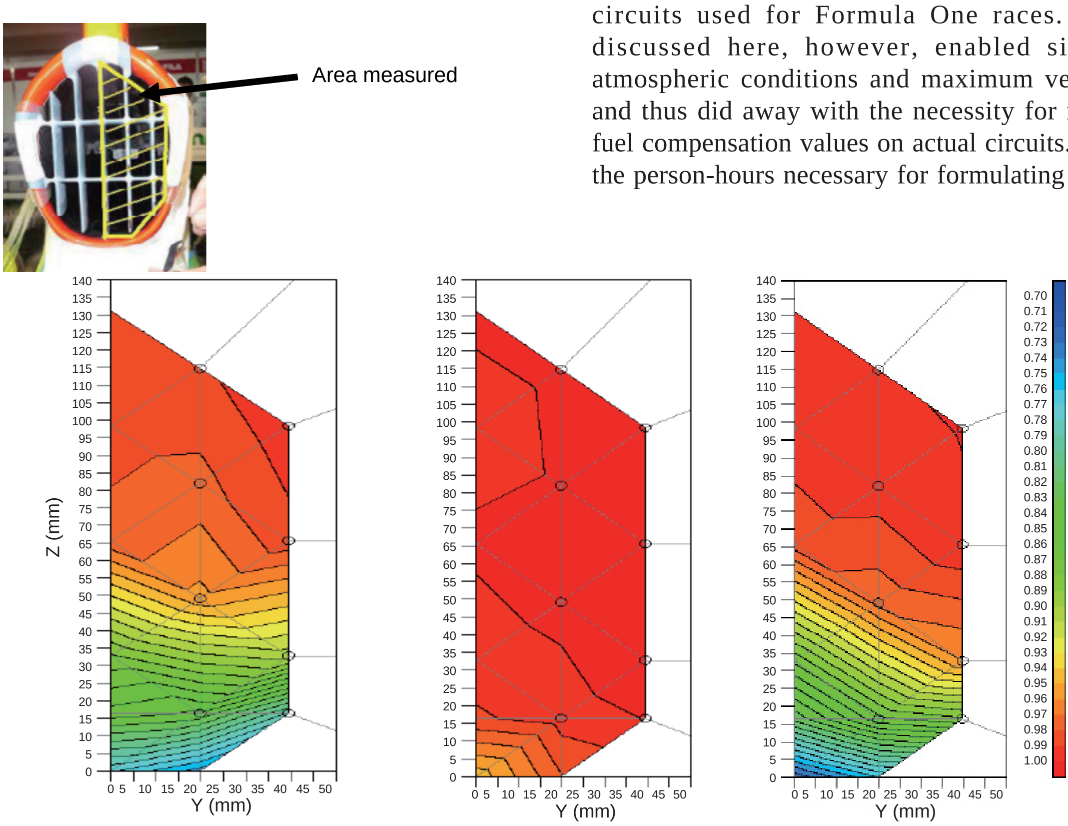
One difference between actual driving conditions and the RV bench was the windscreen used on the vehicle. The windscreen, a plate that sits at the front part of the cockpit and blocks wind, is shown in Fig. 7.

A dummy windscreen that produced the same effect as the actual windscreen and a vertical fin were positioned at the outlet of the RV bench (Fig. 8), and measurements were taken. The total pressure distribution under these conditions [total pressure distribution(c)] reproduced the lower pressure in the lower section of the air box, indicating that the system was now able to simulate actual driving conditions.

Next, in order to determine the effect of the total pressure distribution at the air box inlet on engine output, comparative tests were conducted on the initial RV bench (b) and the RV bench following modification (c). At vehicle speeds of 240 km/h or more, there was a difference of 6 kW in output, equating to a difference of 0.12 sec in lap times. In order to correct the uneven pressure distribution, consideration of the vehicle shape and positioning of the roll hoop were important factors in addition to the shape of the air box.

3.2. Change in Amount of Fuel Compensation between Cylinders in relation to Vehicle Speed

Given the fact that optimization of the air/fuel ratio in each of the cylinders will increase power, fuel compensation values have been studied using conventional bench testing equipment, and refined on the circuits used for Formula One races. The system discussed here, however, enabled simulation of atmospheric conditions and maximum vehicle speeds, and thus did away with the necessity for refinement of fuel compensation values on actual circuits. This enabled the person-hours necessary for formulating settings to be



Total Cp distribution (18500 rpm, WOT, 300 km/h)

(a) Track

Total Cp distribution (18500 rpm, WOT, 300 km/h)

(b) RV bench

Total Cp distribution when 225 mm windscreen is added (18500 rpm, WOT, 300 km/h)

(c) RV bench after modification

Fig. 6 Total Cp at inlet air box

Fig. 7 Windscreen

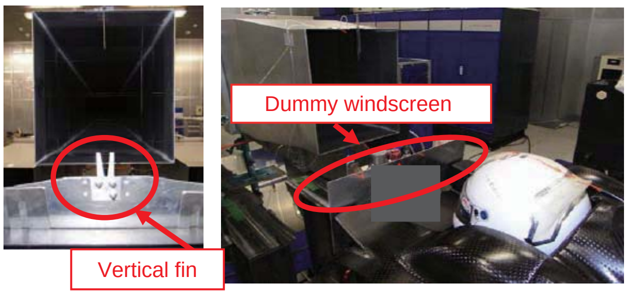


Fig. 8 Vertical fin and dummy windscreen

reduced and data for pit strategy, such as the amount of fuel to be carried by the vehicle, to be formulated in a more timely fashion. Figure 9 shows the fuel compensation values for each cylinder resulting from different ram pressures. The cylinders are compared at vehicle speeds of 120 km/h and 340 km/h, with engine speed set at 18500 rpm. On the low vehicle speed and high vehicle speed sides, increases in ram pressure result in changes in the flow of air in the air box, and the fuel allocation to each cylinder (fuel compensation) therefore also changes. As a result, the fuel compensation values for the forward cylinders (#1, 2, and 5), which receive minimal air intake at 120 km/h, increase by 2-4% at 340 km/h. Fuel compensation increases output by 3 kW.

3.3. Effect of Intake Air Temperature in Engine Cowl

A monocoque used in an actual vehicle was placed in the RV bench, and temperature measurements were conducted in the air box and the engine cowl. Measurements were taken of the:

(1) Temperature of the intake air at the air box inlet
(2) Temperature distribution inside the air box
(3) Temperature distribution around the bulkhead
(4) Temperature distribution around the rear of the engine
(5) Temperature distribution of the pneumatic air cylinder
(6) Temperature distribution around the oil cooler
(7) Temperature distribution around the radiator

Figure 10 shows the interior of the air box.

The temperature distribution in the air box (Fig. 11) was the point of focus among the measurement results. The results showed that the temperature at the front of the air box was 8 °C higher than the temperature at the rear. The cylinders at the front were taking in higher temperature air than the cylinders at the rear.

This was a phenomenon that could not be reproduced without using a monocoque and engine cowl in the system; the use of an engine in isolation in bench tests was insufficient to reproduce the temperature environment of an actual vehicle.

The results of analysis of the temperature increase in the front of the air box showed that the increase was synchronized with changes in the oil temperature in the oil tank positioned between the engine and the bulkhead rather than radiant heat from the engine and the exhaust pipe, and the front of the air box was thus being affected by heat from the oil tank.

The fact that the bulkhead was an enclosed structure with no channels for cooling air resulted in an increase in the temperature of the front wall of the air box, and a consequent increase in the temperature of the intake air. Results showed that the temperature of the wall reached 90 °C.

To help resolve this issue, a duct for cooling air was added between the bulkhead and the engine.

As Fig. 12 shows, this enabled cooling air to flow around the oil tank and between the V-banks, reducing the temperature of the air box wall to 45 °C and the temperature of the intake air at the front of the air box by 6 °C.

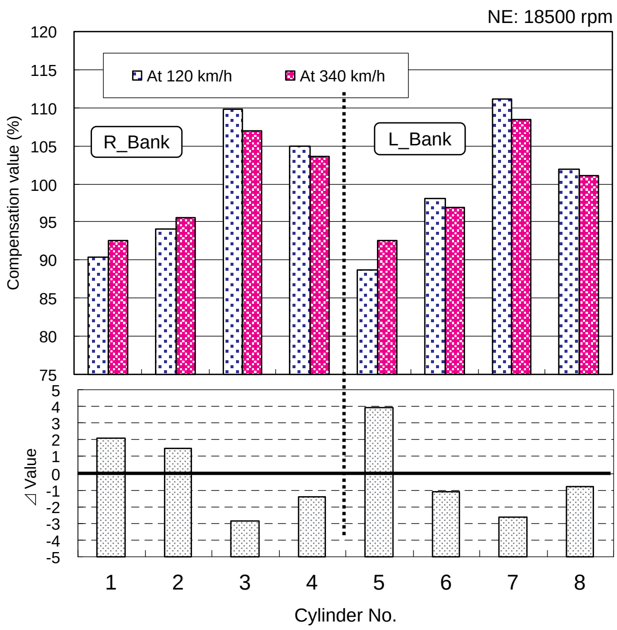


Fig. 9 Speed-fuel distribution for each cylinder

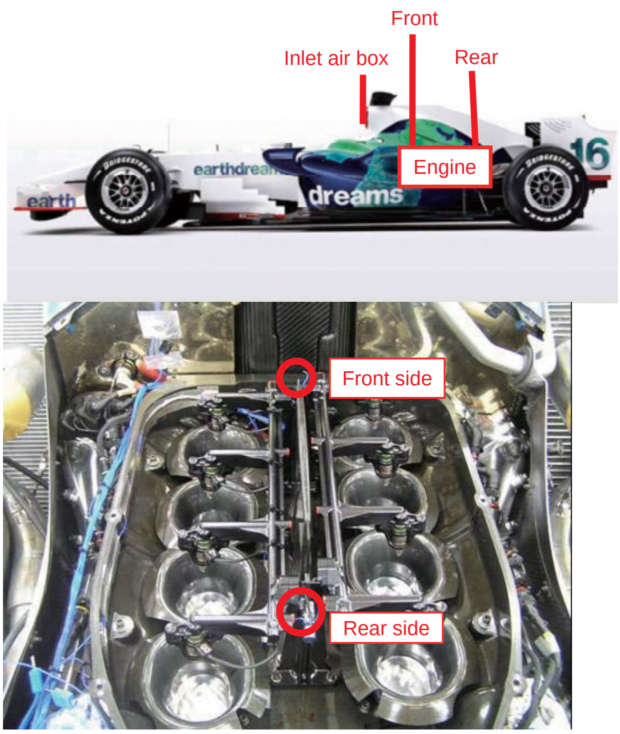


Fig. 10 View of engine cowl and air box

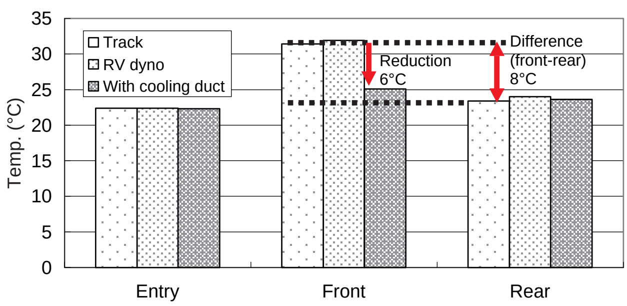


Fig. 11 Temperature distribution in air box

Because, based on the regulations, Formula One engines have not been able to employ variable induction systems from 2006, when these types of changes occur in the temperature of the intake air, the engine speed for peak output changes. An increase in the temperature of the intake air may reduce intake efficiency and become a factor in reducing output.

Heat insulating sheets were tested as a measure against radiant heat, but it was found that temperature increased over time, and that the sheets had no effect in 10-lap mode.

Scavenging by the cooling duct would be an effective measure, but it would be necessary to study the shape of the cooling duct with consideration of its effect on the aerodynamic performance of the vehicle.

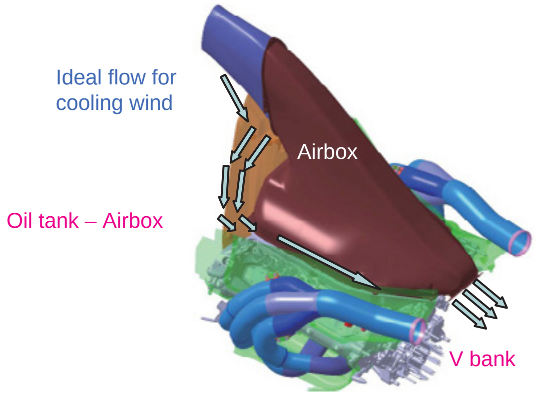


Fig. 12 Cooling duct (ideal flow)

4. Substitution of Data in the System

The RV bench enabled data for settings measured in running tests to be verified in bench tests. The following items were substituted in bench tests:

(1) Pressure and temperature measurements from new vehicle shakedowns
(2) Radiator and oil cooler function checks
(3) Verification of vibration and heat damage (verification of reliability of parts)
(4) Verification of performance in relation to atmospheric conditions and ram pressure
(5) Verification of performance and reliability of electrical system
(6) Evaluation of drivability using vehicle simulation
(7) Evaluation of lap time, evaluation of fuel efficiency
(8) Settings for control data
(race start, traction control, torque mapping for each circuit)

5. Conclusion

(1) A wind simulator (ram pressure generator) and a vehicle simulation system with circuit driving modes were developed, and used to reproduce engine intake phenomena during circuit driving on the bench.
(2) The establishment of a correlation between circuit driving and bench test conditions enabled the length of time necessary for air box development, the formulation of settings for fuel compensation between cylinders, the selection of intake pipe length, and other tasks to be reduced.
(3) The use of a monocoque vehicle in the system enabled the effects of heat damage to be verified, contributing to the achievement of increased power in actual race vehicles.

■ Author ■

Satoshi NAKAMURA

Yasuhiro MOTOHASHI

Shuichi HAYAKAWA

Measurement Technologies for Formula One Engines

Nagao YANAGISAWA* Naoki HANADA* Takeru HAMAKAWA*
Kazushi OGIYAMA* Tetsuo GOTO*

ABSTRACT

In order to attain an accurate grasp of physical phenomena in the high-speed measurement environment of a Formula One engine, it is necessary to be equipped with measurement systems capable of high-volume, high-speed sampling. In addition, the accuracy of sensors and the vibration resistance of measurement equipment are important factors. It is also necessary to reduce the size and weight of measurement systems in order to conduct measurements during circuit driving. To respond to these performance demands, a variety of high-performance engine measurement systems were used in Formula One development projects.

This paper will discuss combustion diagnosis using a misfire detection system, measurement of combustion pressure and crankshaft behavior during circuit driving, oil pressure measurement, friction measurement, and the visualization and measurement of the fuel spray and in-cylinder flows.

1. Introduction

For all automotive engines, including the engines used in Formula One vehicles, engine development is inextricably bound to technological progresses and evolutions in the field of measurement. It has become standard procedure throughout the industry to determine specifications through the analysis of phenomena such as the vehicle's dynamic behavior, as typified by vibration and other parameters, combustion pressure, which is intimately related to the level of power of the vehicle, and the internal pressure of the inlet and exhaust pipes, which determines filling efficiency.

As Table 1 shows, the evolution of measurement technologies between Honda's second Formula One era (1983-1992) and third Formula One era (2000-2008) made it possible to conduct onboard measurements of combustion pressure, vibration, and dynamic behavior in an actual vehicle up to an engine speed of 20000 rpm. This technological evolution resulted from increased efficiency in packaging, for example the reduction of sensor size, increases in CPU speeds, and increases in measurement capacity, which helped to enable high-accuracy measurements in shorter periods.

With these advancements in measurement technologies, the range of areas in which measurements could be taken increased, and it became possible to obtain a more accurate understanding of engine phenomena.

As a result, it became possible to rapidly determine engine specifications and resolve issues, and thus to supply high-quality engines within a shorter time frame.

Table 1 Comparison of contents of measurements

Item	Second era	Third era
Combustion pressure	Only one cylinder is measured (Only dyno)	All cylinders are measured (Dyno and circuit)
Crankshaft twist vibration	Only dyno	Dyno and circuit
Engine vibration	Only dyno	Dyno and circuit
Gear train vibration	Measurement is impossible	Measurement of all gears is possible (Only dyno)

At the end of 2006, new regulations placed restrictions on changes to almost all engine specifications. From 2007, engine development focused on the achievement of increased power and reliability by means of small changes within the scope allowed by the regulations, and it was necessary to change development methods accordingly.

It was important to make accurate decisions on whether or not to employ specific proposals within a restricted scope of potential changes, necessitating a transition to methods incorporating computer aided engineering (CAE) in order to study results in advance.

The establishment of an accurate correlation between actual phenomena and CAE results is an important factor in the introduction of CAE, and thorough studies were therefore conducted using single-cylinder engines.

This paper will provide an introduction to the latest measurement technologies for Formula One engines.

* Automobile R&D Center

2. Measurement Technologies

2.1. Combustion Measurements

2.1.1. Misfire detection system (MDS)

Because of the high speed of Formula One engines, the combustion process is completed in a short period. In addition, the on-off load when switching from wide-open throttle to fully-closed throttle is constantly repeated. Because of this, the frequency of use of the engines in an unstable state of combustion is high. A misfire detection system (MDS) that is able to diagnose the state of combustion under these conditions using ionic current was developed, and employed in all races from 2002.

Figure 1 shows the mechanism of generation of ionic current. It has long been known that radical ions are generated during the combustion process. Ionic current is detected using the following two processes, and the waveforms have two peak values[(1)]:

(1) Detection of relatively long-lived $C_3H_3^+$ radical ions (chemical ions) generated when the flame surface passes through the electrode during the initial stage of combustion.

(2) Detection of NO_2^+ radical ions (thermal ions) generated by thermal dissociation of N_2 in combustion gases at temperatures of 2000 K or higher.

Figure 2 shows the principle of ionic current measurement. With the central electrodes of the spark plugs used as ion probes, a positive electrical potential of approximately 300 V is impressed, and the $C_3H_3^+$,CHO^+, and NO_2^+ radical ions generated by combustion are captured.

The ignition coil energy is stored in the condenser and used as a power source.

Honda's Formula One engines utilized a condenser discharge ignition (CDI) system with a discharge time of 70 μs. This minimized the effect of ignition noise, helping to enable the greater part of the combustion period to be monitored.

Lean air-fuel mixtures are used during low fuel consumption operation, but in this state engine hesitation can occur and acceleration performance out of corners can decline. Engine hesitation results from misfires, or combustion states close to misfire, due to the leanness of the air-fuel ratio falling below the limit for combustion with a rapid increase in the volume of intake air.

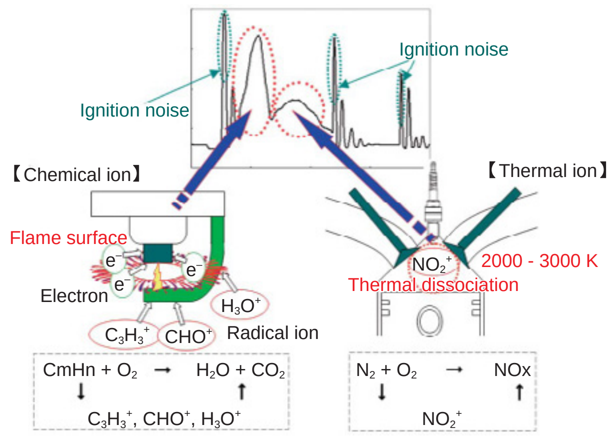


Fig. 1 Mechanism of ionic current

The MDS was developed in order to analyze this phenomenon.

Figure 3 shows a comparison of detected MDS values for the A/F setting for maximum power and a lean A/F ratio during acceleration. For the maximum power A/F ratio setting, the transition from a lean to a rich ratio occurs in a short period. The period during which MDS values are generated is also minimized, and combustion stabilizes rapidly. However, for the lean A/F ratio setting, the period during which MDS values are generated continues, and an unstable combustion state continues. This generates conspicuous hesitation.

The development of a technology that helped to enable monitoring of the state of combustion during vehicle operation made it possible to set appropriate A/F ratios for the atmospheric conditions of each circuit.

2.1.2. Combustion pressure sensor integrated with spark plug (PS plug)

Combustion pressure measurements were conducted constantly during the Formula One engine development program, but the following technological issues arose with regard to measurements using combustion pressure sensors:

(1) Low degree of freedom of layout

Limitation of areas for mounting of sensors due to engine configuration, resulting in inability to measure pressure in all cylinders.

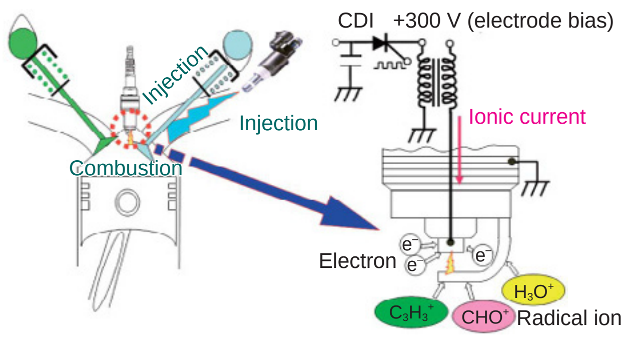


Fig. 2 Ionic current measurement principle

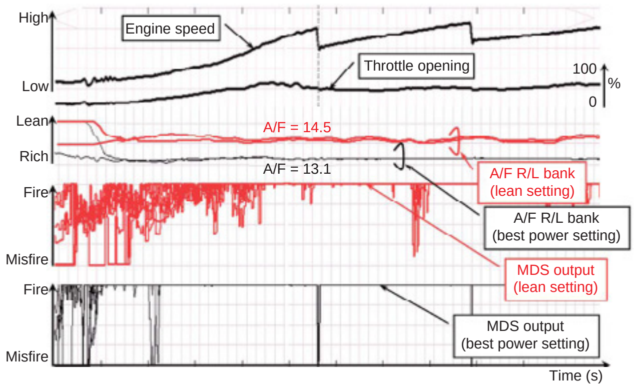


Fig. 3 Detected misfire

(2) Low level of durability and reliability
Sensors damaged by a high level of vibration due to high engine speeds.

Given this situation, a combustion pressure sensor integrated with a spark plug (termed a “PS plug” below) was developed as a technology for conducting pressure measurements in all cylinders.

Figure 4 shows the shape of the PS plug, and a measured combustion pressure waveform.

Comparison of combustion pressure measurements obtained using the PS plugs in test bed tests with pressure measurements taken using existing sensors showed that the plugs performed well.

The use of the PS plugs made it possible to bypass the limitation on sensor mounting positions and measure pressure in all cylinders, and thus to understand the differences in combustion pressure between cylinders.

Figure 5 shows the results of combustion pressure measurements taken during circuit driving and results for engine power calculated from the indicated mean effective pressure (Pmi) from the measurement results.

The indicated power calculated from the Pmi on the circuit displays identical values to those obtained from a torque meter mounted on the gearbox, indicating that the PS plug accurately measures combustion pressure even during circuit driving.

2.2. Crankshaft Torsional Vibration

Normally, the excitation of torsional vibration of the crankshaft is determined by the sum of the inertia of the reciprocating parts such as the pistons and the conrods, and the rotational torque resulting from combustion pressure.

Because Formula One engines operate at high speeds, the inertia of the reciprocating parts is dominant in the load on the crankshaft pin journals, and the load exceeds 50 kN at its maximum.

The crankshaft has a specific vibration frequency determined by its specific torsional stiffness and rotational inertia. However, resonance is generated by the excitation force mentioned above, and this can result in large torsional vibrations, the amplitude of which exceeds 1 deg.

Because Formula One engines use a central oil supply system, as shown in Fig. 6, part of the crankshaft is hollow, resulting in a decline in strength against a solid crankshaft. For this reason, torsional resonance might cause breakage of the crankshaft.

In the 2002 Formula One engine, torsional vibration resonating in three engine speed orders at high speeds under a motored condition became an issue. In terms of circuit driving conditions, this corresponded to closed-throttle in the straight ends; the frequency of such use is high, and this has an effect on the durability of the engine. In fact, crankshafts broke twice on circuits. Because torsional vibration occurred at high engine speeds, it represented an impediment to increasing the speed of the engine in order to increase power.

The analysis system configuration shown in Fig. 7 was used to measure the torsional vibration of the crankshaft on the bench and on the track, and the origins of the phenomenon were analyzed.

Because high-speed sampling was necessary, a data logger able to measure up to 200 kHz was used to record pulse signals for the detection of crankshaft rotation signals. These signals were converted into frequencies and the changing component of the signal was isolated. A Fourier transform was further applied to the changing signal component, helping to enable the torsional vibration of the crankshaft to be understood.

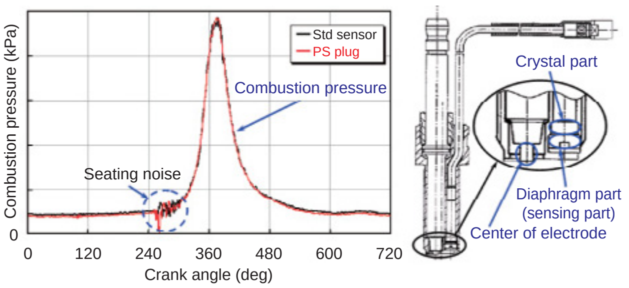


Fig. 4　Combustion pressure force and shape of PS plug

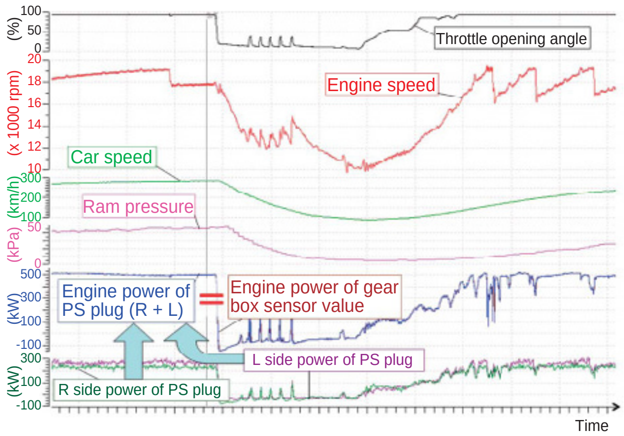


Fig. 5　Circuit running data

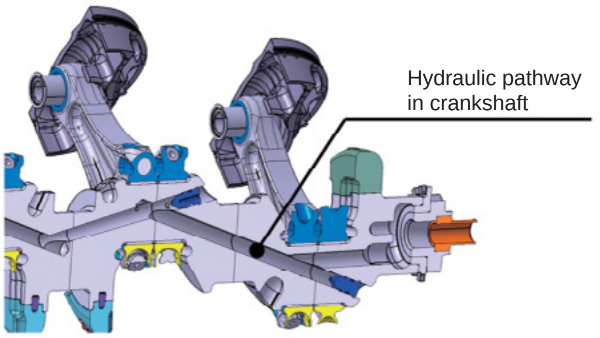


Fig. 6　Section of Formula One crankshaft

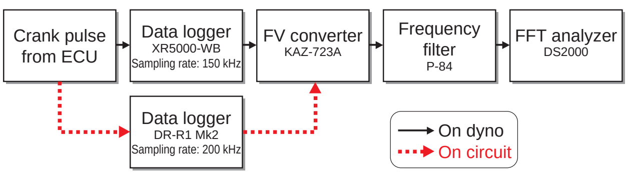


Fig. 7　Analysis system configuration

The results showed that the increase in torsional vibration originated in the bank angle used to lower the center of gravity. In the 2003 Formula One engine, the bank angle was reduced from 94˚ to 90˚, reducing torsional vibration of the crankshaft.

Results were also verified using the vibration simulation Blicks, manufactured by AVL Japan, K.K., in parallel with the measurements.

Figure 8 shows the simulation results obtained using Blicks and the measurement results.

The level of vibration at high engine speeds under a motored condition, which was an issue in the 2002 Formula One engine, was alleviated in the 2003 engine.

The good correlation between the simulation results and the results of vibration measurements in an actual vehicle demonstrated the effectiveness of the simulation.

As indicated by the discussion above, understanding the torsional vibration of the crankshaft when operating at high speeds necessitated high-speed sampling, and advances in data logger technology made these measurements possible.

2.3. Hydraulic Pressure Measurements

The major issues for the durability and reliability of the conrod bearings are wear resistance and seizing resistance. To address these issues, changes were made in the bearing materials, and the viscosity, amount, pressure and aeration of the lubricating oil were modified.

Formula One engines employ a central oil supply system, in which the centrifugal force of the crankshaft is used to supply oil to the conrod bearings.

Because of this, only the pressure in the hydraulic pathway of the crankshaft could be measured in order to measure actual hydraulic pressure. This necessitated a technology to output the signal from pressure sensors in the hydraulic pathway to the exterior of the engine.

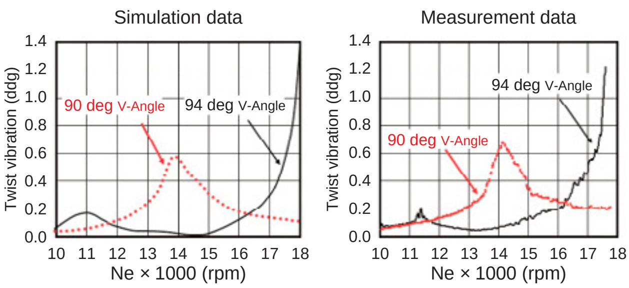


Fig. 8 Simulation results and measurement results

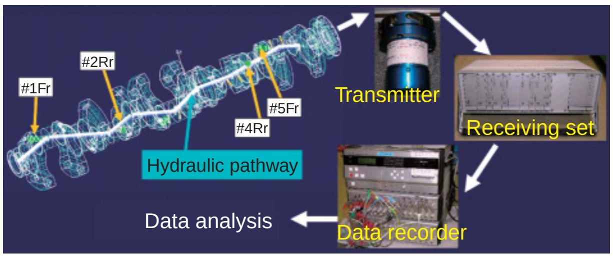


Fig. 9 Analysis system configuration

Figure 9 shows the configuration of the hydraulic pressure measurement system.

Previously, a contact-type slip ring using mercury as a physical signal path had been employed to guide signals from the crankshaft. However, engine speeds continued to increase, and the range of speeds at which this method could be used was limited. This was a result of the limits of the capacity of seals to deal with the heat generated and the fluid leaks occurring when mercury flowed at high speeds in the cylinders.

To replace this method, wireless technology was employed to transmit the signal outside the crankshaft. By contrast with the use of the slip ring, the wireless method involved no contact, and therefore resolved the associated issues. In addition, increases in the computational speed of the CPU of the receiver helped to enable measurements within the actual range of engine speeds. The noise resistance of this method was low compared to the contact method, but insulating materials were used to protect the receiver.

Figure 10 shows an example of measurements taken in a hollow crankshaft.

Because in a hollow crankshaft the hydraulic pathway does not have to pass the center of rotation of the crankshaft, the decline in hydraulic pressure downstream (due to pressure loss in internal pipes and pressure loss due to Coriolis force), a constant issue in central oil supply systems, was minimized and the average hydraulic pressure increased. The effect of the hollow crankshaft was estimated in simulations, but pressure measurements in the internal hydraulic pathway of the crankshaft verified the effect. In addition, the effect of noise was minimized, and a good understanding of the differences between specifications was obtained.

2.4. Linking Method

The use of the linking method to conduct measurements is not a new technology, and is used in mass production engines. However, in high-speed Formula One engines, the acceleration component of the inertial force on the links is 10000 G at 19000 rpm, approximately twice the force in a mass production engine. This difference necessitates high-stiffness and ultra-low weight links, and therefore significantly affects the design of the linking mechanisms.

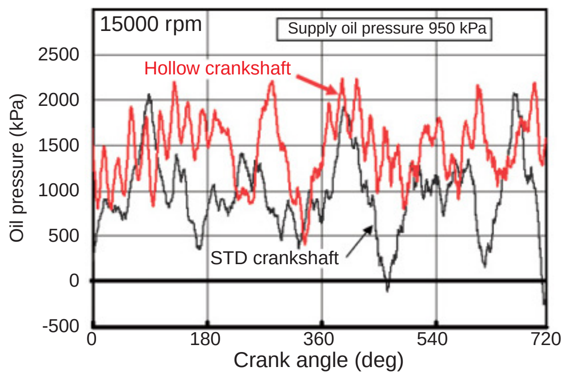


Fig. 10 Pressure in hydraulic pathway

Figure 11 shows a linking mechanism used in a Formula One engine.

For Formula One engines, the linking mechanisms were incorporated in the single cylinder engines used for analyses.

To reduce weight, the linking mechanisms were cut from aluminum alloys. The covers used to hold the lead wires were all produced from CFRP in order to further reduce weight. In addition, to maintain stiffness, a box section was used for the sectional shape of the linking, helping to ensure a secondary moment acting on the section. Small NTN bearings with an internal diameter of 7 mm and an external diameter of 11 mm were employed in the connections to help ensure a smooth sliding motion.

In conrods, the end connected to the piston, which displays a linear reciprocal motion, is called the small end, and the end connected to the crankshaft, which displays a rotary motion, is called the big end. The two ends are joined by the shaft. Because the shafts of the conrods are constantly subject to complex forces representing a combination of linear and rotary motion, stress concentrates in this section, and the magnitude of this stress can exceed the breakage limit of the shaft material. Conrod shaft breakages occurred during Honda's third Formula One era, and the reliability of the conrods formed an important development agenda.

Figure 12 shows an example of a conrod breakage, the result of a CAE analysis, and the set-up used in the linking method.

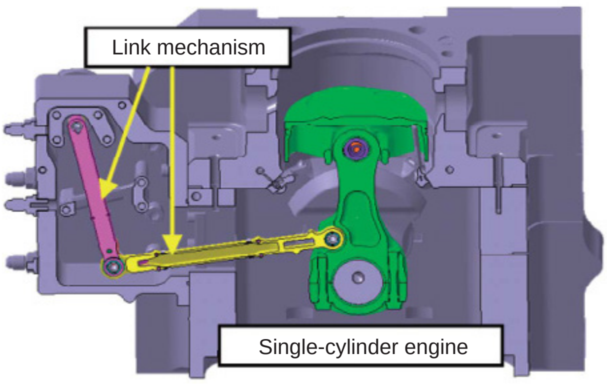


Fig. 11 Link mechanism for Formula One engines

The point of breakage of the conrods was close to the small ends, where strength was lower. To analyze these breakages, CAE analysis was employed to suggest potential reasons for the concentration of stress. Following this, the development of a measurement method using the linking method proceeded, and analysis of the phenomenon in single cylinder engines commenced in 2004.

Figure 13 shows the results of measurements of conrod torsional resonance using the linking method. The measurement results showed that torsional resonance was produced in the conrods at full engine loads. In addition, the fact that resonance did not occur when no load was acting and the inertial force was therefore dominant indicated that the conrods were twisting due to excitation by gas pressure and bending of the crank pins.

As this demonstrates, the linking method as used in Formula One analyses demonstrated a sufficient level of accuracy even in measurements at 19000 rpm, and the link mechanism also displayed sufficient durability. This made it possible to understand internal engine phenomena such as reciprocating motion, and technologies for the quantification of these phenomena contributed in numerous ways to engine development.

2.5. Friction Measurement Using the Floating Liner Method

The friction generated in the various parts of the engine is one of the factors that impedes the achievement of increased power, and the accurate measurement of friction is therefore an important issue. Friction in the reciprocating parts represents approximately 60% of total engine friction, and its mechanism has still not been clarified. This section will discuss the floating liner method, a new method of measuring friction in the reciprocating parts.

Broadly speaking, friction in the reciprocating parts can be divided into the following three types:

(1) Sliding friction between the pistons and the sleeves
(2) Sliding friction between the conrod bearings and the crank pins
(3) Friction due to agitation of the oil in the crankcase

Of these, the floating liner method was developed to measure (1), sliding friction between the pistons and the sleeves.

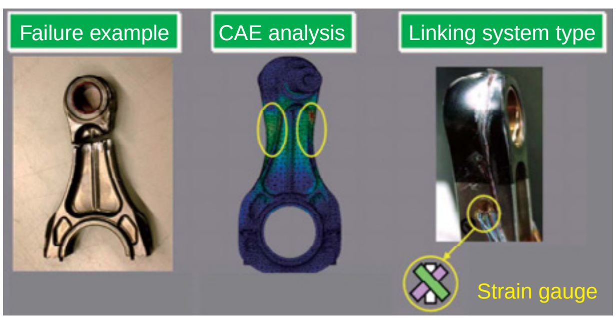


Fig. 12 Failure example, CAE result, linking system type

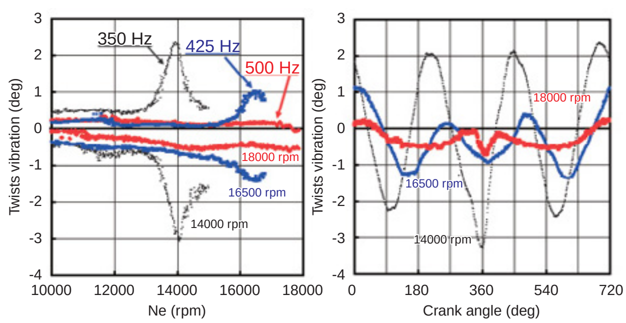


Fig. 13 Twist vibration measurement results for connecting rod

Figure 14 shows a cutaway of the test apparatus used in the floating liner method.

The feature that sets this apparatus apart from standard floating liner test apparatuses is the fact that it has been designed for use at high speeds. Standard test apparatuses are constructed of iron alloys or aluminum alloys, and they have a range of measurement of up to 5000 rpm. However, at high engine speeds the liner resonates in the same direction as the motion of the pistons, and measurement accuracy declines. To resolve this issue, the liner in the new apparatus was constructed from beryllium in order to reduce its weight and control the occurrence of resonance. In addition, the load sensor was positioned at the top of the liner to help ensure stiffness in the mounts. This positioning prevented the mechanism from being used for the measurement of firing, but in a later specification, combustion pressure was artificially produced to simulate firing conditions. Using a mechanism to introduce air to the combustion chamber, the pistons could be used to compress the air, making it possible to freely increase pressure up to 10 MPa.

Figure 15 shows an example of measurements using the initial apparatus specifications.

The graph on the left shows figures from the load sensor in the up-down direction (the direction of piston motion). Adjustment of the crank angle made it possible for the floating liner method to be used to analyze friction in each stroke. The integral of each rotation corresponded to the sliding friction between the piston and the sleeve in each cylinder, making it possible to quantify sliding friction.

The graph on the right shows figures from the load sensor in the left-right direction (the direction of the rotating surface of the crankshaft, orthogonal to the direction of piston motion). These results show the changes in thrust and anti-thrust force for each stroke.

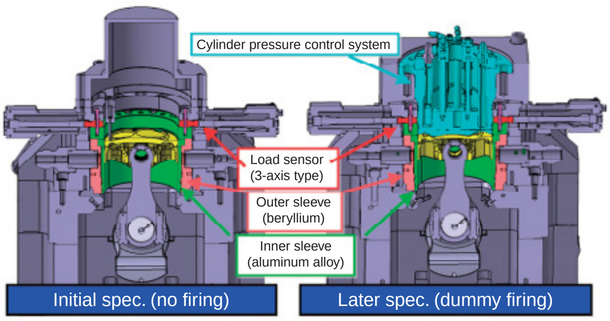


Fig. 14 Floating liner mechanism

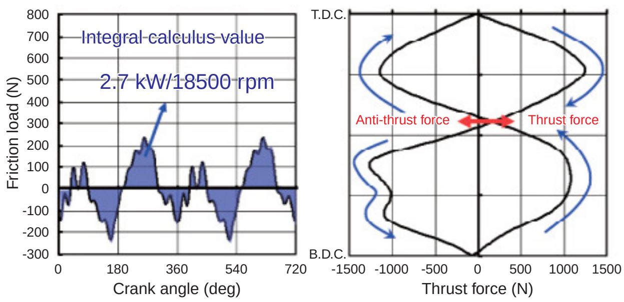


Fig. 15 Sliding friction measurement result

As the discussion above indicates, it became possible to measure the friction in each stroke and use the results in analyses of the form of the piston skirt and other parameters.

2.6. Visualization Technologies

2.6.1. Fuel spray visualization technology

Due to the latent heat of vaporization, the atomization of the fuel cools the intake air, increases filling efficiency, and advances combustion. These effects result from the droplet diameter and the penetration, which change with the fuel injection pressure, but a measurement method for quantifying these parameters was previously unavailable.

The fuel injection pressure of the electronically-controlled fuel injection system (PGM-FI) used in the turbocharged engines employed during Honda's second Formula One era was 0.25 MPa. Following this, a 1.2 MPa high-pressure system was developed in 1987 and used for 17 years until 2004, in Honda's third Formula One era. From 2005, the increase in pressurization was accelerated and a 5 MPa system was used, with a 10 MPa system following in 2006. This section will discuss the fuel spray visualization technology used in the analysis of droplet diameter and penetration in this process of increasing pressure.

The fuel spray has a three-dimensional form that changes with the elapse of time. The following parameters were focused on:

(1) Droplet diameter
(2) Diffusion
(3) Penetration
(4) Form angle and form

Of these, the form angle and form could be controlled to a certain extent by means of the direction of the nozzle, but droplet diameter, diffusion, and penetration are affected by the shape of each injector hole, and a number of nozzle shapes was therefore used in measurements.

Figure 16 shows an external view of the measurement device, a phase Doppler particle analyzer (PDPA), and the measurement principle.

The PDPA bombarded the fuel spray with a continuous laser beam, and measured the droplet diameter and flow speed by applying signal processing to the scattered light. The flow speed was measured at the point where the two laser beams intersected.

Fuel particles entered this point, and the scattered

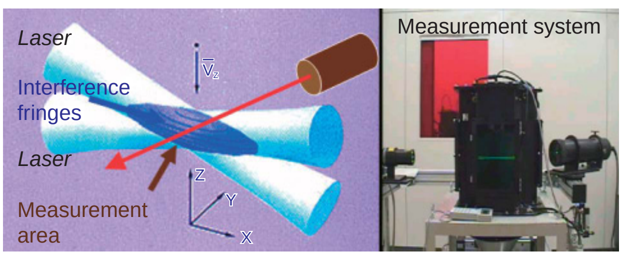


Fig. 16 External view of PDPA hardware and measurement principle

light was captured by a receiver and transformed into an electrical signal. Using a fast Fourier transform (FFT), the frequency was analyzed and the speed of the particles was calculated.

Droplet diameter was measured using the phase difference between burst signals monitored by multiple photomultiplier tubes. The droplet diameter was calculated by correcting the frequencies of the light that was diffracted differently, depending on the size and shape of the droplets.

Figure 17 shows the distribution of droplet diameters in the fuel spray, as measured using the PDPA.

To help enable detailed analysis of the PDPA measurement results, the form of the fuel spray directly underneath the injector holes was visualized using the SprayMaster magnified imaging apparatus manufactured by LaVision GmbH.

Figure 18 shows a fuel spray as imaged using SprayMaster.

Using SprayMaster, transmission images of the cross-sections of sprays were captured using a laser, and employed in parameter comparisons of fuel injection pressure and injector holes, and comparisons of the characteristics of the sprays produced by injectors with different numbers of holes.

The SprayMaster apparatus contributed to the achievement of increased efficiency in the development of fuel systems, assisting in tasks including the determination of specifications in injector development and the simulation of in-cylinder fuel distribution.

2.6.2. Optical engine technology

Understanding the flow and turbulence phenomena of the air-fuel mixture in the combustion chamber is important to the realization of rapid combustion. The optical engines were used to enable measurement of flow and turbulence in the combustion chambers of Formula One engines.

This section will discuss the engine visualization method, measurement results, and validation results of simulation.

Figure 19 shows a cross section of the optical engine.

As in the case of the linking and floating liner methods discussed above, a single cylinder engine was used for the optical engine from considerations of ease of modification and versatility. A special sleeve made from silica glass was fitted to part of the cylinder block. A glass sleeve height of 40 mm made it possible to visualize the entire cylinder stroke.

Particle image velocimetry (PIV) was used for the measurement of the flow motion of the gas in the combustion chamber. The system utilized a YAG laser with a second-order harmonic of 532 mm, and a cross-correlation CCD camera for imaging. A synchronizer was used to synchronize the two devices.

Ultra-lightweight tracers, which are hollow resins with a diameter of 40 µm or less, were used for the visualization of flows. FFT cross-correlation was used in the analyses.

Special piston with extended crown was used, and measurements were performed at engine speeds of 10000 rpm.

Figure 20 shows PIV measurement results and simulation results.

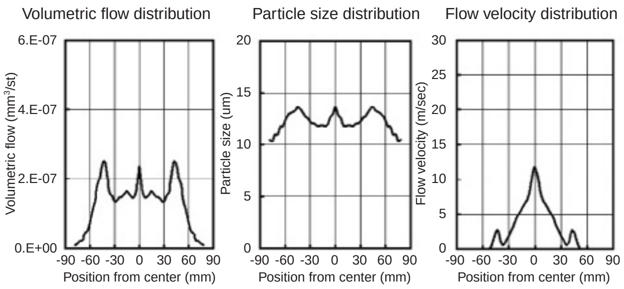


Fig. 17　Fuel spray measurement results

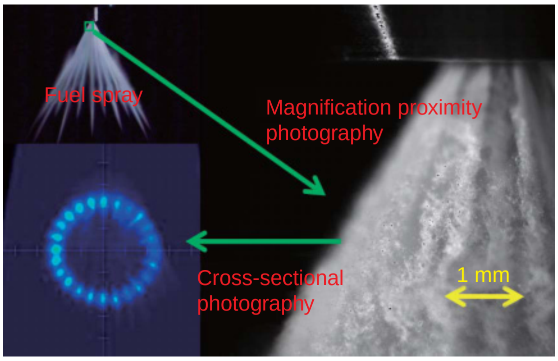


Fig. 18　Macrophotography using SprayMaster

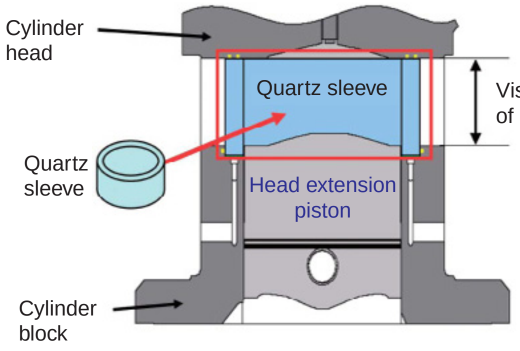


Fig. 19　Visualization engine

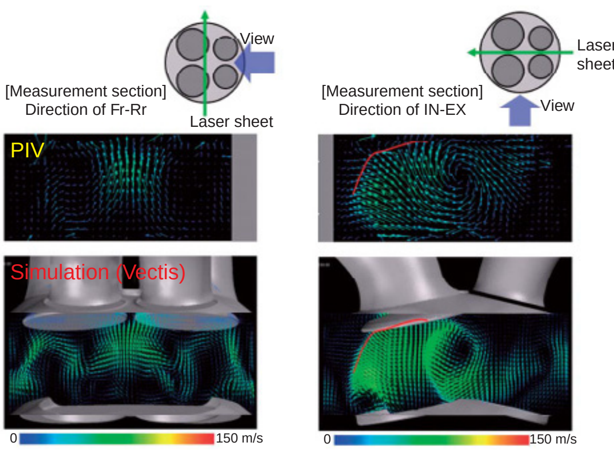


Fig. 20　Measurement result and simulation result

PIV measurements helped to enable the phenomenon of gas flow in the combustion chamber to be understood. Based on the results of visualization, validation of the gas flow simulation software VECTIS made by Ricardo plc was performed. Thereby, it became possible to accurately predict the flow motion of gas in the combustion chamber using the simulation. In addition, by performing similar validations, analysis of the behavior of fuel spray and the process of combustion (flame propagation) in the combustion chamber can be conducted using VECTIS, and the efficiency of development enhanced.

3. Conclusion

This paper has discussed measurement technologies employed in the engine development program in Honda's third Formula One era. The vast majority of these technologies were unavailable during Honda's second Formula One era, and it is no exaggeration to say that they provided the basis for the evolution of Honda's Formula One engines during the third era. The authors believe that the fusion of the measurement technologies fostered by the Formula One development process with the measurement technologies employed in the development of mass production engines, will assist in the development of environmental technologies and in increasing the efficiency of the development process.

Reference

(1) Shimasaki, Y., Sakaguchi, J., Maki, H., Suzuki, H., Kato, A., Nishizawa, K.: Study on Combustion Monitoring System for formula One Engines Using Ionic Current Measurement, Honda R&D Technical Review, Vol. 17, No. 1, p. 56-64

■ Author ■

Nagao YANAGISAWA

Naoki HANADA

Takeru HAMAKAWA

Kazushi OGIYAMA

Tetsuo GOTO

Development of High-Pressure Fuel Supply System for Formula One Engine

Tetsuya TANAHASHI* Kazuji ONO* Masanori HAYAFUNE*
Yosuke SAWADA* Atsushi SHIMIZU*

ABSTRACT

Important factors in boosting the performance of today's Formula One engines include: the realization of the formation of ideal air-fuel mixtures and the achievement of greater combustion efficiency, through the use of shorter fuel injection periods and increased spray atomization resulting from higher fuel pressures; and, in addition to this, the achievement of stable combustion in the low-load operating range.

A comprehensive analysis of injector spray characteristics was conducted, leading to the development of a Honda-made high-efficiency, high-pressure fuel supply system. This enabled the achievement of a 15 kW increase in engine power.

1. Introduction

The importance of development programs for fuel systems in automotive engines is increasing as a means of resolving technological issues related to increasing fuel efficiency and reducing exhaust emissions. Given this, automakers, parts makers, and research organizations are pushing ahead with the development of more advanced technologies, and are conducting more sophisticated analyses of relevant phenomena.

In Formula One engines, the fuel spray has a particularly significant effect on engine power, and optimum spray morphologies exist for specific combustion chamber shapes and intake port shapes. When specifications are changed in these areas, the fuel spray is also studied and redesigned.

Against this background, a basic principle of Honda's Formula One engine development program was the in-house production of fuel supply systems, in order to boost competitiveness through the ability to trace the development process and produce unique technologies.

This paper will provide a technological overview of a fuel pump for the supply of high-pressure fuel, a regulator to control fuel pressure, and an injector to inject the fuel and form a spray.

* Automobile R&D Center

2. Mechanism of Power Increase with High-pressure Fuel Injection

During the development of fuel systems for Formula One engines, a single cylinder engine was used to conduct analyses of intake air and fuel phenomena.

Based on the results of these analyses, the high-pressure fuel systems were in use in race engines until 2006, and further modifications of the spray morphology from 2006 onwards helped to enable an increase in power of approximately 15 kW.

Figure 1 shows changes in the fuel pressure and power of Honda race engines, and Fig. 2 shows an

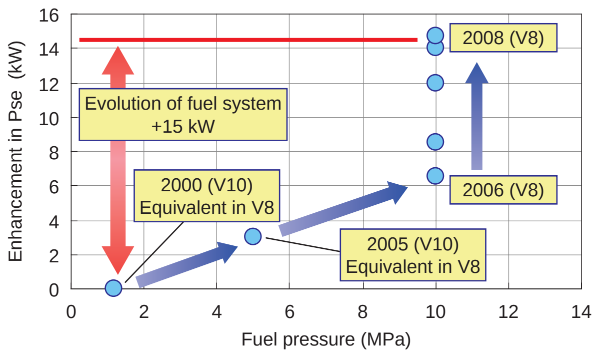


Fig. 1 Evolution of fuel system

overview of the components of a fuel supply system.

The following three mechanisms can broadly be indicated as those responsible for the increase in power achieved when the fuel pressure was increased from its initial value of 1.2 MPa to 10 MPa and the spray morphology was modified:

(1) Increased intake air cooling efficiency due to increased supply flow rate (shorter injection period)
(2) Enhanced combustion due to greater atomization of the spray
(3) Increased intake air cooling efficiency due to modification of spray morphology

2.1. Increased Intake Air Cooling Efficiency with Increased Supply Flow Rate

Under the initial specifications of a fuel pressure of 1.2 MPa and a supply flow rate of 43 L/h, the maximum injector supply flow rate was low because the fuel pressure was low, and a long injection period was necessary in each cycle. By contrast, the use of high-pressure fuel increased the supply flow rate and consequently reduced the injection period, helping to enable injection for an optimal injection period.

Figure 3 shows the relationship between the intake air velocity at the funnel tip and the injection period, as determined using simulations.

The results of this analysis show that the intake air cooling effect is maximized by ending the injection period by the timing of maximum blow at the funnel tip.

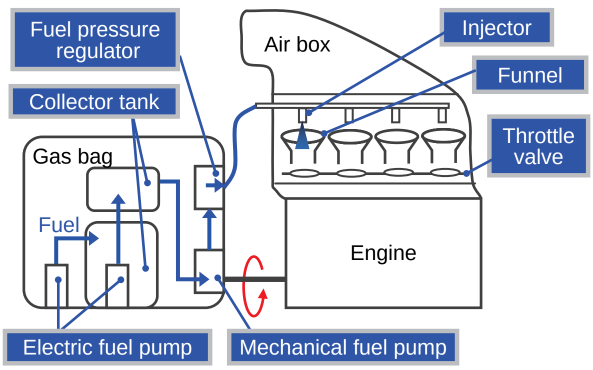


Fig. 2 Components of fuel supply system

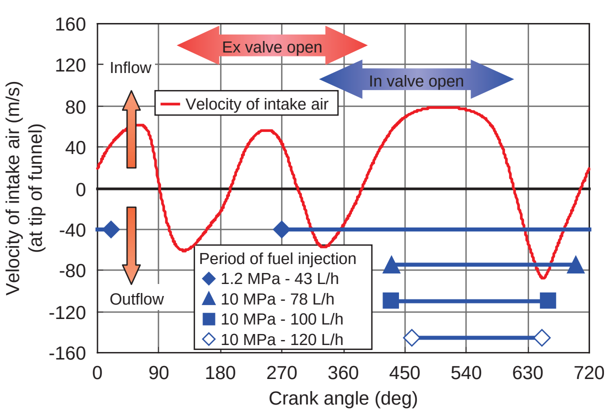


Fig. 3 Relationship between intake air velocity and injection timing

A decline in the temperature of the intake air can be determined from changes in the intake air pulsation. As Fig. 4 shows, a short injection period results in a phase delay in the intake air pulsation, and this can be judged as indicating that filling efficiency has increased due to a decline in intake air temperature.

2.2. Enhanced Combustion Efficiency with Greater Fuel Atomization

The diameter of the fuel droplets, which had been approximately 50 μm at a fuel pressure of 1.2 MPa, was reduced to 20 μm or less at a pressure of 10 MPa. As Fig. 5 shows, power was increased as a result of this refinement of the droplets. The mechanism behind this increase in power is thought to be increased mixing of the intake air and the fuel, and a reduction in the adherence of fuel to the port walls.

2.3. Increased Intake Air Cooling Efficiency with Modified Spray Morphology

As indicated above, the use of increased fuel pressure resulted in increased cooling air efficiency and increased combustion efficiency.

The fuel spray morphology was also studied in order to further boost these effects. Figure 6 shows changes in shaft output in a single cylinder test engine.

The increase in engine power that occurs when an increased number of injection holes are employed is due to increased atomization of the spray. The delay in the phase of intake air pulsation shown in Fig. 7 indicates that a decline in the temperature of the intake air has increased filling efficiency.

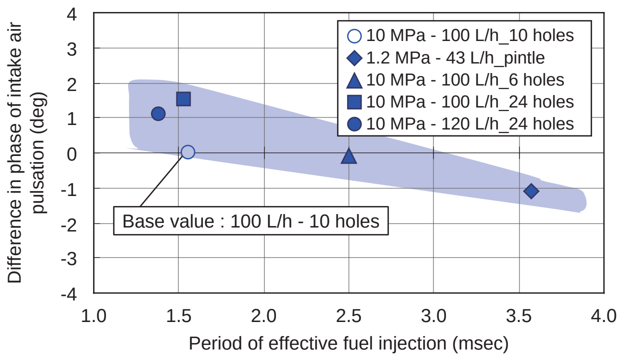


Fig. 4 Effect of injection time on difference in phase of intake air pulsation

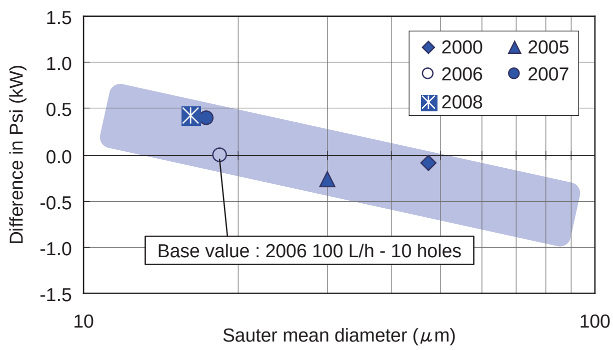


Fig. 5 Effect of droplet refinement on power

The relationship between penetration and the phase of intake air pulsation was also studied. As Fig. 8 shows, there is a peak in the intake air phase delay, indicating the existence of an optimum penetration for the achievement of increased intake air cooling efficiency. If the level of penetration is low, the spray will approach the center of the port due to the effect of the intake air. If the fuel is carried too high, the amount of fuel adhering to the walls of the port will increase.

A technological overview of the high-pressure fuel system based on the results of the analyses discussed above will be provided below.

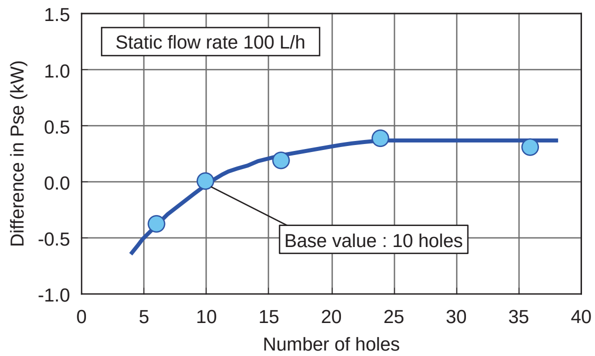


Fig. 6 Effect of number of holes on power

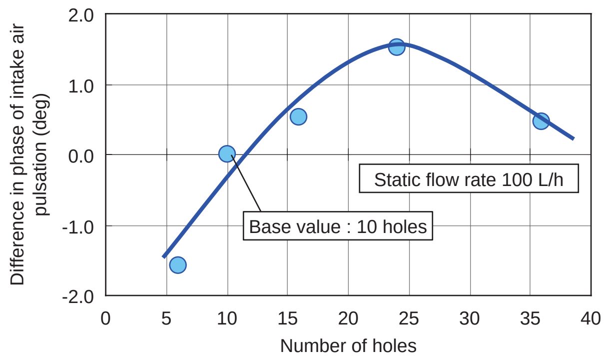


Fig. 7 Effect of number of holes on difference in phase of intake air pulsation

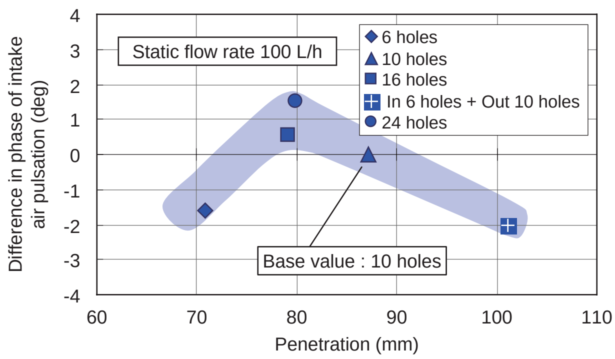


Fig. 8 Effect of penetration on difference in phase of intake air pulsation

3. Development of High-pressure Fuel Pump

In Honda's second Formula One era, the engine fuel supply system was composed of a gear pump producing a fuel pressure (PF) of 1.2 MPa, a fuel pressure regulator, and a side-feed injector, which supplied fuel from its side. At the commencement of engine development for Honda's third Formula One era, this basic configuration was carried over, with only the form of the injector being changed, to a top-feed type that supplied fuel from the upper section of the engine. Development and deployment of parts continued with consideration of their suitability to each model year engine.

Later, from 2005, the focus was shifted to the development of a system that helped to enable the achievement of increased intake efficiency and enhanced combustion by means of refining the fuel spray droplets from the injector and increasing atomization, and reducing the injection period, by increasing the pressure of the fuel supply system.

The achievement of a fuel pressure of 10 MPa was established as a target for the opening race of 2006. In the 2005 season, a commercial 5 MPa system had been used, and development time was concentrated on the achievement of 10 MPa.

In order to realize increased engine power by achieving a PF of 10 MPa, it would be essential to discard the previous gear pump and to conduct an in-house development of a high-efficiency and lightweight mechanical fuel pump that would be able to generate the necessary supply flow rate at high pressure. The reason for this was that at high fuel pressures, the efficiency of the previously used gear pump type would decline due to fuel leaks from the tips of the gear teeth and the gaps between the gear side faces, and it would be unable to transport the necessary amount of fuel.

To resolve this issue, the goal was initially to develop a gear pump configuration employing movable side plates to maintain fixed gaps between the gear side faces and moderate the volume of fuel leakage. However, this represented a technological challenge from the perspective of durability, raising issues such as the seizing of the sliding surfaces of the gear side faces. With the achievement of high fuel pressure and high efficiency as targets, efforts were therefore focused on the development of a new plunger piston-type fuel pump.

3.1. Overview of High-pressure Fuel Supply System

Figure 9 shows a diagram of the fuel system fitted in the fuel tank, incorporating a mechanical high-pressure fuel pump, a fuel pressure regulator, and primary and secondary electric fuel pumps. Formula One vehicles use explosion-proof fuel tanks (called "gas bags") manufactured from plastic liners. These tanks are fitted in the monocoque, and each year they are manufactured in conjunction with the monocoque, to dimensions that match the capacity of the monocoque. The collector tank helped to prevent abnormalities of suction of the gas bag fuel from arising in the electric pumps due to turning or acceleration and deceleration G-forces, and pressurized the fuel to feed pressure for the mechanical high-pressure fuel pump. The collector tank was divided into two parts, a high-pressure collector tank and a low-pressure collector tank. Fuel from the gas bag was sent to the low-pressure collector tank by the primary electric fuel pumps positioned to the left and right on the floor of the gas bag. The fuel was then pumped from the low-pressure collector tank to the high-pressure collector tank by the secondary electric fuel pumps positioned on the floor of the low-pressure collector tank. The high-pressure collector tank employed a bladder, which was a variable volume pressurizing device using air pressure, to help ensure that the feed pressure of the fuel did not fall below the necessary level for supply to the mechanical high-pressure fuel pump. When no fuel was present in the high-pressure collector tank, the bladder swelled to fill the entire tank; when the tank was filled with fuel, the pressure of the fuel compressed the bladder. By means of this mechanism, the high-pressure collector tank could be constantly filled with fuel. The high-pressure collector tank was also provided with a pressure relief valve (PRV), which helped to ensure that the pressure in the tank remained at a constant upper limit. The bladder and the PRV stabilized the feed pressure to the mechanical high-pressure fuel pump, and helped to prevent cavitation when the high-pressure pump was taking in fuel at high temperatures.

3.2. Mechanical High-pressure Fuel Pump Configuration

Figure 10 shows an external view of the fuel pump produced by this development project. The initial specifications for the mechanical high-pressure fuel pump featured three units: a reduction gearbox to decelerate the pump against engine input, a gear pump to maintain the feed pressure for the plunger pump, and a plunger pump for high-pressure fuel supply. Later, in order to reduce weight, the reduction gearbox and the gear pump were integrated to produce a reduction gear pump. The rotational input from the engine was decelerated by the drive gears, while the gear pump supplied fuel. This helped to reduce the total length of the pump by 30 mm and its weight by 340 g. The reduction ratios of the reduction gearbox and the reduction gear pump were optimized against engine speed in order to balance durability with good flow performance.

3.3. Selection of Pump Configuration

The pump was provided with a gear pump to help ensure stable feed pressure to the plunger pump. It was essential for the gear pump both to be small and lightweight and to function to prevent cavitation of the transported fuel when operating at high fuel temperatures in a circuit driving. A shape was designed that reduced pressure loss in the pump inlet port and an optimal gear pump thickness was set in order to reconcile the achievement of the desired supply flow rate with the realization of cavitation toughness. In addition, the optimum plunger piston size was selected and the resistance of the fuel channels was reduced in order to increase fuel transport efficiency, and a variety of modifications were made, including optimization of the stiffness of the pump body. As Fig. 11 shows, these measures resulted in the achievement of a level of total efficiency of 80% in the pump. As Fig. 12 shows, this total efficiency level was well beyond that of mass production pumps.

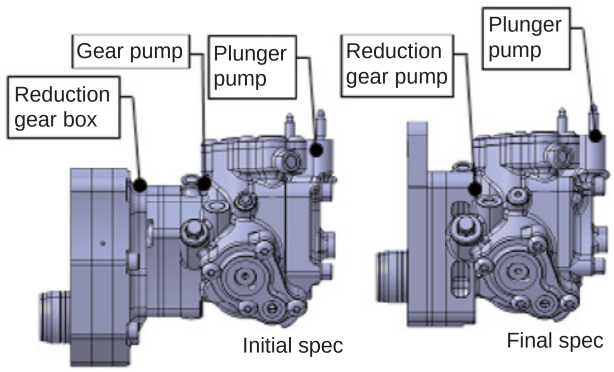


Fig. 10 High-pressure fuel pump

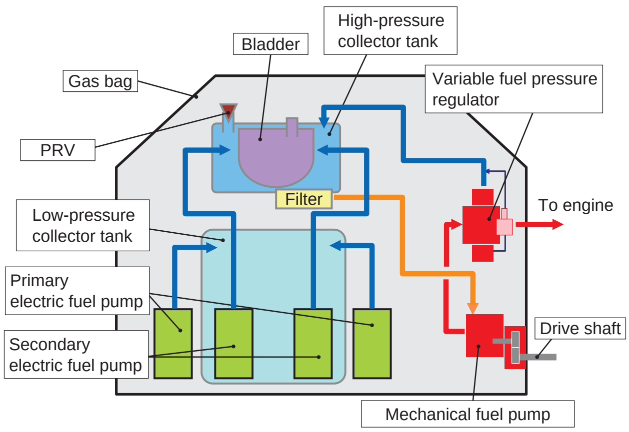


Fig. 9 Fuel system for Formula One engine

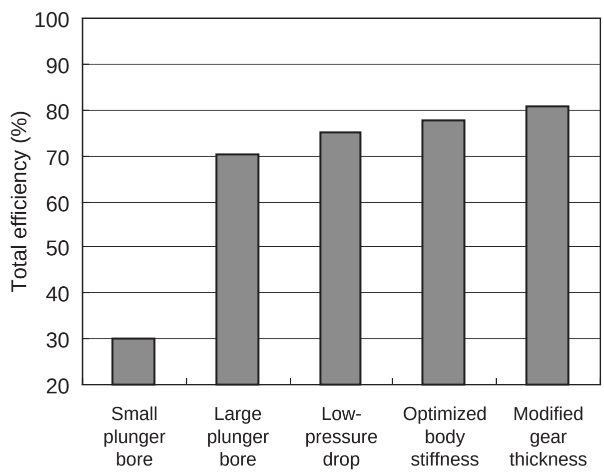


Fig. 11 Fuel pump characteristics

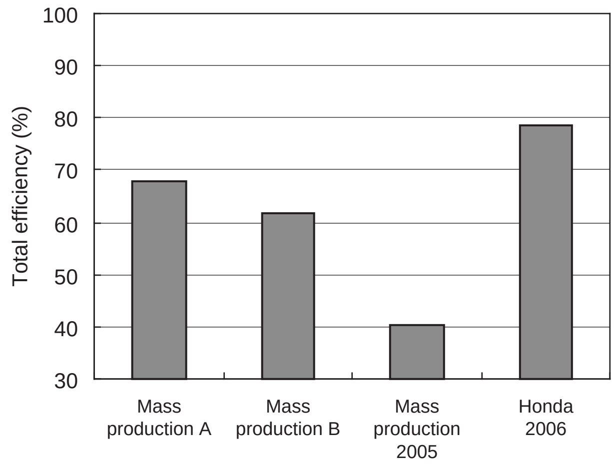


Fig. 12 Fuel pump characteristic comparison

4. Development of Variable Fuel Pressure Regulator

The aim of fuel supply system development for the 2006 season was to increase power by realizing a shorter fuel injection period through an increased injector supply flow rate. This presented the technological issue of the challenge of controlling the injector supply flow rate at high fuel pressures in the low load and idling ranges, ranges in which ultra-low supply flow rates are used. This led to the development of a variable pressure regulator using diaphragm back pressure control, which was able to reduce fuel pressure in the low load and idling ranges, and to increase fuel pressure in the high load range in order to maximized engine power, as a means of controlling the supply flow rate of the high-fuel-pressure, high-supply-flow-rate injectors.

4.1. Variable Pressure Regulator Configuration

Figure 13 shows the configuration and the operating principle of the variable pressure regulator. The variable pressure regulator was based on a pressure regulator using a diaphragm for return. The unit was provided with a back pressure spring chamber (“back pressure chamber” below) that applied back pressure to the regulator diaphragm. Forced application of fuel pressure to the back pressure chamber helped to enable the pressure difference of the diaphragm to be maintained at a constant level while the pressure adjustment function responded to higher fuel pressure. Fuel was supplied to the back pressure chamber via orifices in the fuel supply channels, and a fixed supply flow rate was maintained. A small needle valve regulator was used to hold the back pressure chamber fuel pressure constant. Because the small needle valve regulator controlled only an ultra-low supply flow rate, an orifice was positioned in front of it to help ensure that it was able to adjust pressure.

Pressure control in the variable pressure regulator operated as follows: When the regulator was switched to the low-pressure side, a solenoid valve was opened and the pressure was released from the back pressure chamber; pressure was adjusted using the pressure difference of the diaphragm, as generated using spring power alone. When the regulator was switched to the high-pressure side, the solenoid valve was closed and fuel was supplied to the back pressure chamber.

4.2. Variable Pressure Regulator Performance

The main performance demand on the variable pressure regulator was responsiveness when switching fuel pressure between high and low pressures. If fuel pressure switching performance was unsatisfactory, the actual injector supply flow rate would become unstable, the engine combustion state would vary and air-fuel control would become challenging, resulting in unstable engine output. To avoid these issues, the responsiveness for switching from the low-pressure to the high-pressure side was set to within 500 ms, and the responsiveness of switching from the high-pressure to the low-pressure side was set to within 20 ms.

Switching responsiveness from the low-pressure to the high-pressure side was set via the size of the orifices in the fuel channels of the variable pressure regulator. Responsiveness from the high-pressure to the low-pressure side was set via the supply flow rate of the electronically-controlled solenoid valve used for pressure switching. The desired responsiveness was achieved through optimization of these specifications, helping to enable the achievement of a stable air-fuel ratio (Fig. 14).

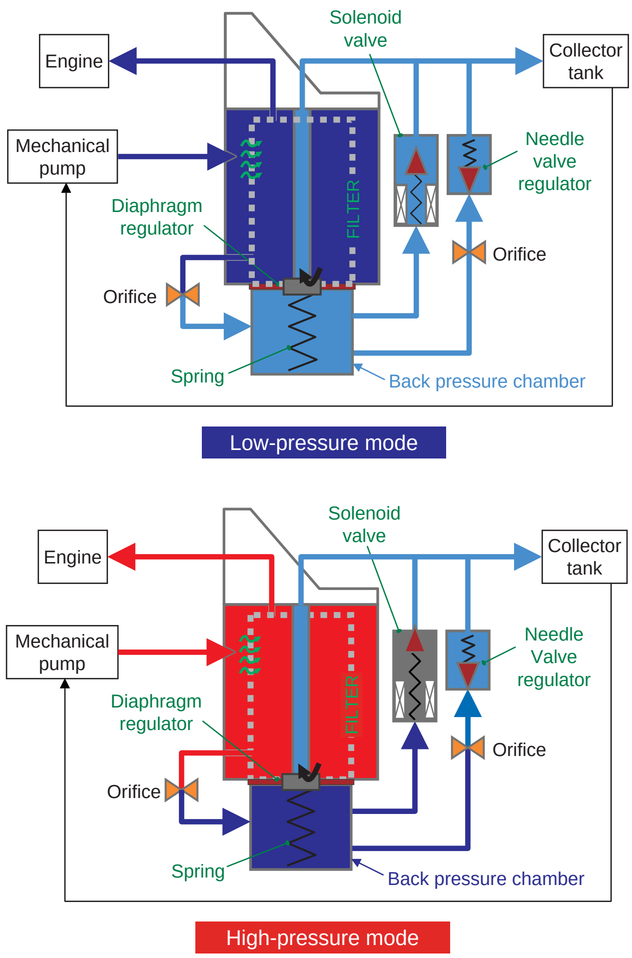


Fig. 13 Variable fuel pressure regulator system

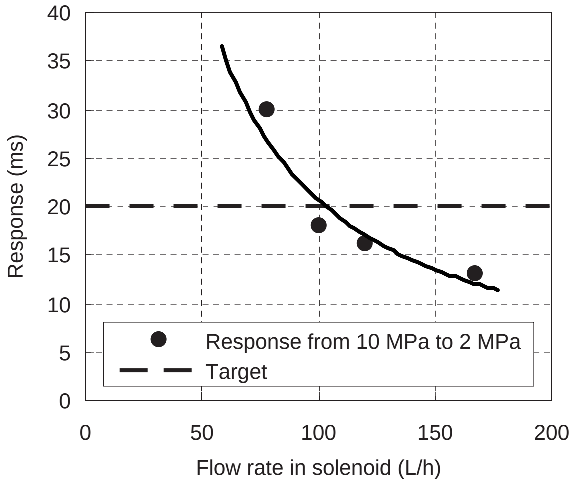


Fig. 14 Fuel pressure response characteristics

5. Evaluation of Durability of High-pressure Fuel System

The durability of the high-pressure fuel pump and the variable pressure regulator under actual operating conditions was verified. Unit tests were conducted to test durability at the pump speed frequency during mode operation. Dedicated pump, regulator, and injector unit test equipment that was able to reproduce circuit driving modes using an identical engine control computer to the computers employed in actual vehicles was used to conduct the evaluation.

The important points of focus in terms of fuel pump durability included body stiffness, lubrication performance, and the toughness of each constituent part; for the regulator they included body stiffness and the toughness of the plastic internal parts against swelling.

Focusing on these factors, sufficient durability was achieved to realize a maintenance interval making it possible to use the pump for multiple races.

6. Injector Development

6.1. Injector Specifications

As Table 1 shows, the injectors employed by Honda in races until 2008 can be grouped into five phases. This section will follow these groupings in discussing injector development.

6.1.1. Phase 1 (2000-2003)

Figure 15 shows the injector layout. The injectors were positioned at the top of the intake funnels (these will be termed “top injectors” below). Fuel was injected at a pressure of 1.2 MPa, and pintle-type single-hole nozzles produced a conical spray.

6.1.2. Phase 2 (2004)

To increase intake efficiency and responsiveness in the transient range in this phase, injectors were positioned close to the intake valves (“near valve injectors” (NI) below), to provide, with the top injectors, two injectors per cylinder. Because the NI were positioned close to the valves, the use of multi-hole nozzles helped to enable the realization of sprays from two directions directly aimed at each port. Because it was necessary to accurately control an ultra-small flow, the supply flow rate for the NI was set lower than that of the top injectors.

Table 1 Injector type

	Phase 1	Phase 2	Phase 3	Phase 4	Phase 5
Year	'00 - '03	'04	'05	'06 - '07	'08
Inj layout	Top	Top + NI	Top + NI	Top	Top
Fuel pressure (MPa)	1.2	1.2	5	10	10
Static flow (L/h)	43	43 (NI:33)	55 (NI:55)	78, 100	100
Hole type	Pintle	Pintle (NI:Multi)		Multi	Multi
Number of holes	–	(NI:6)		6, 10, 24	24

6.1.3. Phase 3 (2005)

In this phase, a system using a fuel pressure of 5 MPa was introduced in order to boost engine performance by increasing the atomization of the spray through the use of high-pressure fuel injection. The Honda-made NI was modified for high fuel pressure use, but commercial units were employed for the top injectors, pump, and regulator.

6.1.4. Phase 4 (2006-2007)

The Phase 3 NI was modified for use with a fuel pressure of 10 MPa, and was employed in races. From 2006, regulations set an upper limit of 10 MPa for fuel pressure, and also stipulated that only one injector could be used per cylinder. Given this, development was conducted to achieve the effect of the NI using only a top injector, and the effect of the spray on engine response was added as spray selection criteria in the injector spray development process.

To boost engine performance, a development program was conducted during the season to help enable the use of a shorter injection period by increasing the injector supply flow rate and generating further atomization of the spray by increasing the number of injector holes. The

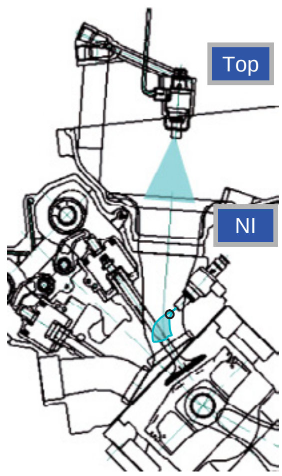


Fig. 15 Injector layout 2

final spray configuration in Phase 4 was a conical spray using 24 holes. In addition, because the fuel pressure regulations had increased the importance of the spray characteristics, research on new methods of evaluation of these characteristics and study of next-generation injector specifications was commenced.

6.1.5. Phase 5 (2008)

Using spray measurement and evaluation methods developed in the previous year, the effect of differences in spray on actual engine performance was analyzed, and a concept of the optimum spray was formulated.

This concept was reflected in a new injector design, and the injector was employed in races from the opening race of the 2008 season. Regulations had prohibited any changes to injector specifications that would boost output performance, but modifications were made as needed during the season to increase reliability.

The next section will discuss the Phase 5 specifications, the final injector specifications used by Honda in Formula One.

6.2. Injector Development Concept

6.2.1. Reduction of size and weight

A reduction in the size of the injectors would reduce their weight, increase the degree of freedom of layout, and also help to enable the delivery pipes and other parts to be reduced in weight. The achievement of weight savings in the injectors positioned at the top of the engine and in fuel system-related parts would contribute to lowering the center of gravity of the engine.

6.2.2. Modification of supply flow rate characteristic

In the high engine speed environment (maximum: 19000 rpm) of a Formula One engine, it is necessary to promote the dispersion of the fuel spray and achieve an intake air cooling effect by injecting a large quantity of fuel in a short time period.

In addition, stable injection is also necessary when ultra-low quantities of fuel are injected, when the engine is idling or off-throttle. The achievement of increased responsiveness in the injector needle valves was focused on in order to achieve a fuel spray characteristic that satisfied these performance demands.

6.2.3. Modification of spray characteristic

A balance between the form angle, droplet diameter, dispersion, and penetration is necessary for the spray characteristic. However, increased atomization is in a trade-off relationship with dispersion and penetration, and an increase in atomization will cause a decline in dispersion and penetration, and vice versa. The previous injectors had reached the limit values for each parameter, making the realization of enhanced performance an urgent issue.

Against this background, the regulations had established an upper limit for fuel pressure. In order to efficiently utilize the energy of the fuel pressure in the spray characteristic, the actual fuel pressure on the holes (the hole plates) in the injectors (termed "plate feed pressure" below) was focused on (Fig. 16). The reduction of fuel pressure loss in the injectors increased the plate feed pressure, helping to enable the realization of a spray characteristic equivalent to that of an actual increase in fuel pressure.

Table 2 Injector specifications 1

Year	2007	2008
Length (mm)	57	48
Diameter (mm)	20	16
Weight (g)	65	38
Number of coil windings	268	100
Coil resistance (Ω)	2.8	1.0
Operating current (A)	Peak: 2.4 Hold: 0.6	6.0

6.3. Developed Injector Technologies and Performance

6.3.1. Reduction of size and weight

Figure 17 shows an external view of an injector. A new magnetic material developed for use in the injector (cobalt steel) was employed to reduce the size and weight of the units. In addition, the structure of the needle valves was modified, helping to enable the total length of the valves to be reduced. Because the size of the injectors was determined by the number of coil windings, the number of windings was also reduced.

However, reducing the number of coil windings would result in a decline in magnetic attraction. The magnetic circuits were therefore optimized and drive current increased in order to increase attraction.

Table 2 shows the size of the injectors and the coil specifications.

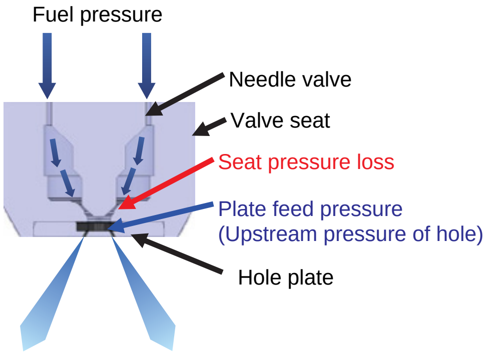


Fig. 16 Injector internal fuel flow (2007 model)

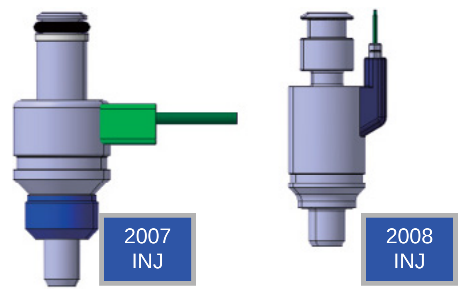


Fig. 17 Injector body shape

6.3.2. Supply flow rate characteristic

Without variable fuel pressure control, the 2007 injectors were not able to conduct stable fuel injection from idling to wide-open throttle. A development project was therefore initiated to produce a new injector that would be able to inject fuel across the operating range under high pressure. In order to realize this goal, the weight of the needle valve (a movable part) was reduced; the magnetic characteristic was enhanced by modifying the magnetic materials and magnetic paths; the rise in magnetomotive force was increased by the increase in the rise in current achieved by reducing the number of coil windings (reducing the resistance value); and the responsiveness of the needle valve rise was enhanced by means of the increase in the magnetic attraction of the valves achieved by increasing the drive current. The spring set load could also be increased by the amount of the increase in magnetic attraction, helping to enable an increase in needle valve fall responsiveness.

In addition, the drive current waveform was reexamined and the linearity of the fuel supply flow rate against current was enhanced. As a result, when the injector spray characteristic was equivalent to the 2007 model, the injector was able to inject fuel at high pressure across the entire operating range.

However, the needle valve stroke and the diameter of the valve seats were increased in order to boost the plate feed pressure, with the aim of enhancing the spray to increase power. This made control of the ultra-low supply flow rates at low loads and in the idling range a challenge, and as a result, the use of variable pressure control was continued.

This was the result of a comparative examination of the effect on engine performance of enhancing the spray characteristic and the effect on actual vehicle performance (weight savings) of discarding the variable pressure control system. It was determined that the enhancement of the spray characteristic was a more important consideration.

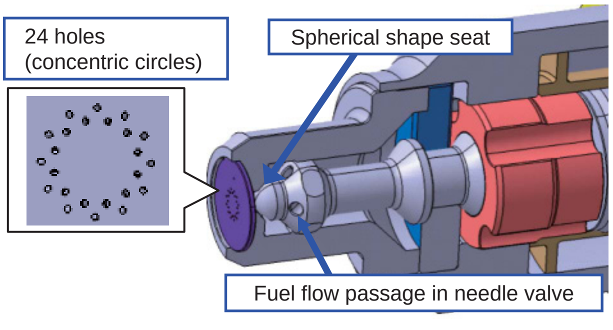


Fig. 18 Injector internal structure (2008 model)

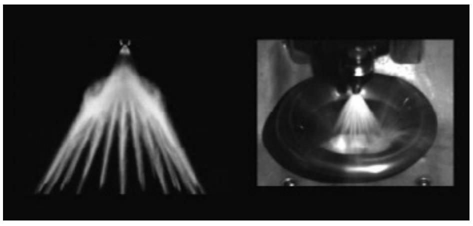


Fig. 19 Atomization achieved by 24-hole injector

Table 3 Injector specifications 2

Year	2007	2008
Static flow (L/h)	100	100
Angle of spray (deg)	53	53
Seat diameter (mm)	ϕ 1.30	ϕ 1.48
Valve stroke (mm)	0.10	0.15
Plate thickness (mm)	0.4	0.2
Number of holes	24	24
Hole diameter (mm)	0.100	0.090

6.3.3. Spray characteristic

The needle valve stroke and the diameter of the valve seats were increased in order to reduce the pressure loss resulting from the squeezing of the valves, resulting in increased plate feed pressure. Table 3 shows the points that were modified, and Fig. 18 shows the configuration of the injectors.

In addition, the fuel flow was modified by optimizing the shape of the internal fuel channels and employing spherical valve seats. These modifications increased plate feed pressure from 4.7 to 8.9 MPa at a fuel pressure setting of 10 MPa.

The enhancement of spray performance (droplet diameter, penetration, diffusion) helped to enable the achievement of a good balance between these parameters, which are in a trade-off relationship. The 24-hole conical spray configuration that was judged to be optimal in terms of vehicle performance based on the analysis in Section 2.1. was adopted (Fig. 19).

With regard to atomization, an average droplet diameter (SMD) of approximately 17 μm was achieved.

7. Conclusion

(1) Development of high-pressure pump and variable pressure regulator

A pump efficiency of 80% was achieved through a variety of measures, including reduction of pressure loss in the pump inlet port and optimization of the size of the plunger pistons.

In addition, the employment of a variable pressure regulator helped to enable the realization of high-supply-flow-rate injection, resulting in increased engine output and stable injection of ultra-low fuel supply flow rates at low loads.

(2) Development of high-pressure injectors

The use of new magnetic materials and the modification of the injector configuration and the drive current helped to enable the injectors to be reduced in size.

With regulations stipulating an upper limit on fuel pressure, the development aimed to increase combustion efficiency by means of atomization of the spray; reduction of internal pressure loss in the injectors helped to enable the achievement of a spray characteristic that produced a performance enhancement equivalent to that of an actual increase in fuel pressure.

(3) Engine performance

Increasing fuel pressure helped to enable an increase in the maximum injector supply flow rate; the resulting shorter injection period promoted dispersion of the fuel spray, and an intake air cooling effect was obtained. In addition, increased atomization promoted greater mixing of the intake air and the fuel, enhancing combustion efficiency. As a result of these measures, fuel system development produced an increase in power of approximately 15 kW against previous levels at the commencement of Honda’s third Formula One era.

■ Author ■

Tetsuya TANAHASHI

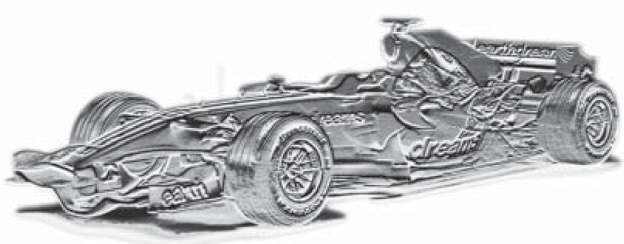
Kazuji ONO

Masanori HAYAFUNE

Yosuke SAWADA

Atsushi SHIMIZU

Descriptions of Gearbox Technologies



Development of Seamless Shift for Formula One Car

Takashi YOSHIOKA* Takeshi UCHIYAMA*
Katsumi KUBO* Ryo MATSUI*

ABSTRACT

Honda focused on gearbox development during its third Formula One era. The reduction of shift time is an effective means of maximally increasing race competitiveness within the constant-mesh 7-speed gearbox regulations. In the standard shift process, the current gear was disengaged, the system went into a neutral state, and the following gear was engaged. Honda’s seamless shift realized up-shift with a torque loss time of zero, by engaging the following gear and then disengaging the current gear. Normally, this process would lead to damage due to double engagement, but in the developed system double engagement was prevented and transmission of deceleration torque was enabled by adding one-way clutches with a locking function to the conventional shift mechanism. The selective use of these one-way clutches, positioned between the gear hubs and the mainshaft, and the use of cooperative control with the engine, enabled the realization of seamless shift across the entire shift range. As a result, lap time was reduced by 0.4 sec per lap, and the system was used in races from 2005 as the first shift mechanism of its type in the Formula One world.

This paper will also discuss the removal of the shift forks and shift rings as well as the fitting of the gear selection mechanism inside the mainshaft in order to reduce the total length and weight of the mechanism while maintaining its seamless shift performance.

1. Introduction

The shift mechanisms used in Formula One vehicles are constant-mesh parallel twin shaft types, in which gear stages are changed using dog clutches of the type frequently employed in motorcycles. In the project discussed in this paper, a shift mechanism was modified at the initial stage of gearbox development in order to boost the competitiveness of Formula One vehicles. The development aim established to realize this goal was to reconcile rapid shift (reduced shift time) with secure shift (durability and reliability). In order to realize rapid shift, acute chamfer angles were employed in the dog clutches, barrel inertia was reduced, and the speed of operation of the shift system was increased. The increase in shift speed generated a number of issues, including operational irregularities originating in shift fork overshoot and inclination of the shift rings. In order to prevent these issues, the forms of the barrel cams and the shift forks were optimized to stabilize shift operation. This helped to realize an equivalent level of competitiveness with the vehicles of other teams, but innovative technologies are essential to achieving victory in the fast-evolving world of Formula One racing. In conventional shift mechanisms, it was necessary to reduce engine torque to close to zero during up-shift, and the time of deceleration produced by the air resistance of the vehicle had a significant effect on performance. However, further reductions in shift time could not be expected using this type of mechanism. The ultimate goal for attempts to reduce shift time is the realization of shift with no torque loss. It was assumed that achieving this within the regulations would involve a variety of issues, but it was also considered to represent an excellent opportunity to increase Honda’s competitiveness against other teams, and a development project was therefore commenced.

2. Development Aims

Shifts using one-way clutches, as are often employed in automatic transmissions in mass-production vehicles for shift from 1st to 2nd gears, enable at least one shift with no torque loss. The development aim established for the project was to realize seamless shifts by developing a shift mechanism in which this function could be applied to all the gear stages in a Formula One vehicle gearbox.

* Automobile R&D Center

3. Methods of Achieving Development Aim

3.1. Alteration of Dog Clutch Engagement Timing

In a conventional shift mechanism, when torque is acting on the dog clutches, friction on the dog clutch coupling sections prevents shift (engagement and disengagement of the dog clutches). Because of this, during shift the engine torque was reduced, and the dog clutch of the current gear stage was disengaged. When this operation was completed, the dog clutch of the next gear stage was then engaged simultaneously with the recovery of engine torque. However, the engagement of the dog clutch of the next gear stage while the current gear stage was still transmitting torque, and the disengagement of the dog clutch of the current gear stage after the next gear stage commences transmitting torque (when the current gear stage has ceased transmitting torque) would enable shift with no drop in engine torque. Figure 1 shows the difference in dog clutch engagement timings for shifts from 5th to 6th gears in the conventional and seamless shift mechanisms as an example. As the figure shows, in conventional shift there is a neutral state between the stages, while in the seamless shift, the dog clutches for both 5th and 6th gears are engaged.

3.2. Prevention of Double Engagement during Up-shift

As Fig. 1 shows, if this shift timing is applied using the conventional shift mechanism, there will be a period during which the current gear stage and the next gear stage will be engaged simultaneously, i.e., double engagement. It is widely known that double engagement may cause issues that make the vehicle inoperable. A mechanism enabling double engagement to be prevented was therefore developed for the new system. Figure 2 shows the configuration of the parts of the seamless shift mechanism. Figures 3 and 4 show the movement and function of each of the parts during up-shift.

Figure 3 shows the parts during acceleration, as seen from the front. The direction of rotation is counterclockwise. Torque from the engine is transmitted to the clutch, the lay shaft gear, the mainshaft gear, the shift ring, the gear hub, the strut, and the mainshaft, and is output to the tires. In a conventional shift, the gear hubs and the mainshaft are connected by splines, but the seamless shift mechanism employs struts that function as one-way clutches between the gear hubs and the mainshaft.

Figure 4 shows a state of double engagement, when the next gear stage has commenced transmitting torque and the dog clutch of the current gear stage is still engaged, during up-shift using the seamless shift mechanism.

When the dog clutch of the next gear engages and the gear commences transmitting torque, the rotation of the next gear stage becomes faster than that of the current gear stage, and the strut of the current gear stage is taken into the mainshaft pocket inside the gear hub where it functions as the idling side of a one-way clutch, enabling double engagement to be prevented. Because

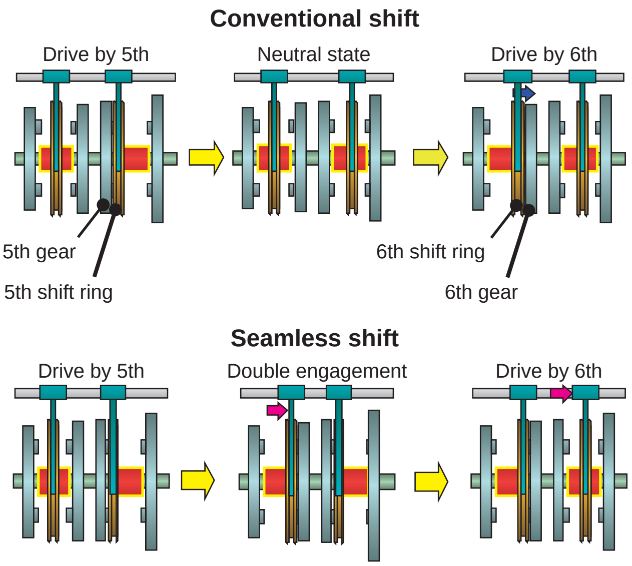


Fig. 1 Dog engagement timing

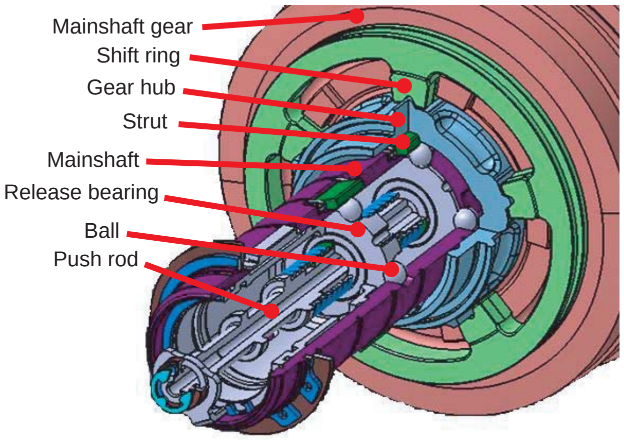


Fig. 2 Parts configuration of seamless shift mechanism

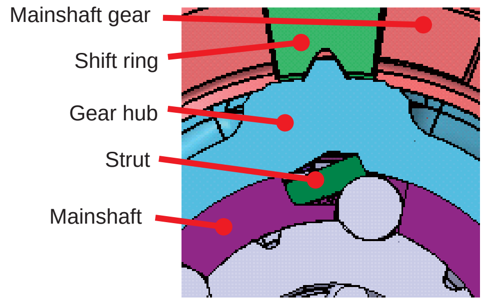


Fig. 3 Up-shift state

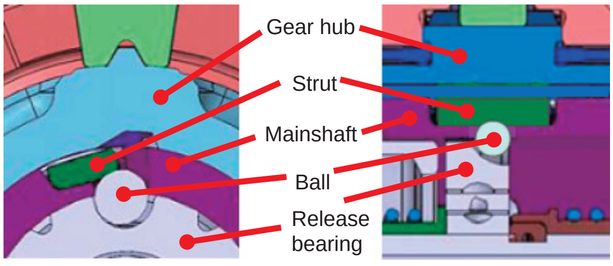


Fig. 4 Ratcheting state

the current gear stage is no longer transmitting torque, the shift ring can be released from the dog clutch without causing engine torque to drop, enabling the realization of seamless shift.

The strut of the next gear stage engages automatically with the gear hub due to centrifugal force, and functions as the engaging side of a one-way clutch.

3.3. Engine Brake

This simple one-way clutch mechanism using the struts enables torque transmission during acceleration, but it is unable to transmit torque during braking. In order to resolve this issue, a ball is inserted beneath the strut in order to lock it, enabling deceleration torque to be transmitted. Figure 5 shows the system during application of the engine brake. The path of deceleration torque is opposite to the torque path during acceleration. The input from the tires is transmitted to the driveshaft, the differential gear, the bevel gear, the mainshaft, the ball, the strut, the gear hub, the shift ring, and the mainshaft gear, and is finally input to the engine. The release bearing is locked or released by a pushrod.

3.4. Downshift

The new system features the following major differences from a conventional shift mechanism to enable the realization of seamless shift:

(1) A shift barrel profile enabling a double engagement timing to be obtained

(2) A mechanism enabling double engagement to be prevented

To enable downshift using mechanisms that focus on up-shift, when the release bearing has been released and the system has downshifted from the current gear to the next gear, the release bearing is locked again. In order to release the release bearing, it is necessary to reduce torque. This method increases the length of time that deceleration torque is reduced against a conventional shift mechanism. A shift barrel profile enabling downshift without releasing the release bearing was therefore proposed in order to achieve downshift in the same short period as a conventional shift mechanism while realizing seamless up-shift. Figure 6 shows the shift barrel profile during up-shift. The shift rings and the dog clutches engage when the shift fork stroke reaches 39% or more, and the diagonally-shaded area in the figure therefore represents a state of double engagement.

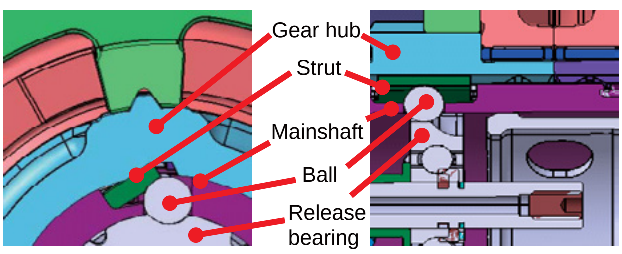


Fig. 5 Overrun state

Figure 7 shows the shift barrel profile during downshift. The diagonally-shaded area in the figure shows the system in a neutral state. This neutral state enables downshift with the release bearings locked.

This shift barrel profile also features free areas, in which the shift forks move freely. These are shown as the vertically-shaded areas in the figures. Unwanted movement of the shift forks has been prevented by positioning détente springs between the selector rails and the shift forks. The engagement and disengagement of the dog clutches on the up-shift and downshift sides at different timings has enabled the reconciliation of the desired up-shift performance with downshift in the same time period as a conventional shift.

4. Effects

The time-chart of engine speed, engine torque, and wheel speed during up-shift were compared in order to verify the effects of the seamless shift mechanism developed in this research. Figures 8 and 9 show data for conventional shift and seamless shift respectively. The seamless shift realizes up-shift while maintaining engine torque, enabling shift to be completed with no loss of drive power. In a bench test simulating the Silverstone Circuit, figures of 4 km/h when converted for speed and 7.6 m when converted for distance at the end of the home straight were obtained. The new mechanism was employed in races from the 1st race of 2005.

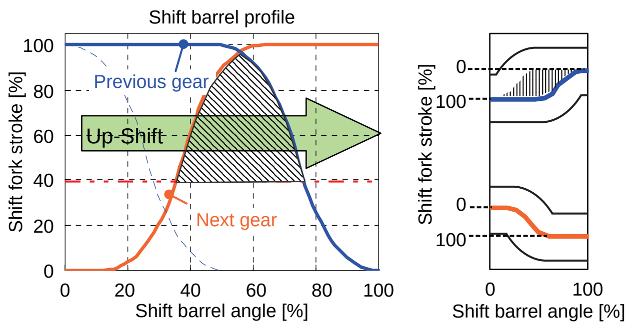


Fig. 6 Up-shift shift barrel profile

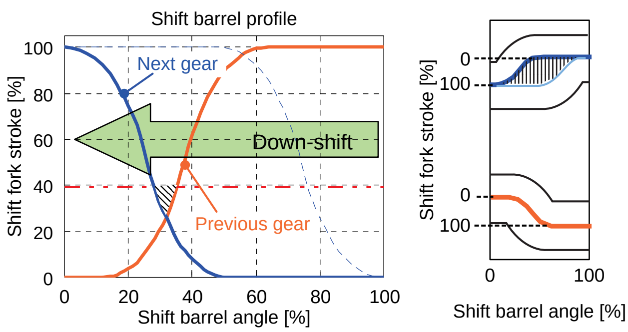


Fig. 7 Down-shift shift barrel profile

5. Evolution

5.1. Further Development Aims

Honda led the world in introducing the seamless shift mechanism, but by the end of the following season, almost all other teams were adopting systems that offered the same benefits, and Honda's advantage from the perspective of shift performance had weakened. The system had maximally increased performance in terms of preventing torque loss during acceleration, but Honda's gearbox fell behind those used by the other teams from the perspectives of weight and compactness. A development program was therefore conducted with the aims of making the system more lightweight and compact while maintaining the same level of shift performance.

5.2. Mechanism

5.2.1. Salient features

An in-shaft shift mechanism was developed that positioned one-way clutches able to control torque transmission and idling between all the shift gears and the mainshaft, and that was fitted within the mainshaft. Doing away with the shift forks, shift rings, gear hubs, and other equipment, which were conventionally positioned between the shift gears, and using only shift gears arrayed in a line enabled the total length of the gearbox to be reduced. At the same time, the weight of the gearbox itself was reduced (Figs. 10 and 11).

5.2.2. Configuration

The in-shaft shift mechanism features an operating range in which double engagement is used, and the principle of preventing torque loss is the same as that used by the seamless shift mechanism discussed above. However, the configurations of the shift mechanisms differ significantly. This section will discuss the parts forming the mechanism and the roles of those parts. Figure 12 shows the shift mechanism, including shift gears. Each strut positioned between a shift gear and the mainshaft moves in a seesaw fashion by means of small and large balls, producing the following three essential states for shift:

(1) An in-gear state, in which a strut simultaneously engages and is locked in place, enabling acceleration and deceleration torque to be transmitted (Fig. 13)

(2) A neutral state, in which the angle between the struts and the mainshaft in the circumferential direction is smaller than it is in the in-gear state and the gears and

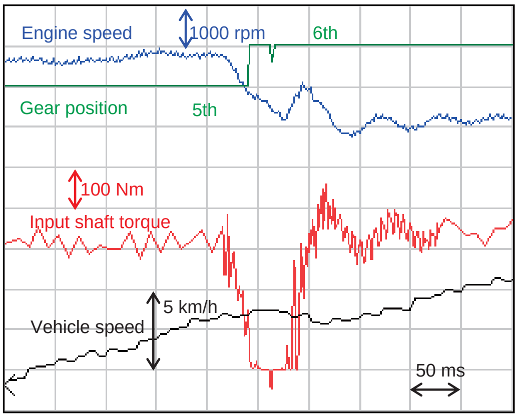


Fig. 8 Data for conventional shift

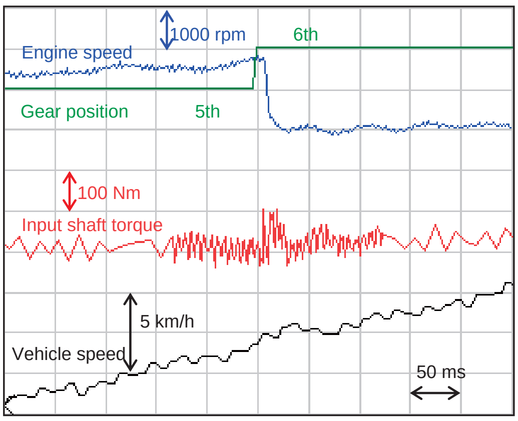


Fig. 9 Data for seamless shift

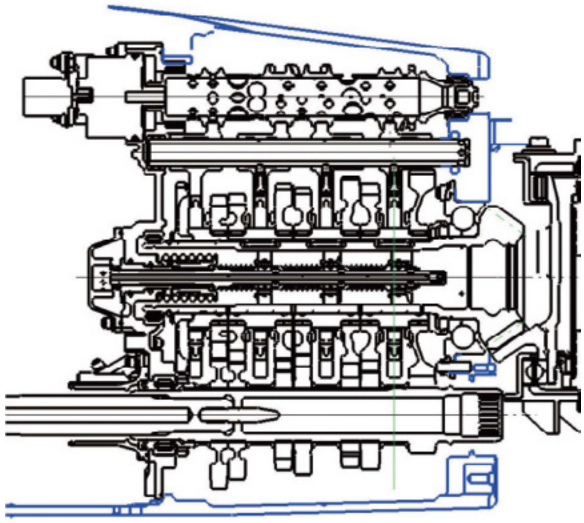


Fig. 10 Conventional seamless shift

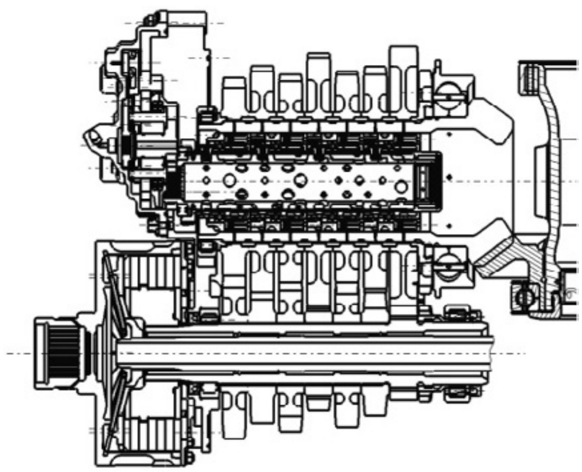


Fig. 11 In-shaft seamless shift

mainshaft idle during both acceleration and deceleration (Fig. 14)

(3) A one-way state, in which only acceleration torque is transmitted (Fig. 15)

The state of the struts is controlled by the positions of two large and small balls in the radial direction. The positions of the balls are adjusted by the movement in the axial direction of a slide cam that is provided with a cam groove in the axial direction designed specifically for use with the balls. The movement of the shift bearings in the axial direction adjusts the position of the slide cam in the axial direction by means of a spring. The position of the shift bearings in the axial direction is adjusted by the movement in the axial direction of a pin integrated with the shift bearings, which follows a barrel cam groove formed on the inside of the shift bearings in the circumferential direction. As a result, it is possible to selectively control the three states of the struts by means of barrel rotation.

5.2.3. Operation

The process of shifting from the current gear to the next gear is as follows:

(1) When the current gear is driving, the current gear is in an in-gear state, and all the other gears are in neutral states.

(2) Shift is commenced by the rotation of the barrel; the struts of the current gear and the next gear are put in one-way states simultaneously.

(3) When torque transmission shifts from the current gear to the next gear, the strut of the current gear, which has ceased transmitting torque, is put in a neutral state.

(4) The next gear is put in an in-gear state, and shift is completed.

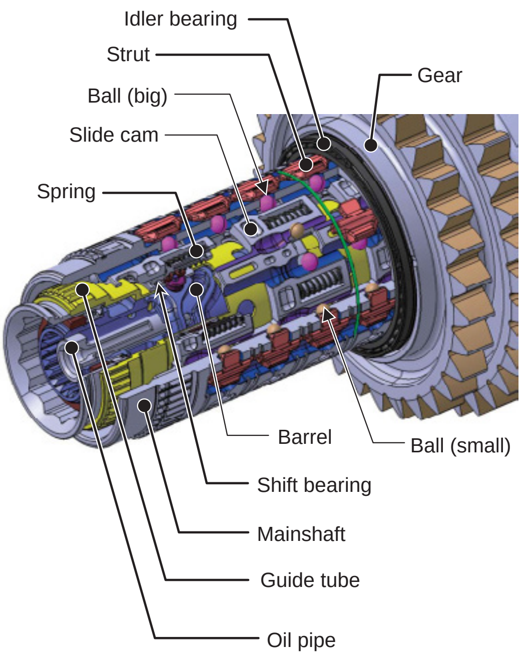


Fig. 12 Parts configuration of in-shaft seamless shift mechanism

Figure 16 shows the timing of in-gear states for up-shift and downshift. “a” shows the disengagement timing of the current gear during downshift, “b” shows the engagement timing of the next gear during up-shift and downshift, and “c” shows the disengagement timing of the current gear during up-shift. Different timings are necessary for the disengagement of the current gear during up-shift and downshift. This is necessary for the same reason as was the case for the seamless shift discussed above, but the method of achieving it differs

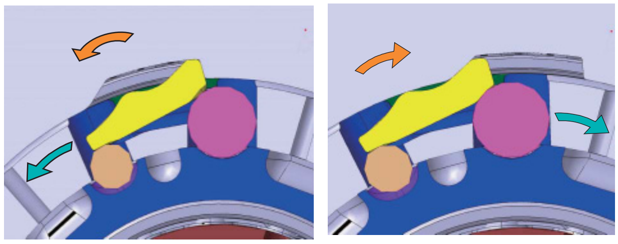


Fig. 13 In-gear state

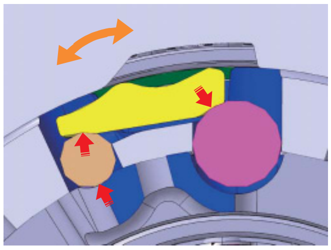


Fig. 14 Neutral state

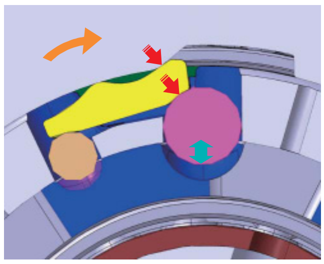


Fig. 15 One-way state

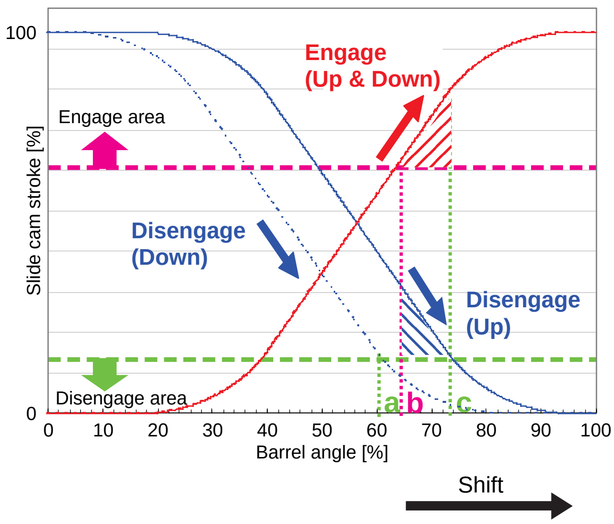


Fig. 16 Shift timings

for reasons of space. In the new system, the different timings are enabled by positioning springs between the slide cams and the shift bearings. During up-shift, the strut of the current gear cannot be disengaged until the current gear has ceased transmitting torque. By means of compressing the spring even when the barrel is rotating, the movement of the slide cam is delayed, and as a result the transition to a neutral state is delayed, thus creating a range in which double engagement occurs. During downshift, because a load is applied in the direction of disengagement of the strut, the transition to a neutral state can be made with no delay in the rotation of the barrel against the movement of the slide cam, thus enabling double engagement to be prevented.

5.2.4. Effects

The employment of the newly developed in-shaft seamless shift mechanism has reduced the total length of the mechanism by 19% (from 192.7 mm to 155.1 mm) and the weight of the mechanism by 12% (from 10.4 kg to 9.1 kg) against a conventional shift mechanism.

5.3. Verification of Performance and Reliability

Shift performance figures for the in-shaft seamless shift mechanism were identical to those for the seamless shift mechanism, shown in Fig. 9. The new system was being used under the four-race gearbox regulation introduced by the FIA in 2008, making it important to verify reliability. Bench tests were commenced on the system, but Honda then announced its withdrawal from Formula One racing, and the development project was discontinued when the tests had reached the 1300 km mark of a projected 2500 km.

6. Conclusion

A shift mechanism that has a modified shift sequence to prevent torque loss during up-shift and selectively uses one-way clutch mechanisms to prevent double engagement was realized within the scope of Formula One regulations. This development produced the following outcomes:

(1) As the first shift mechanism of its kind in the Formula One world, the new system enabled lap times to be reduced by 0.4 sec per lap, and was used in races from 2005.

(2) The realization of up-shift with no torque loss and minimal torque fluctuation during shifting enabled shifting in situations of low tire grip, such as when cornering or during rainy conditions.

The quest for increased compactness while maintaining seamless shift performance by removing the shift forks and dog rings and fitting the mechanism inside the shaft enabled the total length to be reduced by 19% and weight to be reduced by 12%.

■ Author ■


Takashi YOSHIOKA


Takeshi UCHIYAMA


Katsumi KUBO


Ryo MATSUI

Development of Honda Gears for Formula One Gearbox

Koichi KONISHI* Hiroaki NOJIMA*
Teruoki NAGANUMA* Hiroshi KIMURA*

ABSTRACT

Development programs were conducted to enable the development of Honda-made gears that balanced low weight and compactness with reliability. Development of a technology for the formulation of gear specifications using FEM analysis and a method of predicting the lifespan of the gears based on S-N curves, which formed the basis for FEM analysis, helped enable continuous short-term development efforts. As gears would sometimes fail in actual vehicle tests, a method of evaluation of lightweight gears optimized for high-load, short-lifespan race use was established with the plastic deformation of the gears as an index. The lightweight, compact, and high-reliability gears developed in this process were employed in races from 2003, and were entirely trouble-free.

In addition, a diamond-like carbon (DLC) coating optimized for race use was applied to all the gear tooth surfaces to boost the performance of the powertrain, helping to achieve a transmission efficiency of 97%. The coating was applied to different gear sets in sequence from 2007.

1. Introduction

At the beginning of Honda's third Formula One era, gears manufactured by a specialist race-gearbox maker were used. However, the top teams were using lighter and more compact in-house gears to increase their competitiveness.

Formula One gearboxes are positioned close to the rear-ends of the vehicles, and their weight makes a significant contribution to dynamic performance. A wide variety of ratio gears is necessary to help enable drive force to be set for each race circuit. In addition, gearbox issues will directly result in retirement from the race. These factors necessitated methods of determining specifications that would reduce the weight of the gears as well as helping to enable accurate prediction of gear life. Regulations freezing engine development also increased the importance of reducing loss in the drivetrain.

In response to these demands, Formula One gear design methods using FEM analysis and standards for quantitative evaluation were formulated, and methods of dealing with exceptional overloads were developed, helping to enable the development of gears that reconciled low weight with reliability. In addition, the transmission efficiency of the gears was identified as the major factor in drivetrain loss, and a DLC coating adapted to race conditions was developed for the gears.

* Automobile R&D Center

2. Layout of Formula One Gearbox Internals

Given considerations of vehicle dimensions, aerodynamic packaging, and the stipulations of regulations, a longitudinal two-parallel-axis configuration in which the intermediate shaft (cross-shaft) was positioned one stage upstream from the final reduction gears, was employed for the gearbox internals. Figure 1 shows this gearbox configuration.

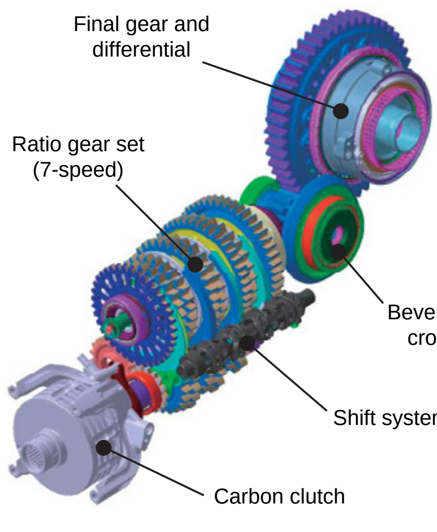


Fig. 1 3D model of Formula One gearbox internals

3. Formula One Gearbox Design Method using FEM Analysis

3.1. Issues of Applying Mass-Production Gear Design Methods to Formula One

The conventional gear design process involves the use of in-house gear specification simulation tools to study gear specifications and evaluate the stresses generated on gear-tooth bottoms. For mass-production gears, it is normal for a specific thickness of the gear-tooth bottoms to be determined by the gear module, and this is used as a precondition for the above-mentioned simulation tools. Formula One gears must balance resistance to extremely high loads and weight saving, even in short lives, in comparison to the gears used in mass-production vehicles, and the concepts used in determining specifications are consequently different. Tooth root strength and pitting strength are necessary in Formula One gears, and the gear modules are therefore increased in size. The thickness of the tooth roots is reduced in order to save weight. The resulting decline in stiffness must be taken into consideration, so conventional design methods were not sufficient to the task. FEM analyses that considered contact between gear teeth were therefore conducted.

Figure 2 shows the difference in shape between a mass-production gear and a Formula One gear.

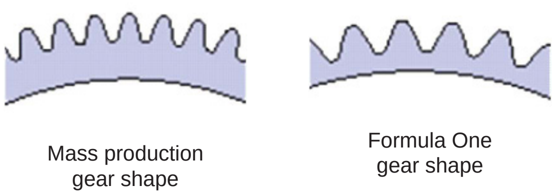


Fig. 2 Gear shape comparison

3.2. FEM Analysis Conditions

Analysis conditions are important in FEM analyses, and in the FEM analyses of spur gears in particular, it is necessary to reproduce the worst-case conditions for a variety of meshing states. The contact gear ratio for the spur gears alternates between 1 and 2, and the gear meshing condition is therefore a continuous sequence of one tooth meshing – two teeth meshing – one tooth meshing. FEM analysis showed that of these, one tooth meshing, and in particular the meshing condition in which the meshing point was highest in the direction of the depth of the gear tooth, placed the greatest stress on the gear tooth root.

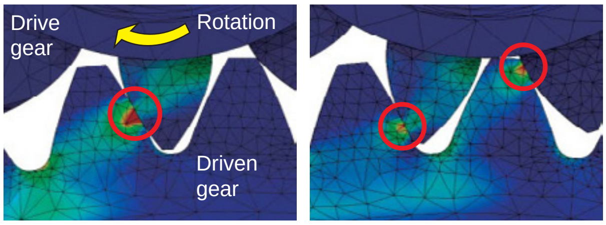
Figure 3 shows differences in gear meshing conditions and stress distribution, and Fig. 4 shows changes in the stress on the tooth roots when the meshing phase was altered.

In one tooth meshing, stress concentrates at a single meshing point (Fig. 3), while in two teeth meshing there are two contact points, and the stress concentration is therefore relieved (Fig. 4).

The use of the gear specification simulation to determine from among a variety of tooth meshing conditions the condition in which the stress on the tooth roots was greatest, and the reflection of this result in the FEM analysis, helped to reduce in length the period necessary for analysis.

3.3. Optimization of the Degree of Tip Relief

Tip relief refers to optimizing the form of the gear-tooth tip in relation to an involute curve with consideration of elastic deformation when a load is placed on the tooth. In Formula One gears, the reduction in stiffness mentioned above results in a high level of elastic deformation, which changes tooth contact and has a significant effect on tooth strength and contact face pressure. The level of elastic deformation of the gear teeth tips was therefore predicted using FEM analysis. The degree of tip relief was optimized based on the results.



(a) 1 tooth meshing (b) 2 teeth meshing

Fig. 3 Gear meshing condition

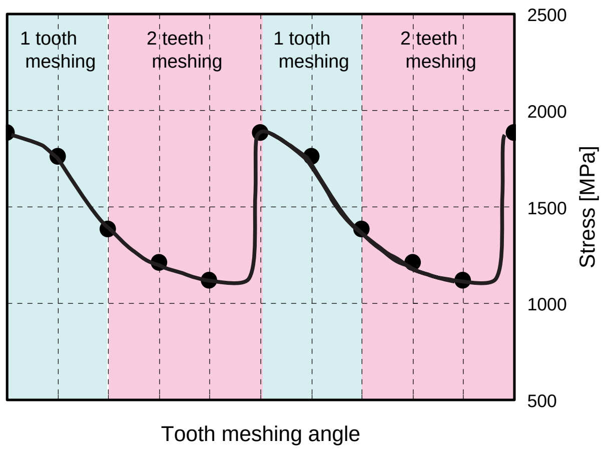


Fig. 4 Change in stress with gear meshing condition

4. Quantitative Evaluation Method for Formula One Gears

4.1. Gear Fatigue Strength Tests

Up to this point, it had been normal to evaluate the durability of race gears using vehicle tests alone. For the development of in-house Formula One gears, however, rig tests of gear units using S-N curves (Wohler curves) and torque frequency measured in actual vehicles were employed in gear evaluation. The reliability of high-load, low-cycle fatigue S-N characteristics represented a concern, but cross-referencing with the results of vehicle test evaluations conducted previously indicated that unit rig tests functioned adequately for evaluations. The determination of gear specifications on this basis helped to enable weight savings to be reconciled with high reliability (Fig. 5). The value of stress at the tooth roots was calculated from input torque, and the S-N curve was formulated based on test results.

4.2. Early Breakage of Gears

The technology for prediction of gear lifespan using rig tests reduced gear breakages, but breakages sporadically occurred earlier than the end of the predicted lifespan of the gears during vehicle tests. Figure 6 shows a representative example of breakage.

When gears that had been used in races were measured, plastic deformation was observed in from one to several teeth (Fig. 7). Given that the plastic deformation only affected part of the gears, it was conjectured that it resulted from instantaneous overloads, and that this resulted in the early breakage of the gears. The phenomenon was therefore analyzed.

Because the amount of plastic deformation was minimal, criteria for judgment of tooth profile data were set, and a difference in the degree of pressure angle correction of 0.2 μm or more was defined as plastic deformation.

Plastically deformed gears were put through fatigue strength tests. The results showed in all cases that the life of the gears was reduced, with the lifespan until breakage only 11.6% of the projected figure in one case (Fig. 8).

4.3. Measures against Overload

In Formula One gearboxes, control is applied to produce an optimum clutch clamp load in order to protect the drivetrain components from instantaneous overloads when the vehicle takes off or shifting is engaged.

Despite this, instantaneous overloads due to inertia and other forces in the gearbox resulting from variations in the clutch friction materials, tire traction on uncertain road surfaces, and irregular steering operation cannot be entirely prevented. However, if gear design incorporated allowances for occasional overloads, increases in weight would be unavoidable, and competitiveness would decline.

In the Formula One gearbox, torque input during vehicle operation was monitored by a torque sensor directly connected to the engine input shaft. The input torque on plastically deformed gears was studied from torque sensor log data. The results, when converted to figures for stress at the gear teeth roots in a simulation, showed that plastic deformation of the gears occurred when the torque input exceeded a threshold of 2300 MPa.

Gear durability tests also showed that plastic deformation occurred at levels of tooth root stress above

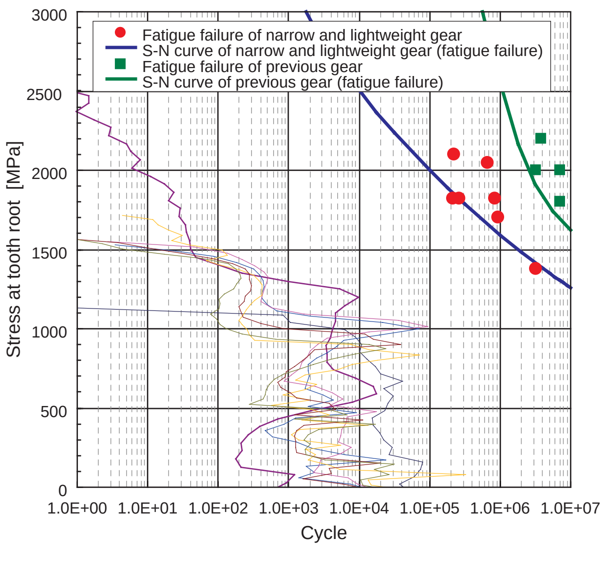


Fig. 5 Fatigue failure test results

Fig. 6 Broken gear

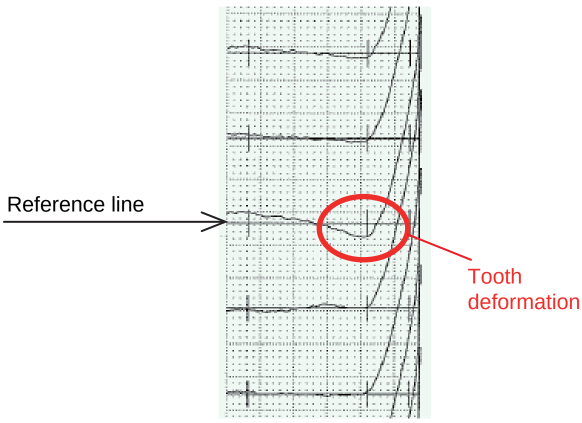


Fig. 7 Tooth profile data of deformed gear

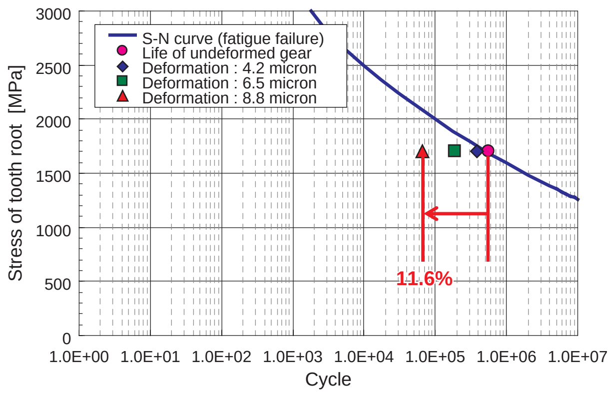


Fig. 8 Fatigue failure test result of deformed gear

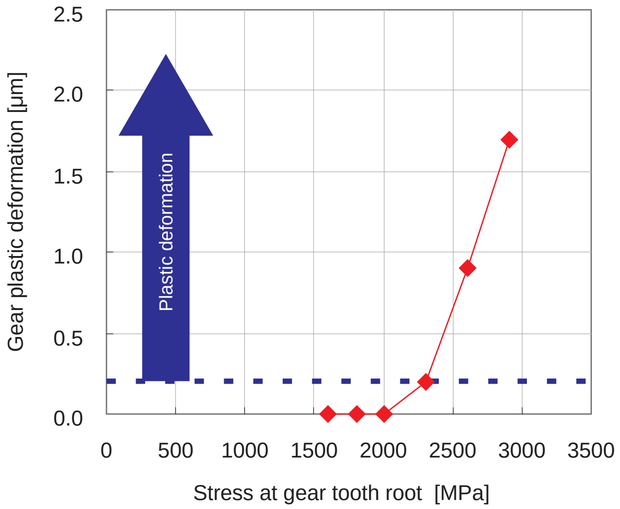


Fig. 9 Gear strength test results

2300 MPa (Fig. 9).

Based on these results, a gear torque limit corresponding to a level of gear-tooth root stress of 2300 MPa was set.

In addition, telemetry data from the torque sensor was monitored in real time, and in the event that a torque input exceeded the torque limit, the relevant gear was replaced during gearbox maintenance.

The measures discussed above helped to reconcile the achievement of low weight with high durability, and the gears were trouble-free during races.

5. Achievement of Increased Efficiency in Formula One Gearboxes

5.1. Analysis of Status of Transmission Loss

A situation analysis of factors in transmission loss was conducted. These factors were isolated by measuring friction in an unloaded state and efficiency in a loaded state (i.e., during transmission). The following factors were identified:

(1) Loss due to oil churning
(2) Loss due to dragging of the oil seals and other parts
(3) Loss due to torque transmission
(4) Oil pump drive loss

Of these, loss due to torque transmission was the reason that loss increased or decreased with the magnitude of torque input. Bearing loss as a factor in transmission loss in a loaded state was calculated using a theoretical formula to separate it from gear transmission loss. Figure 10 shows the rate of contribution of each power loss factor.

These results showed that gear transmission loss was the major factor in gearbox loss, representing 62% of the total figure.

5.2. Methods of Increasing Efficiency

Transmission loss resulted from slipping of gear surfaces against each other during meshing, and could be reduced by reducing the coefficient of friction of the gear surfaces.

The following three methods were available to reduce the coefficient of friction of the gear surfaces:

(1) Reducing the roughness of the gear surfaces
(2) Applying a coating to the gear surfaces
(3) Using a low-friction gear oil

With regard to (1), the gears used in Formula One engines receive a polishing finish and barrel polishing, and no further roughness reduction could be expected. Transmission loss was therefore reduced through (2), the application of a coating to the gear surfaces, and (3), the development of a low-friction oil.

5.3. Selection of Coating Film Type

Under actual Formula One use conditions, the velocity of gear surface slip reaches 20 m/s or more, and gear surface pressure reaches 2 GPa or more. It was necessary to employ a coating which would resist wearing even under these conditions. The gear coating film was selected from among the following candidates:

(1) Metal DLC, which displays good wear resistance, and is used on race engine components
(2) An increased-hardness metal DLC

The following two procedures, which are used to help prevent pitting of the gears in mass-production transmissions and could be expected to reduce the coefficient of friction by increasing slidability, were also included in the comparative study:

(3) Solid-film lubrication (Molybdenum disulfide resin coating)
(4) Sulfurizing

These four coating types were studied for effectiveness in reducing loss and levels of durability. Gears to which only a carburizing treatment had been applied, which were previously standard, were used as a benchmark for comparison.

5.4. Verification of Effects of Coatings and Durability

Figure 11 shows the results of verification of loss reduction in unit rig tests, and Fig. 12 shows the surface condition of gears following durability tests.

Loss was reduced by 2 kW in the gears with the DLC coatings. Some damage occurred to the metal DLC coating, but only minute scuffing of the tips of the gear teeth was observed in the gears with the hardened DLC

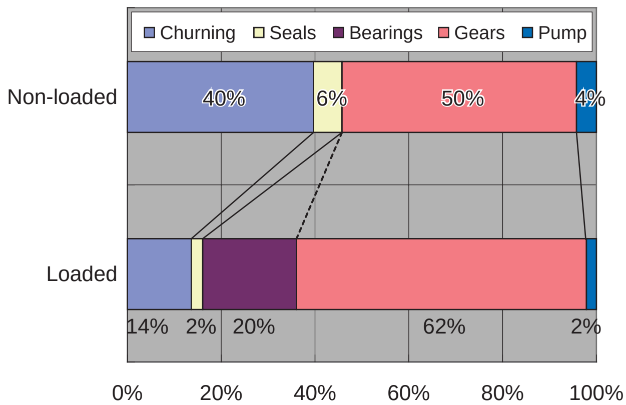


Fig. 10 Effective rate of power loss factors

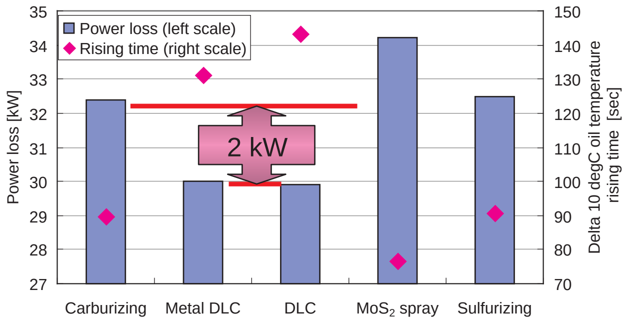


Fig. 11　Confirmation of loss reduction effect

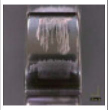
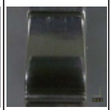
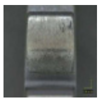
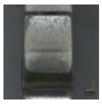
Coating	Carburizing	Metal DLC	DLC	MoS_2 spray	Sulfurizing
Gear surface condition	 Teeth touch mark only	 DLC peeling DLC wear	 Minute scuffing	 Coating peeling Gear surface wear & scoring	 Gear surface wear & scoring

Fig. 12　Surface condition after durability test

coating, and this did not affect the actual performance of the gears.

Solid-film lubrication and sulfurizing demonstrated no transmission loss reduction effect, and wearing of the coatings was also observed. This is because it is necessary for the surfaces of the gears to be made rougher than the bases in order to apply these coating treatments, and the consequent increase in roughness results in a decline in transmission efficiency.

5.5. Use of DLC Coating in Races

DLC-coated shift gears and final gears were employed in races from the middle of the 2007 race season, and DLC-coated bevel gears were employed from the 2008 season. Optimization of tip relief and the coating pre-treatment helped to enable the achievement of a sufficient level of durability to comply with the regulation that gears must be used for a full four races (Fig. 13).

Fig. 13　Final gear condition after 4 races

5.6. Low-friction Gear Oil

A low-friction oil was developed in cooperation with an oil manufacturer. At the initial stage of the development, reductions of loss of 1 kW or more were obtained in gears without DLC coatings, but when DLC coatings were applied, no loss reduction effects was obtained.

The compatibility between the DLC coating and the oil was therefore examined, helping to enable the development of an oil that reduced transmission loss by 0.4 kW. This oil was used in races from 2008.

6. Conclusion

The following results were obtained from the Formula One gear development program:

(1) A Formula One gear design method using FEM was developed.

(2) Methods of predicting the lifespans of gears in rig tests and of quantitatively evaluating exceptional overloads were developed. These methods formed the basis of a method implemented in gearbox management during races.

(3) A gear coating and an oil were developed that helped to achieve a gearbox transmission efficiency of 97%.

The development results listed above helped to achieve a good balance between function and reliability, and the developed gears demonstrated increased transmission efficiency and produced no gear-related issues during races.

Acknowledgments

The authors wish to take this opportunity to express their gratitude to the staff members of Nippon Oil Corporation, who generously assisted in the development of a low-friction oil.

■ Author ■

Koichi KONISHI

Hiroaki NOJIMA

Teruoki NAGANUMA

Hiroshi KIMURA

Development of Lightweight and Compact Differential for Formula One Car

Toshio HIYOSHI* Yuichi SUENAGA*
Haruki YOKOYAMA* Yoshikazu KATSUMASA*

ABSTRACT

The differentials used in Formula One vehicles comprise a differential mechanism and a differential restriction device (LSD), which together form a bias-adjusting mechanism, and a final reduction gear set. Because the differential is positioned close to the rear end of the vehicle, it contributes to dynamic performance. Reducing the weight of the differential helps to centralize the mass of the vehicle, and increasing its compactness boosts aerodynamic performance. Development efforts were concentrated on the achievement of a lightweight and compact differential, with the focus in the initial stage on the centralization of mass. Previously mounted on the final driven shaft, the bias-adjusting mechanism was positioned on the final drive shaft, helping to reduce the center of gravity and the yaw moment of inertia. A full pinion engagement planetary gear was employed as the differential gear. A study of the merits and drawbacks of the mechanism resulted in the development of an ultra-short differential (USD) employing a full pinion engagement double pinion planetary gear positioned on the final driven shaft as the differential gear. It was predicted that this would increase the compactness of the unit and reduce its weight by 1.2 kg.

1. Introduction

Normally, the bias-adjusting mechanisms of Formula One differentials are positioned on the final driven shaft, and transmit the drive force that has been reduced by the transmission to the left and right drive wheels. This necessitates a large number of transmission parts and a high level of torque transmission, and the differential therefore represents a significant percentage of the weight of the drivetrain. In addition, due to restrictions on the operating angle of the drive shaft, the final driven shaft is positioned close to the rear end of the vehicle, at approximately the same height as the centers of the wheels. This means that a heavy component is located in a high position, far from the vehicle's center of gravity. Reducing the weight and size of the differential would contribute to lowering the vehicle's center of gravity and reducing yaw moment of inertia; reductions in these parameters help to boost cornering performance. In addition, the bias-adjusting mechanism also incorporates a limited-slip differential (LSD) that controls the right-left differential restriction torque, and the performance of this unit is also an important factor.

This paper will discuss the development of a lightweight and low center of gravity differential reconciling the satisfaction of these performance demands with durability.

* Automobile R&D Center

2. Development Aims

The following aim was established in the initial development project in order to realize a low center of gravity and a low yaw moment of inertia.

(1) Positioning weight as low and as close to the center of gravity as possible

The development project aimed to achieve a low center of gravity and a low yaw moment by shifting the bias-adjusting mechanism from the final driven shaft to the final drive shaft, closer to the vehicle's center of gravity, and to a lower position.

Furthermore, the following aim was established for the development of the ultra-short differential, based on knowledge gained from the initial development and consideration of advantages and disadvantages:

(2) Reducing the weight of parts located far from the center of gravity and in a high position

This development aimed to contribute to the realization of a low center of gravity and a low yaw moment by reducing the size of the differential in order to reduce its weight.

Regulations stipulate the total weight of Formula One vehicles, meaning that a weight reduction would be compensated for by using ballasts. By this means, it would be possible to realize an even lower center of gravity and yaw moment. Figure 1 shows an image of the aims of the developments discussed here.

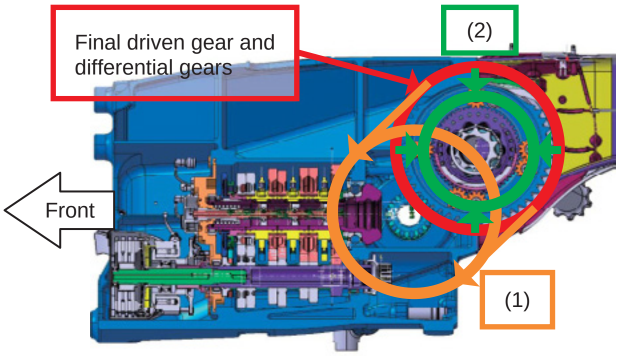


Fig. 1 Method of achievement of low center gravity and low yaw moment

3. Methods of Achievement of Development Aims

3.1. Efforts to Centralize Mass

The first stage of the development project focused on centralizing mass that would be effective in lowering the center of gravity and reducing yaw moment of the vehicle. A mechanism was developed that shifted the differential, the heavy component of the gearbox, from the final driven shaft to the final drive shaft, closer to the vehicle's center of gravity.

The transfer of the differential from the final driven shaft to the final drive shaft necessitated the use of two pairs of final gears. With respect to the width of the gears, because it was sufficient that one of the gears was able to transmit torque to the left or right wheel, the tooth width was reduced by approximately half against a conventional configuration.

The transfer of the bias-adjusting mechanism from the driven shaft side to the drive shaft side, which has a lower reduction ratio, helped to enable torque capacity to be reduced and the unit to consequently be reduced in size. Because the differential gears would be incorporated in the final drive bevel gears, a compact configuration was essential. A full pinion engagement planetary gear configuration that reduced the shared load through meshing of all contiguous pinion gears was adopted.

The cases of Formula One gearboxes must be made narrow, to help prevent interference with the flow of air to the rear of the vehicle. To help enable this, a configuration was adopted in which the LSD pistons were positioned opposite the clutch, and rods were introduced to connect them.

Figure 2 shows this bias-adjusting mechanism.

This configuration ultimately transferred approximately 2.5 kg of mass to the final drive shaft, reducing the yaw moment of inertia by 0.19% and lowering the center of gravity by 0.4 mm.

Figure 3 shows an image of this transfer of mass.

Rig tests showed that this mechanism achieved the necessary level of durability. However, the configuration was not employed in a vehicle, because the transfer of the differential to the final drive shaft would necessitate significant structural changes to the gearbox case, and the consequent increase in weight would nullify the merit of lowering the center of gravity.

This development project demonstrated that the bias-adjusting mechanism could be reduced in size through the use of a full pinion engagement planetary gear mechanism.

3.2. Efforts to Reduce Weight (USD)

It was proposed that the bias-adjusting mechanism should be positioned in the interior of the final driven shaft between the right and left drive shaft joints in order to achieve weight savings by reducing the size of the final gears and the differential.

Figure 4 shows the layout of a standard planetary bias-adjusting mechanism. This is a single-pinion planetary configuration that is composed of four pairs of mutually meshing single pinion gears, as differential gears, and left and right drive shaft drive gears positioned in parallel that mesh with the single pinion gears. The configuration is positioned on the final shaft outer circumference side and the inner side of the final

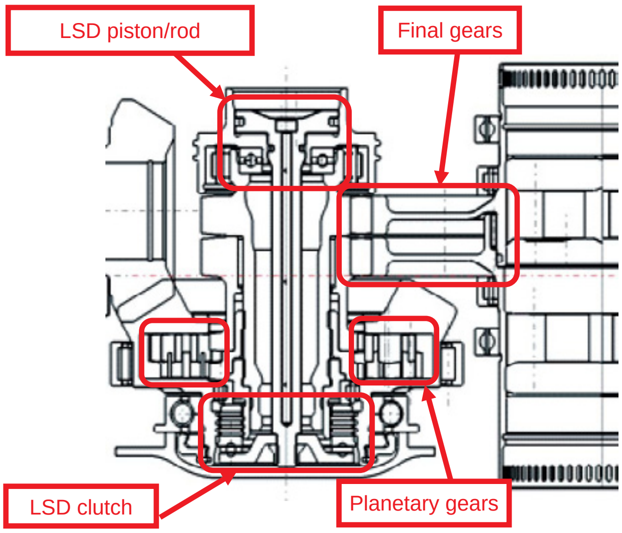


Fig. 2 Bias-adjusting mechanism on final drive shaft

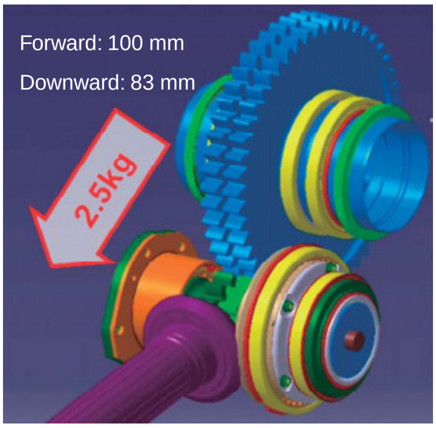


Fig. 3 Image of movement of weight

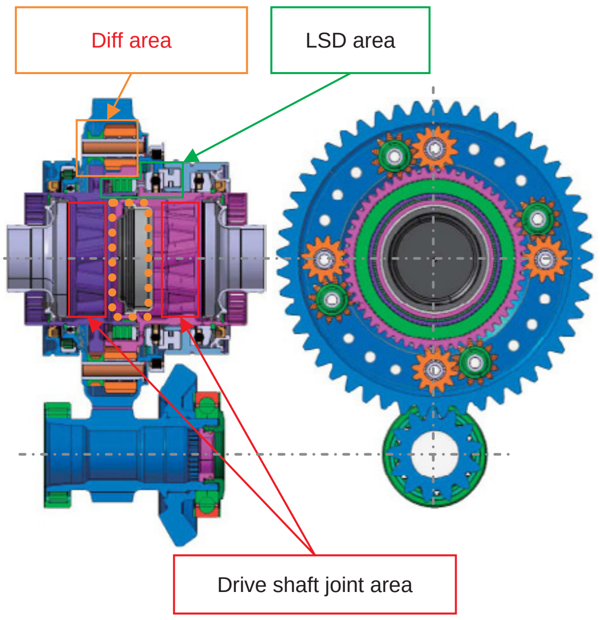


Fig. 4 Standard differential

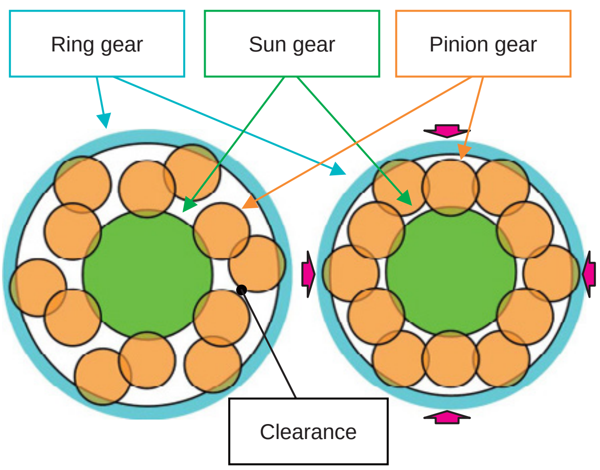


Fig. 5 Standard double pinion planetary gear

Fig. 6 Full pinion engagement double pinion planetary gear

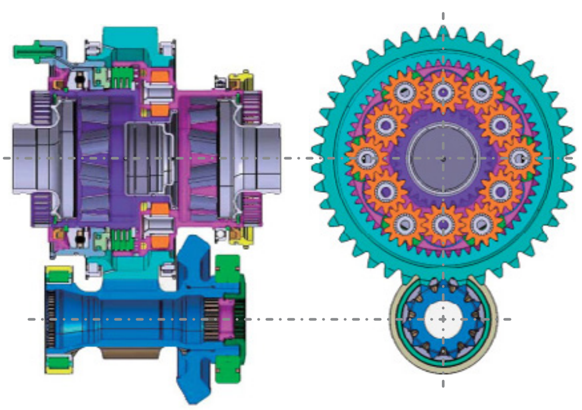


Fig. 7 USD

driven gears (the differential area) to drive the pinion gears. The unit is also provided with an LSD that produces left-right differential restriction torque using hydraulic control (LSD area).

From considerations of strength, the existing external diameters of the drive shaft joints could not be reduced. In addition, an increase in the total width of the differential was unavoidable if the existing mechanism was to be made more compact (the diameter of the final gears reduced), as it is fitted inside the final driven gear. The configuration of the LSD had also become complex, and no weight saving benefit could be obtained.

In order to resolve these issues, a double pinion planetary bias-adjusting mechanism was adopted to help enable the mechanism to be positioned between the drive shaft joints (in the drive shaft joint area) and reduced in size.

In the conventional single pinion planetary configuration, three gear meshing positions – left sun gear-pinion gear, pinion gear-pinion gear, and pinion gear-right sun gear, were arrayed in parallel. By contrast, the gear meshing positions in the double pinion planetary gear – ring gear-pinion gear, pinion gear-pinion gear, and pinion gear-sun gear - were arrayed vertically, and in-line engagement was obtained in parallel. This helped to enable the width of the bias-adjusting mechanism to be reduced. In addition, in order to further reduce the width and diameter of the double pinion planetary gears, the number of pinion gears was increased to six pairs from the former four pairs, reducing the shared load. However, increasing the number of pinion gears made it necessary to respond to interference between contiguous gears. If the diameters of the pinion gears were reduced in order to help prevent interference, it would be necessary to increase the width of the teeth to maintain strength. Again, if the angles of the interior and exterior pinion gears were increased to help prevent interference, the diameter of the ring gears would increase (Fig. 5). A full pinion engagement pinion gear configuration was therefore adopted to help enable the size of the ring gears and the final gears positioned outside the ring gears to be reduced. Figure 6 shows this configuration.

As a result, the distance between the final gear shafts was reduced from 125 mm to 100 mm, and reductions in size and weight were realized while the width of the total assembly remained unchanged (Fig. 7).

4. Verification of Performance and Durability

4.1. Differential Friction

The purpose of a hydraulically controlled differential mechanism is to restrict the differential motion between the left and right wheels in response to the operating status. If differential restriction torque exists during steady cornering with no drive force acting, then understeer is generated and this impedes drivability. Because friction exists in the differential limiting mechanism itself, even if oil pressure is entirely removed, differential restriction occurs corresponding to the amount of this friction. Given this, the development aimed to achieve the same differential friction as that of the proven 2008 specifications, and achieved almost identical performance.

4.2. LSD Control Performance

The differential restriction torque demanded from the differential during a race changes moment by moment in response to the operating status of the vehicle. A sufficiently short response time from the torque command to the generation of differential restriction torque is essential to controlling vehicle behavior as desired.

Figure 8 compares the level of displacement of the clutch assembly and torque response time for two differential specifications with different levels of clutch assembly stiffness. When engaged, the time from commencement of hydraulic pressure rise to 80% of maximum demand torque was evaluated, and when disengaged, the time from the commencement of hydraulic pressure drop to 5% of the same torque figure was evaluated. The vertical axis of the graph shows the level of axial displacement of the clutch assembly under a specific clamp load. The higher this figure, the lower the stiffness of the clutch assembly. Response time when the clutch was engaged satisfied the target, bettering the response time of the 2008 specifications. Neither specification satisfied targets when the clutch was disengaged, but a correlation with stiffness was verified. The clarification of stiffness targets guided development.

4.3. Durability

Verification of gearbox durability was essential, given the regulation stipulates that gearboxes must be used for four consecutive races. Rig tests were therefore conducted using loads corresponding to four races on the Monaco circuit, on which the frequency of use of the differential is highest. In the initial durability test, fatigue breakage occurred in the ring gears at approximately 60% of the target load. The following measures were effected before the next durability test:
(1) The tooth root shape was modified.
(2) Shot peening treatment was applied.

The modification of the shape of the tooth roots resulted in gear teeth being connected with a single radius at the bottom. An image is shown in Fig. 9.

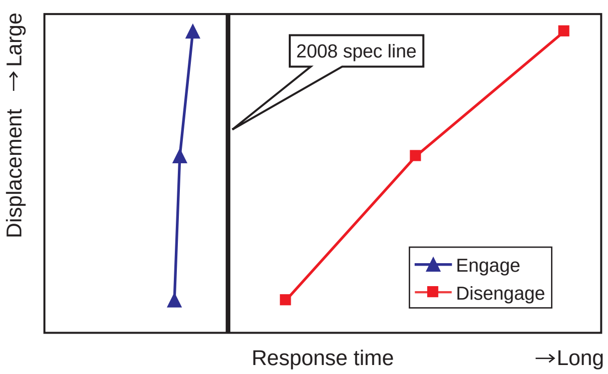


Fig. 8 Torque response time

The destruction rate of damaged final gears is 100%. Figure 10 shows an S-N curve for the ring gears in this development estimated based on this fact from the rig test results for the final gears up to this point.

Structurally speaking, minimal benefit would be obtained from reducing the width of the teeth of the final gears in the USD against the previous final gears, and some margin was thought to exist in terms of tooth root bending stress. Easy-to-manufacture shapes were therefore selected for the tooth roots, but target durability could not be achieved due to a decline in strength.

The effect of the single radius produced by the modification of the tooth root shape was verified in the same rig tests that had been conducted previously on the final gears. As in the case of the single radius, the research experience confirmed that the application of shot peening treatment to the tooth root increased the life of the gears by approximately 15%. Applying these results to the USD, the destruction rate corresponding to four races, the period for which regulations stipulate that the gears must be used, was 2.5%. The use of a single radius for the tooth roots and the shot peening treatment applied to the tooth roots would therefore increase durability. These results are shown in Fig. 11.

The effectiveness of these measures was, however, not verified due to the termination of the development program.

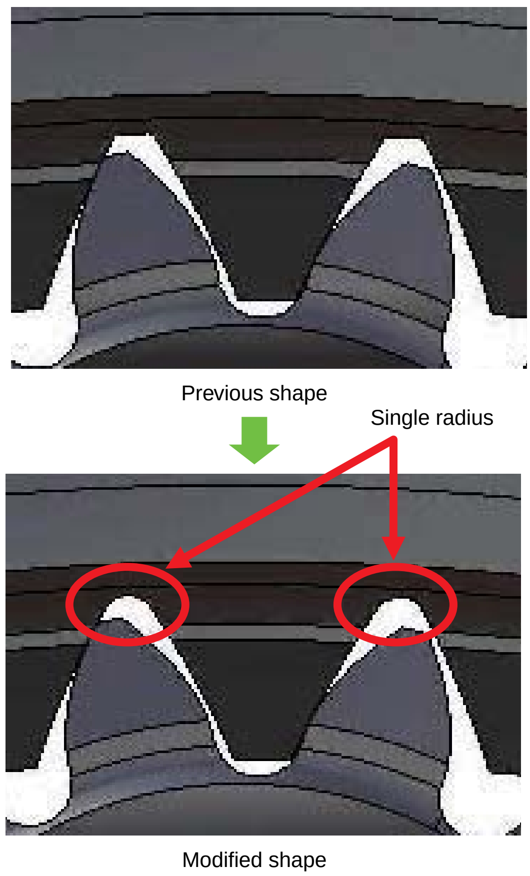


Fig. 9 Modification of root shape

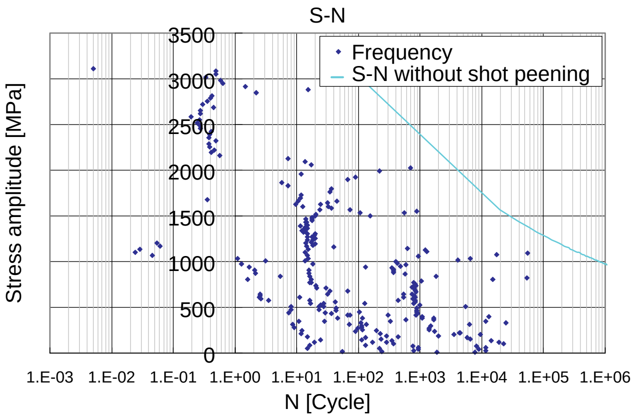


Fig. 10 Until fatigue destruction

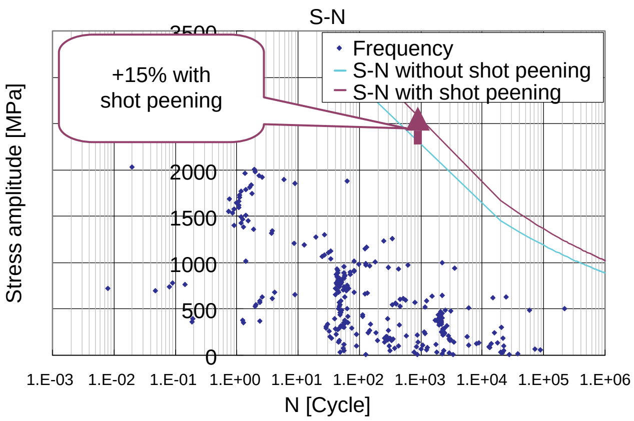


Fig. 11 4 races with large root R and shot peening

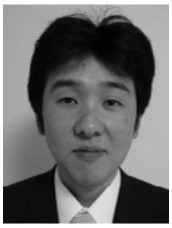
5. Conclusion

A full pinion engagement double planetary gear was employed in a differential mechanism in order to reduce the weight of the mechanism, lower the center of gravity, and reduce yaw moment of the vehicle. The following results were obtained:

(1) Compared to the 2008 specifications, the weight of the unit was reduced by 1.2 kg (-14% in terms of weight ratio), and the vehicle's inertial moment was reduced by 0.30% and its center of gravity lowered by 1.1 mm.
(2) The distance between the final drive shafts was reduced from 125 mm to 100 mm.
(3) Tests showed that the performance of the LSD could match the performance of the 2008 specifications.
(4) Tests showed that a sufficient level of reliability could be achieved to satisfy the regulation that gearboxes were not to be replaced for a four-race period.

■ Author ■


Toshio HIYOSHI


Yuichi SUENAGA


Haruki YOKOYAMA


Yoshikazu KATSUMASA

Development of Clutch System for Formula One Vehicle

Koichiro INUKAI* Yasuo NAKAGAWA* Hiroyuki KUSHIYAMA*
Ryuhei KATAOKA* Yoshiki NAGATOSHI* Keiji SHIOTA*

ABSTRACT

A direct push clutch (DPC) was developed to enhance the controllability of clutch torque in a Formula One clutch system. A direct push mechanism in which clamp force was generated by a plate with the function as lever and a hydraulic actuator was employed in place of the previous mechanism, in which the clamp force was generated by a diaphragm spring. The transition from stroke control to direct control of the clamp force did away with elements that formerly produced changes in system characteristics, such as variations in the diaphragm spring characteristic and spring hysteresis, and thermal expansion throughout the system. In combination with other enhanced control technologies, this enabled the development targets for the system to be realized. This helped to enable half-clutch starts (slip starts) to be employed from 2006, even after regulations prohibited the use of feedback control. As a result, a maximum reduction in time from 0-100 km/h of 0.46 sec (corresponding to a distance of approximately 11 m) was obtained, giving the vehicle a competitive advantage. The developed system was used in races from the first race of the 2006 season.

1. Introduction

One of the important functions demanded from the clutch systems used in Formula One vehicles is the control of torque at the start of the race in order to maximize tire performance. In 2003, feedback control using the tire slip rate as a parameter was commonly used to control clutch torque, enabling optimal clutch torque to be realized. However, changes to the Formula One regulations in 2004 prohibited the use of feedback control, necessitating the development of a clutch system in which clutch torque could be predicted with a high degree of accuracy.

The friction coefficient, μ, of the clutch friction material and the clamp force acting on the clutch friction material (clamp load) are factors that affect the accuracy of prediction of clutch torque. Of these, the clamp force can be controlled in a clutch system.

The conventional Formula One clutch system was termed “pull clutch” due to its mode of operation, and it originated in the clutches used in mass-production manual transmissions. A diaphragm spring was used to apply a clamp load to the dry carbon clutch friction material. A hydraulic actuator controlled the stroke in the direction that would detach the diaphragm spring, and clutch torque was controlled with the stroke as a parameter. Factors including variations in the dimensions of the component parts, variations in the diaphragm spring characteristics and spring hysteresis, and thermal expansion of the parts would cause the clamp load to change, making it challenging to accurately control clutch torque. As a result, slip start, a start method necessitating accurate clutch torque prediction, could not be used. A race start method known as dump start was employed to enable these issues to be avoided. This start method is used in conjunction with engine over-speed control, in which the clutch is instantaneously engaged and the tires are intentionally made to slip. However, when the tires are made to slip, their coefficient of friction, m, temporarily declines, and ideal acceleration is not obtained. Development therefore commenced on a configuration that would do away with unstable elements and employ a direct clamp load control method, applying the clamp load directly by means of hydraulic pressure.

In 2004, Honda worked in collaboration with ZF SACHS Race Engineering GmbH (“SRE” below) to develop a dry multi-plate direct push clutch (DPC) provided with a lever disk plate with a return load that helped to ensure zero torque operation and a lever function that amplified the clamp load on the clutch friction material. A twin-piston clutch actuator in which hysteresis was reduced by lowering friction was simultaneously developed. These developments enhanced

* Automobile R&D Center

the accuracy of clamp load control, and helped to enable slip start to be employed. The reliability of the developed system was confirmed in vehicle tests in 2005, and it was employed on the track from the first race of the 2006 season. This helped to enable the achievement of a maximum reduction of 0.46 sec, corresponding to a distance of approximately 11 m, when accelerating from 0-100 km/h at race start.

2. Development Aims

The mechanism of production of clutch torque is simple, and can be expressed by the following equation:

$$Tc = FN \cdot \mu \cdot rm \cdot i \tag{1}$$

Tc: Clutch torque
FN: Clamp load
μ: Friction coefficient
rm: Effective radius of friction material
i: Number of disk surfaces

The controllability of the clamp load and the stability of the friction coefficient, μ, are therefore important factors in the accurate prediction of clutch torque.

In the conventional pull clutch configuration, the clamp load was generated as the reaction force to compression of the diaphragm spring by the clutch cover (Fig. 1).

The clamp load therefore varied with variations in the characteristics of the diaphragm spring and spring hysteresis, and with variations in the dimensions of the component parts and thermal expansion of the parts, making accurate control a challenge.

The development project discussed in this paper was conducted with the aim of doing away with factors introducing uncertainty into clamp load control and thus of enhancing the accuracy of clutch torque prediction, in order to allow slip start to be used in races and to obtain a competitive advantage over competitors.

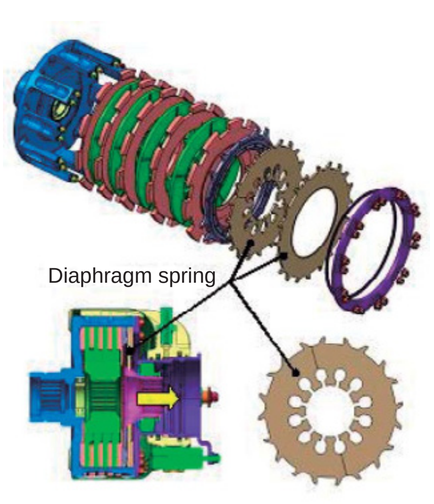


Fig. 1 Pull clutch mechanical section

3. Methods of Achievement of Development Aims

3.1. Development of Direct Push Clutch (DPC) with Lever Plate

Employing a configuration in which the clamp load was obtained directly from hydraulic pressure, the developed system replaced the diaphragm spring used to generate clamp force in the conventional pull clutch with a lever plate (Fig. 2). The lever plate was fixed in place between the foot of the clutch basket and the clutch cover to enable it to produce a return load that would help ensure zero torque. The hydraulic actuator pushed the lever plate, and the lever plate made contact with the pressure plate. The lever plate became a lever with this point of contact as its working point, and a clamp load was generated on the clutch friction material. The clamp load was determined exclusively by the pushing load of the hydraulic actuator and the lever ratio of the lever plate, and there was thus no effect from the characteristics of the diaphragm spring and variations in these characteristics, or variations in dimension due to thermal expansion of the component parts. Because the clamp load was directly determined with the pushing load of the hydraulic actuator as the parameter, the controllability of the clamp load was enhanced, and as a result the accuracy of clutch torque prediction was also enhanced.

3.2. Development of Low-friction Actuator

The reduction of hysteresis in the actuator was an important factor in enabling high-accuracy control of the clamp load using hydraulic pressure to be realized. The major determinant of actuator hysteresis was seal friction, which can be expressed using the following equation:

$$Fr = \mu s \cdot \pi \cdot dp \cdot bs \cdot P \tag{2}$$

Fr: Friction
μs: Friction coefficient of the seal
dp: Piston diameter
bs: Seal width
P: Hydraulic pressure

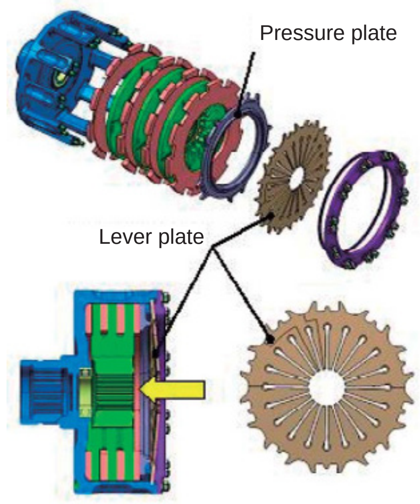


Fig. 2 Push clutch mechanical section

Seal friction is therefore proportional to piston diameter (the seal length), and it can easily be seen that plunger-type pistons present a greater advantage than annular pistons. Taking the friction reduction effect and the potential for prevention of stick by inclining of the piston into consideration, a twin-piston actuator was therefore adopted, reducing seal length by approximately 80% against that of an annular piston (Fig. 3, Fig. 4).

3.3. Clutch Engage System

Engine start was one of concern in using DPC. Because the clutch was fully engaged when the engine was stopped (zero hydraulic pressure) in the conventional clutch system, the engine could be started by externally rotating the layshaft on the gearbox input side, which is positioned coaxially with the crankshaft. However, in the case of a DPC, because no clamp load is generated when the engine is stopped, the crankshaft will not rotate even if the layshaft is rotated.

A clutch engage system (CES) was therefore employed. In this case, a hydraulic pressure storage unit known as a clutch disengage system (CDS), which had previously been used to temporarily disengage the pull clutch to enable a vehicle with a stopped engine to reach a safe stopping point on the track, was used as a system to engage the clutch before engine start.

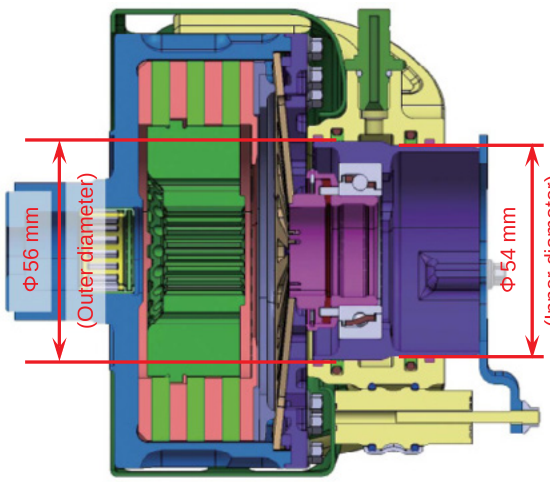


Fig. 3　Push clutch single-piston actuator

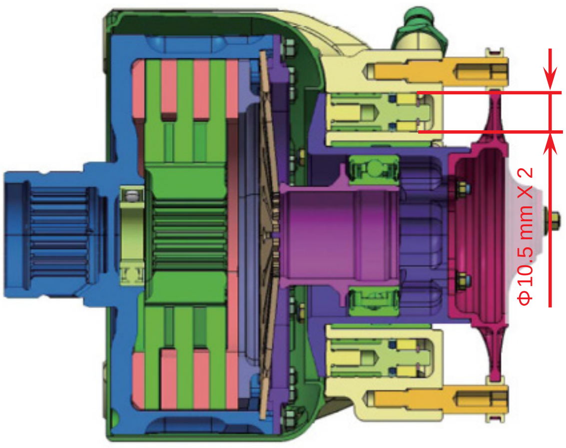


Fig. 4　Push clutch twin-piston actuator

3.4. Organization of Development Process and Allocation of Roles

The development team discussed whether the development process, from clutch to actuator, should be conducted 100% in-house, or whether the development should be conducted in collaboration with a specialist clutch manufacturer. For the reasons listed below, it was decided that the clutch unit would be developed in a joint development with SRE, and that in the case of the actuator, only the parts used in bench tests would be manufactured in Japan and supplied, with the technology of the parts for vehicle tests and race use to be transferred to the race team (HRF1).

(1) The clutch is a consumable part needing day-to-day maintenance. The support from a location close to the race team was therefore judged to be more efficient.

(2) The lead time for manufacturing the carbon material that was used in the clutch was long, and the production of the material in small batches was challenging. In addition, the level of technological challenge was high, and it was therefore decided that in view of the development schedule for the DPC, this aspect of the development should be separated from the DPC project.

(3) In order to incorporate feedback from the race team rapidly, it was judged that transfer to HRF1 would be the most efficient means of maturing the design quickly.

4. Results

4.1. Effect of Use of Slip Start for Race Start

The application of the developed clutch system enabled slip start to be used in the absence of feedback control. Figure 5 compares time for acceleration from 0-100 km/h using slip start and dump start.

A comparison of acceleration times to 100 km/h shows that the use of slip start provided a competitive advantage by helping to enable a maximum time reduction of 0.46 sec, corresponding to a distance of approximately 11 m. This would enable the race vehicle to overtake one lead vehicle by the first corner.

4.2. Characteristics of Clutch Unit

This section will discuss results obtained for the differences in characteristics between the DPC and the

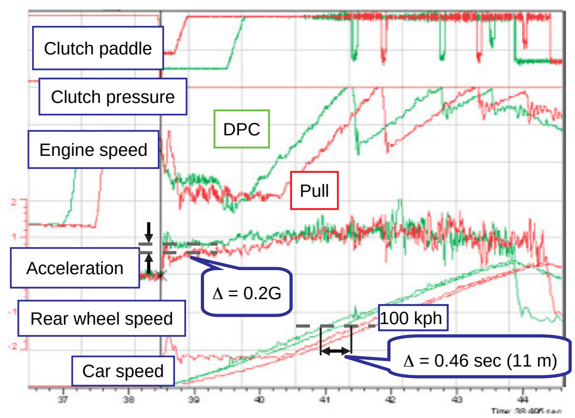


Fig. 5　0 - 100 km/h pull clutch - DPC comparison conventional pull clutch. In the case of the pull clutch, the clamp load of close to 150 daN at race start was in the range for maximum diaphragm spring hysteresis (range of maximum pulling load), and a hysteresis of ±20 daN was generated (Fig. 6). By contrast, the lever plate used in the DPC was employed in an entirely insensitive range to the diaphragm spring characteristic, i.e., a range in which it worked as a lever to amplify the load, and its hysteresis was controlled to around ±10 daN (Fig. 7).

In addition to spring hysteresis, the pull clutch also incorporated other uncertain elements, including changes in the clamp load due to thermal expansion of the constituent parts, variations in the diaphragm spring characteristic, and stroke sensor offset due to heat, which increased the uncertainty of controlling the clamp load.

4.3. Actuator Friction Reduction

Figure 8 shows the results of measurements of clamp load hysteresis against hydraulic pressure for the annular-type single-piston actuator and the plunger-type twin-piston actuator. The twin-piston actuator displayed 80% lower friction in a static test. This is a result of reducing the seal length of the hydraulic piston by approximately 80%.

4.4. Characteristics of System

Figure 9 shows results for the characteristics of the system combining the DPC and the twin-piston actuator. The graph in Fig. 9 plots clutch torque against actuator pressure as measured during tests in circuit simulation mode bench test with an engine. As in the case of the actuator unit tests, torque was output in a linear relationship with hydraulic pressure, with minimal effect from uncertain elements such as wear of the clutch friction material or thermal expansion. These results indicated that the system fulfilled the initial development targets established for it, accurately controlling clamp load throughout its period of use, and helping to enable clutch torque prediction.

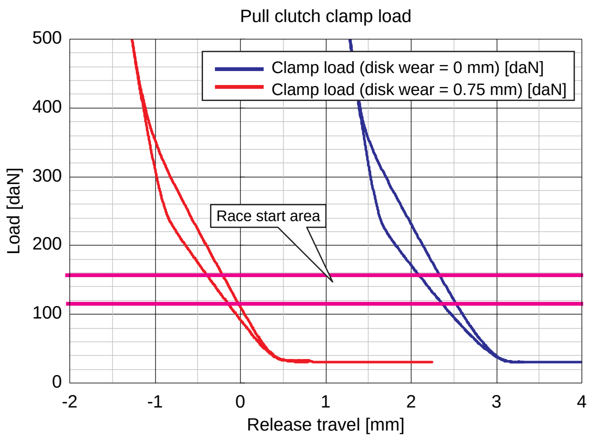


Fig. 6 Pull clutch: Stroke - clamp load

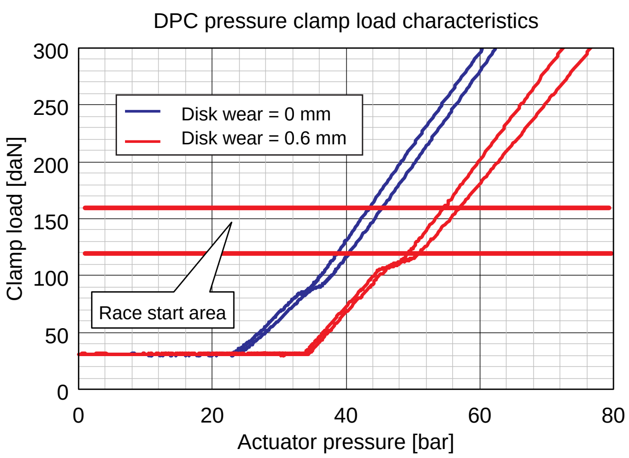


Fig. 7 Push clutch: Pressure - clamp load

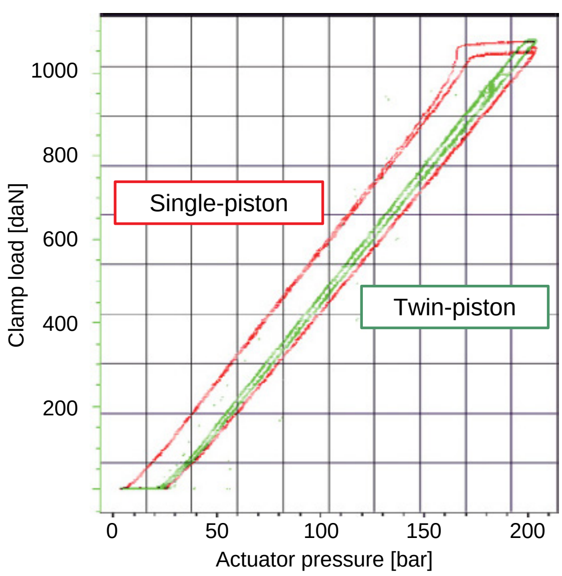


Fig. 8 Actuator pressure - clamp load

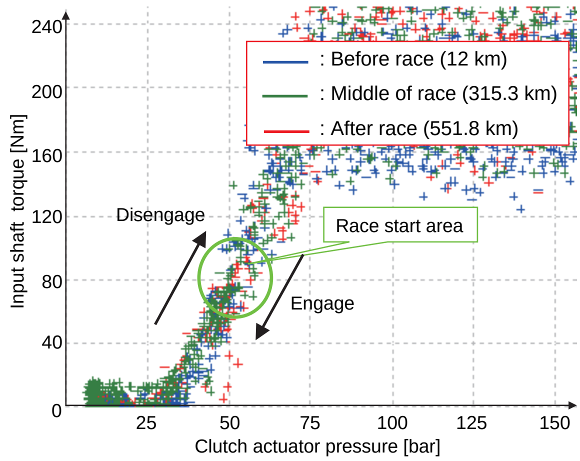


Fig. 9 Clutch pressure - torque

5. Conclusion

Factors resulting in variations in the clamp load and the friction of the seal in the actuator were focused on in order to enhance the accuracy of torque control in a carbon dry multi-plate clutch for use in a Formula One vehicle. The use of a hydraulic direct push mechanism employing a lever plate in the clutch unit and the use of two low-diameter pistons in the thrust load generator of the actuator produced the following results.

(1) A hydraulically-controlled DPC system enabling accurate control of clamp load (clutch torque) was developed.

(2) In 2006, the use of slip start without feedback control was realized, enabling the achievement of a maximum reduction of 0.46 sec (corresponding to approximately 11 m) in time for acceleration from 0-100 km/h.

The following issues for future study can also be indicated.

(1) Variations in clutch characteristics occurred due to friction material wear and the load characteristic of the return spring in the lever plate (Fig. 7). This represents an uncertain element. A wear correction system for the hardware to help ensure that there is no change in the air gaps even when wear occurs would be an ideal solution.

(2) In the range immediately following the generation of the clamp load, a “shelf characteristic” appears by the clearance between the lever plate and the surrounding parts (Fig. 7). This is a factor that increases the complexity of control, and correction of this characteristic is therefore desirable.

(3) While a great deal of research has been conducted on the subject, the friction coefficient of the carbon disk, another element that determines clutch torque, is at present not fully understood. The prediction of the friction coefficient is a major theme in research aimed at producing ideal clutch systems.

Acknowledgments

The authors wish to take this opportunity to express their sincere gratitude to the staff of ZF SACHS Race Engineering GmbH and technical consultant Leo Ress, who collaborated in the development and manufacture of the clutch unit discussed in this paper.

■ Author ■

Koichiro INUKAI

Yasuo NAKAGAWA

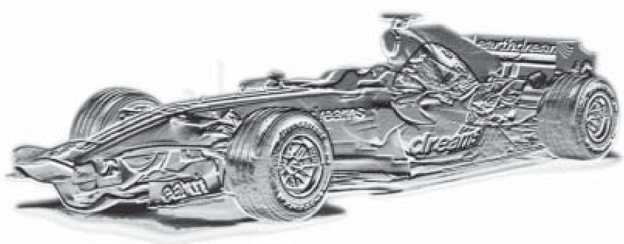
Hiroyuki KUSHIYAMA

Ryuhei KATAOKA

Yoshiki NAGATOSHI

Keiji SHIOTA

Descriptions of Chassis Technologies



Development Methodologies for Formula One Aerodynamics

Atsushi OGAWA* Shujiro YANO* Susumu MASHIO*
Takashi TAKIGUCHI* Shinsuke NAKAMURA* Mitsuru SHINGAI*

ABSTRACT

The greater part of aerodynamics development for Formula One involves optimization using objective functions of downforce and the lift-drag ratio (L/D), and verification of effects using wind tunnels and CFD in the development process. As examples of development methodologies used to advance this optimization process, this paper will discuss aerodynamics development tools, including wind tunnels and CFD, and the conventional development indices that employed these tools. In addition, the paper will introduce new indices for analysis of the effect of tire deformation on the aerodynamic load and analysis of transient aerodynamic characteristics during deceleration, and will also consider the findings made regarding air flows using these indices.

1. Introduction

Downforce can be used to increase the side-force limit on the tires when the vehicle is cornering and the braking-force limit when the vehicle is decelerating. Air resistance is an important factor in determining the acceleration performance of the vehicle. The front-rear downforce balance also contributes to vehicle stability. The purpose of aerodynamics development is to maximize either downforce or lift-drag ratio with consideration of the trade-off between these three elements.

During the development process, the vehicle shape was optimized using a 50% scale model in wind tunnel tests, following which effects were verified using full-scale wind tunnel tests. Simultaneous analysis of aerodynamic phenomena using CFD and particle image velocimetry (PIV) helped to enable the development in the model-scale wind tunnel to move forward in an efficient manner. It also became possible to a certain extent to quantitatively evaluate aerodynamic load using CFD, making it a tool capable of supporting part of the optimization process. The importance of CFD is also increasing as a bridge between wind tunnel tests and the vehicle actually running on a race track. For example, using CFD to reproduce the air flow when the tires were deformed by side force, which could not be reproduced using an actual vehicle in a wind tunnel, produced new findings regarding the air flow around the vehicle running on a race track. Some of these findings were verified in wind tunnel tests.

In recent years, attention is beginning to be directed towards unsteady aerodynamic characteristics when the vehicle is vibrating or decelerating. In Formula One, the maximum deceleration force reaches 5 G during braking, and transient aerodynamic characteristics were frequently an issue during deceleration. It was possible, however, to elucidate the effects of some transient aerodynamic phenomena by means of slow-motion tests in water towing tanks.

2. Aerodynamics Development Tools

This chapter will discuss moving-belt wind tunnel equipment and CFD as aerodynamics development tools.

2.1. The Wind Tunnel

2.1.1. Wind tunnel configuration

The Formula One aerodynamics development process generally used a low speed closed circuit wind tunnel. Open and closed test sections were employed. When closed sections were employed, adaptive walls were used to reduce blockage.

When a boundary layer formed on the floor of the wind tunnel, this boundary layer would sometimes separate prior to the boundary layer underneath the vehicle, due to the adverse-pressure gradient underneath the vehicle. The following countermeasures were put into place to address this issue (Fig. 1).

First, knife-edge and perforated-plate structures were used at the inlet of the test section to remove the boundary layers on the floor. Additionally, downstream

* Automobile R&D Center

from these structures, a moving belt was used to control the occurrence of boundary layers. As a result, ground surface boundary layers were not built above the limit height (approximately 3 mm) of the boundary layer pitot rake measurement, and the free stream was confirmed to be maintained.

To help ensure that the belt was not lifted by the low pressure under the vehicle, suction was applied from underneath the belt when a plastic belt was used, and a tensile force was applied between the front and rear drive shafts when a stainless steel belt was used.

In order to minimize the minute deformation of the different parts of the model when aerodynamic loads were applied, the effect of expansion and contraction of materials due to temperature, and temperature drift in measurement devices, the temperature at the test section was maintained at a constant level during tests. For 50% model tests, the temperature was held at 25˚ C, and a wind speed of approximately 45 m/s was used during the optimization process.

2.1.2. Configuration of wind tunnel models and measurement instruments

This section will first discuss the basic configuration of a 50% model (Fig. 2). The model was supported by a strut from the ceiling. The lower section of the strut was fitted with a pivot to help enable the orientation of the model to be changed, an actuator, and a six-component load cell to measure the aerodynamic load on the model.

The spine (framework) of the model was covered by a replaceable bodywork. The front and rear of the spine were fitted with laser ride-height sensors, to control pitch angle and vehicle height. Three-component load cells were positioned at the bases of the front wings (FW) and the rear wings (RW) to measure downforce, drag, and pitching moment.

The model was also fitted with yaw-, roll-, and steer-control mechanisms, for the reproduction of complex vehicle orientations. In addition, the entire floor of the wind tunnel, including the moving belt, was capable of yaw rotation. The model was also fitted with anemometers at the radiator (vane anemometers or differential pressure gauges), pressure scanners to measure the static pressure at each part, and single-axis load cells able to measure the drag on each wheel. The wheel support method is discussed below.

Two types of wheel support were available: wheel-off and wheel-on. In the wheel-off configuration, the wheels were supported by wheel support arms from outside. The wheels were not in contact with the model, and changes in ride height, therefore, did not generate a reaction force in the model. However, slight inaccuracies remained in geometries, including infinitesimal changes in the camber of the wheels, and the distance between the inner faces of the wheels and the adjacent vehicle parts. The wheel-on configuration used a suspension system with the same geometry as an actual vehicle. The suspension was specially designed not to constrain the motion of the pushrod, and could therefore remove a reaction force from the ground. However, bump and rebound of the suspension would produce a slight amount of friction and a spring force which results in a reaction force to the model and for this reason care was necessary on corrections of the zero point (tare measurements) of the measurement devices.

Formerly, rigid tires with trapezoidal cross-sections (manufactured from aluminum or CFRP) which reproduced the squashed shape close to the contact patch around their entire circumferences were employed. More recently, however, it has become standard practice to use 50% scale rubber tires provided by tire suppliers for wind tunnel model use.

Tare measurements were normally conducted while rotating the belt at a low speed, with a wind speed of zero. Reproducing all the vehicle orientations that were to be measured and taking tare measurements at each measurement point during this process would help enable more accurate measurements. However, this method was disadvantageous in terms of measurement time. In many cases therefore, measurement time was reduced by taking one vehicle orientation as representative for tare measurements, or multiplying by the linear approximation of the ride height at this representative orientation and the highest and lowest ride heights.

2.1.3. Flow visualization techniques

PIV measurements were conducted with the technical assistance of the Honda Fundamental Technology Research Center. This section will discuss the set-up for three-dimensional PIV measurements.

A seeding device was positioned behind a heat exchanger, in a space between flow-smoothing honeycombs. The seeding particles were glycol-based, and had a diameter of 1 micrometer. The laser optical system, situated outside the path of the wind, illuminated the periphery of the model. By changing the path of the optical system and the position of the cameras, it was

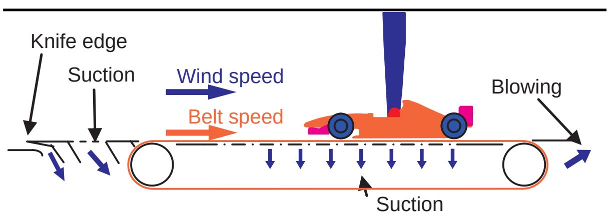


Fig. 1 Moving belt configuration

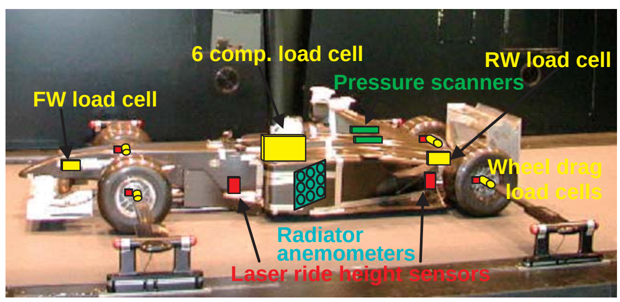


Fig. 2 50% model measurement system

possible to take measurements in the x, y, and z directions (Fig. 3). However, there were limits to the ability to take measurements in areas where the laser could not physically reach, such as the x cross-sections of the inboard faces of the tires.

2.2. Computational Fluid Dynamics

As indicated above, the importance of CFD in Formula One development projects has today grown to rival that of wind tunnels. The point to be observed in the use of CFD is whether the air flows around vehicles in wind tunnels and on a race track are qualitatively and quantitatively reproduced. To provide an example, this section will consider the method used when obtaining a correlation between wind tunnel and CFD results for a wake behind a tire.

The air flow around a rotating tire was elucidated in a wind tunnel using a tire and PIV measurement equipment. Figure 4 shows a vector diagram for a tire center-section obtained using PIV measurements. Here, the position of the commencement of separation from the top of the tire and the size of the wake will be focused on. (a) shows the results of PIV measurements, and (b) and (c) show CFD results. The surface mesh was approximately 1 mm, and the realizable k-ε model was used as the turbulence model. (b) shows results for y^+-1, and (c) shows results for y^+-10. y+ expresses the state of the flow as a dimensionless number calculated from the friction velocity at the nearest wall, the distance to the nearest wall, and local kinetic viscosity. For the simulation of separation, it was taken as effective to place the first mesh on the viscous sub-layer (y^+-1). The height of the first mesh for Y^+-1 was 10μm.

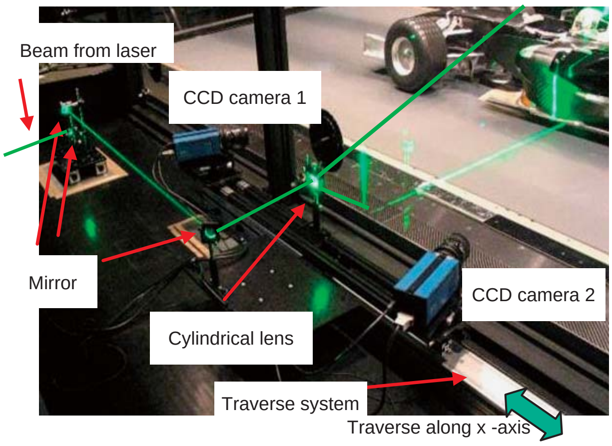
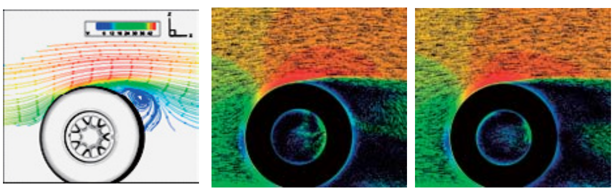


Fig. 3 PIV setting



(a) PIV (b) CFDy$^+$ - 1 (c) CFDy$^+$ - 10

Fig. 4 Velocity vector at tire center section

Considering the results, (b) shows a good separation position and wake size, but the wake is excessively large in the case of (c). The result is in accord with the simulation settings in which the air speed around the tire was over-estimated.

As in the case of the tires, simulation methods for other parts were also studied, and it was decided that the following simulation methods would be used for the full-vehicle simulation:

The tetra mesh was used as the basic elements, and prisms were created from surface triangle mesh on the wings. 1 mm surface mesh was used for the airfoil sections, and 2-4 mm surface meshes were used for other parts. Figure 5 shows the mesh used for the FW. The actual first height of the boundary layer mesh was 10 μm.

Using a cocoon around the periphery of the vehicle, the surface mesh size was increased to 10 mm. Figure 6 shows an image of a cocoon. Total pressure on the cocoon was largely controlled within an overall iso-surface pressure of CpT=0.7.

In the case of the volume meshes, the mesh number differed depending on the growth rate from contiguous meshes. A growth rate of 1.7 would generate a mesh number of about 50 million, while a growth rate of 1.1 would produce a mesh number of about 300 million. Figure 7 shows the difference in space mesh density produced by a difference in the growth rate. It is clear

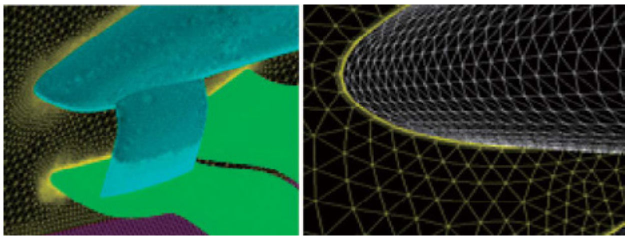
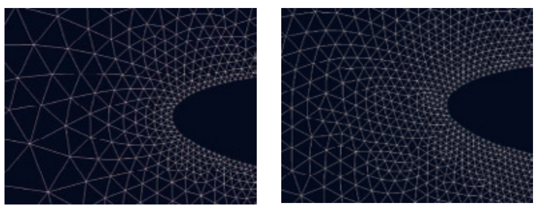


Fig. 5 Surface mesh of FW and symmetry plane

Fig. 6 Cocoon (iso-surface of CpT =0.7)



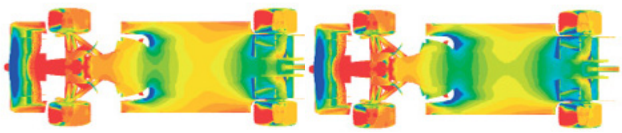
(a) Propagation 1.7 (b) Propagation 1.1

Fig. 7 Mesh propagation

that a low growth rate helps to enable a high mesh density to be maintained even in areas distant from the object.

Figure 8 shows the Cp distribution under the vehicle. It was possible to maintain a higher mesh density around the vehicle in the case of (b), which was generated using a cocoon, than was the case for (a), generated without using a cocoon. This minimized the effect of numerical diffusion and, as a result, produced strong suction. This indicates the effectiveness of using a high mesh density and conducting the simulation within the effective wave numbers.

A Linux device with a 128 GB memory was used to run PrePost. The calculation server was equipped with a 1024-core Xeon® 3.0 GHz CPU, using InfiniBand interconnects. At a growth rate of 1.1 generating 300 million meshes, 24 hours was required to converge enough for FLUENT 6.3 with 256 parallel computations.



(a) Propagation 1.7 (b) Propagation 1.1

Fig. 8 Cp contour of underbody, range from -2 to 0.

3. Wind Tunnel Optimization Indices

3.1. Regulations

The major part of aerodynamics development for Formula One consists of optimization of the form of the vehicle within the scope allowed by the FIA regulations. In addition to specifying the “regulation box,” the scope within which any aero parts must fit (Fig. 9), the regulations also contain a flat-bottom rule prohibiting the three-dimensional shape of the vehicle floor, a shadow rule prohibiting the use of wings and other projections that would provide ground effects in certain areas, and a rule specifying the number of RW vanes.

The regulations are also revised every few years in order to control vehicle speed increases resulting from the ongoing evolution of aerodynamics.

3.2. Ride Height Map

3.2.1. The ride height map defined

Using an advanced control logic, a simulation was conducted of the 50% model in the wind tunnel at the ride height (RH) of an actual vehicle, and aerodynamic performance was evaluated as the arithmetic mean of multiple RH. These multiple vehicle orientations form what is termed an RH map, which plays an important role in determining in what direction the vehicle can be optimized. Figure 10 shows an example of an RH map.

The most basic type of RH map is formed by selecting representative RH figures that enclose groups of RH measurements drawn from circuit data. The number of RH points per map varies with the purpose of the map. The maps used in development feature a comparatively low number of RH, while the maps used in vehicle dynamics analyses (termed “mapping data”) are selected to incorporate a larger number of RH.

Using these RH maps, vehicle downforce was optimized in relation to target drag for each circuit.

3.2.2. Adding yaw (cross-winds), steer, and roll

The recent advancement in wind tunnel facilities and the associated control technologies have enabled yaw, steer, and roll to be added to the tests in addition to RH control. This has helped to enable aerodynamics development using vehicle orientations closer to actual conditions. Yaw (or cross-winds) and steer have become indispensable elements in recent aerodynamics development for Formula One, and developments are now conducted using complex vehicle orientations in a single RH map in which yaw, steer, and roll are interwoven.

3.2.3. Trajectory map and curved flow

Because the aerodynamic load on a vehicle is proportional to the square of its speed, it is sometimes thought that the only case in which aerodynamics dominates vehicle performance is that of high-speed corners taken at close to 250 km/h. However, today it is understood that the effect of aerodynamics is not insignificant even in the case of low- and medium-speed corners.

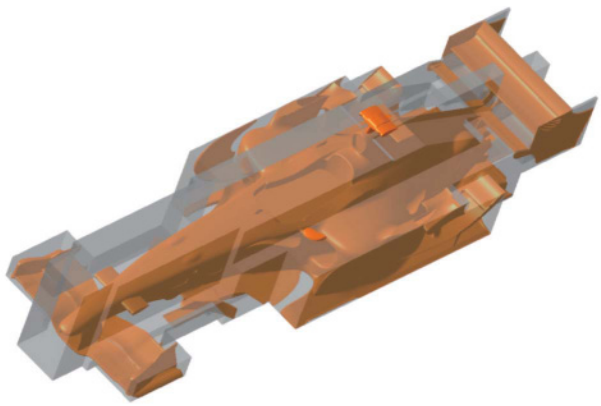


Fig. 9 Regulation box and F1 chassis

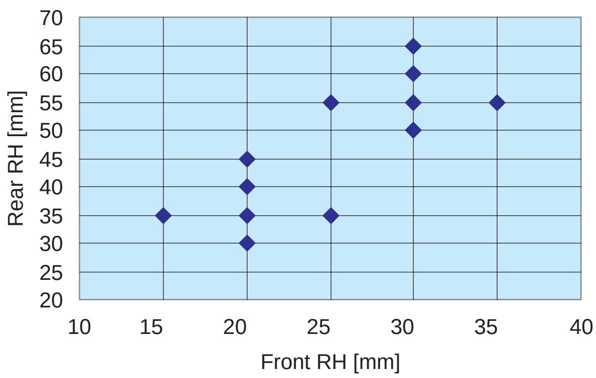


Fig. 10 Ride height map

Given this, together with the evolution of systems for controlling changes in vehicle orientation, a method of discretely reproducing a sequence of vehicle orientations linked to cornering while braking and accelerating on an RH map was developed. This is termed a trajectory map.

During actual cornering, a vehicle turns at a specific angular velocity. Because of this, the air flows at the front and rear of the vehicle have different inflow angles. The angular velocity is greater in the case of a Formula One vehicle than a normal car, and its effect therefore cannot be ignored. The difference in the front-rear inflow angle is particularly pronounced during low- and medium-speed cornering, when the angular velocity is high in relation to vehicle speed, and viewed from the perspective of a vehicle, the air flow appears bent. This is termed a curved flow (Fig. 11).

Curved flow could not, of course, be reproduced in wind tunnel tests. CFD was therefore used to investigate its effect, and the phenomenon was responded to by adding a type of yaw correction to the trajectory map.

3.2.4. Weighting and penalties

Trajectory maps functioned very well to reproduce the characteristics of specific corners, but lacked flexibility in terms of evaluation of the overall dynamic performance of the vehicle attributed to aerodynamics. In addition, the importance of minimizing shifts in the vehicle's aerodynamic center (CoP) when the vehicle orientation changes was reconfirmed from the perspective of vehicle dynamic performance.

Given this, the viable options were weighting of the RH and the application of penalties in response to shifts in the CoP. Weighting of the RH involves consideration of importance of low-, medium-, and high-speed corners, and employing a weighted means weighted towards RH corresponding to the corners where the level of contribution to lap times is higher. The application of penalties is a method of quantifying and evaluating the level of robustness to changes in vehicle orientations, in which shift of the CoP as a result of changes in vehicle orientation is considered as a reduction in effective downforce.

Even today, methods of evaluation of aerodynamic performance and the vehicle dynamic performance it generates are still being explored at the wind tunnel test stage.

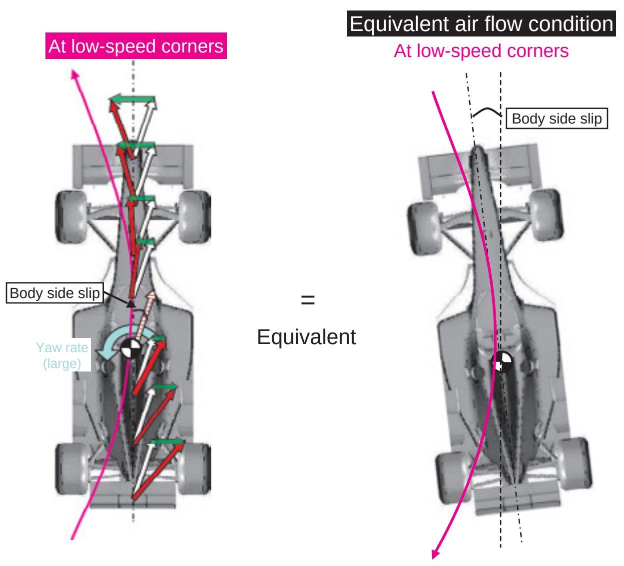


Fig. 11 Curved flow

4. Formulation of New Development Indices

This chapter will discuss the new development indices to be analyzed, the effect of tire deformation on aerodynamic load and the transient characteristics of the aerodynamic load produced on the RW during deceleration.

4.1. Effect of Tire Deformation on Aerodynamic Characteristics

When the vehicle is cornering, accelerating, or decelerating, the tires are constantly deformed due to vertical, lateral, and longitudinal forces (Fig. 12). However, the only loads that can be applied to the tires during full-scale wind tunnel tests are the vertical force from the weight of the vehicle and the aerodynamic downforce. Similarly to model-scale wind tunnel tests, it is only possible to apply vertical loads from the wheels and tires, and some side force from the belt. In other words, the deformation of the tires due to side force or longitudinal force when the vehicle is cornering, accelerating or decelerating cannot be accurately reproduced in a wind tunnel. It was therefore necessary to study the effect of tire deformation on the aerodynamic performance of the vehicle when it is actually running on a race track, and use the data obtained to formulate new indices for aerodynamics development, such as methods of reproducing tire deformation effect in a wind tunnel. This new index then enables an aerodynamically robust vehicle against tire deformation to be developed.

First, the forces acting on the tires when the vehicle is running on a track were reproduced in rig tests, and the shapes of deformed tires were obtained using three-dimensional shape measurement systems. Using these shapes, CFD was employed to analyze air flows, helping to clarify the effects of the deformed tires to the aerodynamic performance of the vehicle. Next, a method of reproducing these flows in a wind tunnel was formulated.

This chapter will focus on tire deformation due to side force, which displayed the highest level of contribution to vehicle aerodynamics.

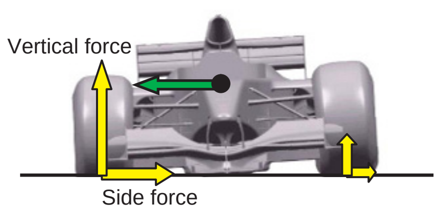


Fig. 12 Forces applied at cornering

4.1.1. Circuit data

The forces acting on the tires when the vehicle is in operation were calculated from the pushrod load and other parameters. As an example, Fig. 13 shows the side force on the left front tire during a single lap of a circuit obtained from circuit data, and Fig. 14 shows the vertical position of the tire axis resulting from the force acting on the tire. The maximum side force acting on the outer tire reaches a maximum of 10000 N, and the vertical displacement of the tire axis, i.e., the amount of vertical squash displaced by the tire, is as high as 25 mm due also to the vertical force from the downforce acting at the moment. In addition, during deceleration, a load of approximately 5000 N acts on the tire in a rearward direction.

4.1.2. Shape measurement of deformed tires

The loads acting on tires for vehicles running on track were reproduced in rig tests (Fig. 15), and the shapes of the tires were measured using three-dimensional measurement systems, including FARO, Vectron, and T-Scan. However, because these were static loads, the deformation of the tire due to centrifugal force during rotation was not reproduced. As an example, Fig. 16 shows the shape of a tire subjected to a side force of 7000 N. The side wall of the tire is deflected by approximately 20 mm in the lateral direction at close proximity to the road surface.

4.1.3 Analysis of flow when front tire is deformed by side force

The measured front tire shape was analyzed using CFD. This analysis showed that, in comparison to a tire with no side force acting on it, approximately 5% of the vehicle’s downforce was lost when a side force of 9000 N acted on the tire. Figure 17 shows the total pressure distribution close to the road surface with and without a side force acting on the tire. The results show that the position of the separation point on the outboard-side wall of the tire moves back significantly when a side force acts on the tire.

This backwards shift of the separation point changes the circulation around the tire in the XY planes, and the tire wake which previously flowed to the outboard of the vehicle now flows under the vehicle (Fig. 18). The fact that this reduces the dynamic pressure underneath the

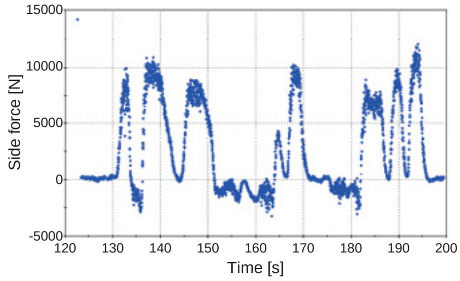


Fig. 13 Front tire side force on track

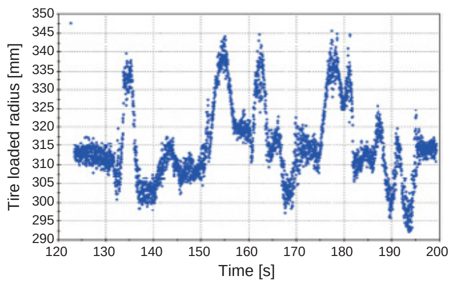


Fig. 14 Front tire loaded radius on track

Fig. 15 Tire test rig

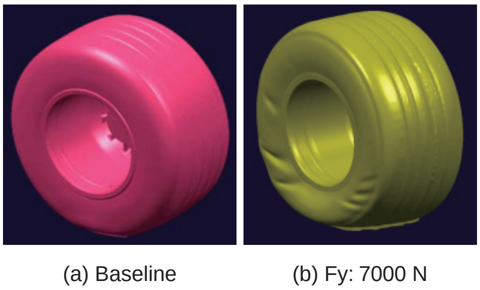


Fig. 16 Snapshots of scanned tire

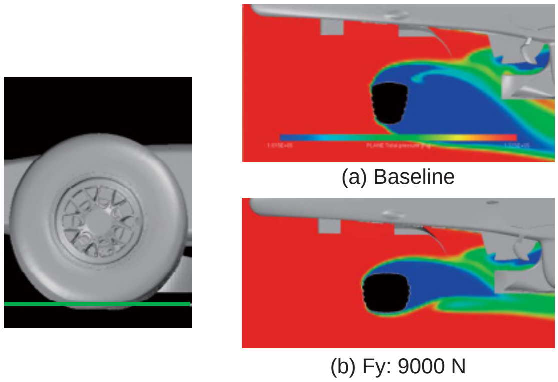


Fig. 17 Total pressure distribution

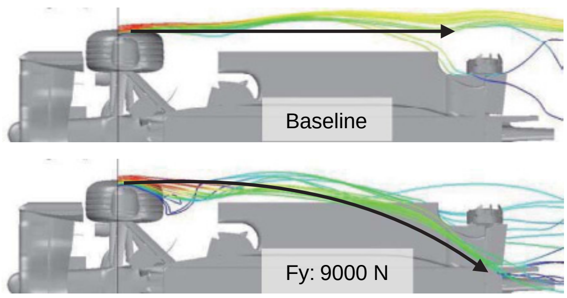


Fig. 18 Tire wake direction

vehicle, resulting in a decline in downforce, can be seen from the change in the static pressure underneath the vehicle when the tire goes from a state of no side force to one in which side force is acting (Fig. 19).

Next, the reason for the shift of the separation point was analyzed. First, a simulation was conducted to apply the surface form of a tire with no side force acting on it to outboard side of a tire with a side force applied (Fig. 20). The resulting forward shift of the separation point resulted in a recovery of 4% of the 5% loss in downforce.

Next, the shape of the tire contact patch was focused on, and a simulation was conducted with an edge applied to the corner of the leading edge of the outboard side of the tire with a side force applied (Fig. 21). In this case, 4.5% of the downforce was recovered as well, producing a result almost identical to that for a tire with no side force acting on it.

Next, this phenomenon was reproduced using a 50% model in a wind tunnel. However, because side force cannot be applied to the tires in a wind tunnel, at first only the vertical displacement was reproduced.

Figure 22 shows the results of measurement taken using two-component PIV. As in the case of the CFD results, the deformation of the tire resulted in the backwards shift of the separation point on the outboard-side wall of the tire. It also showed 3% loss of downforce, which agrees with the CFD results qualitatively as well.

Despite differences in the exact shape of tire deformation, both CFD and wind tunnel results showed that the deformation of the tires produces a backwards shift of the separation point, which causes the loss of downforce on the vehicle.

4.1.4. Analysis of flow when rear tire is deformed by side force

CFD was used to conduct an analysis for the rear tires in the same way as previously conducted for the front tires. The results showed that approximately 4% of the vehicle's downforce was lost when a side force of 6500 N was applied, as compared to a state in which no side force was applied. Figure 23 compares the total

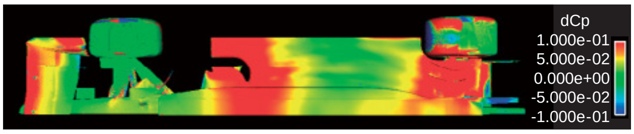


Fig. 19　Deformed tire effect (delta Cp)

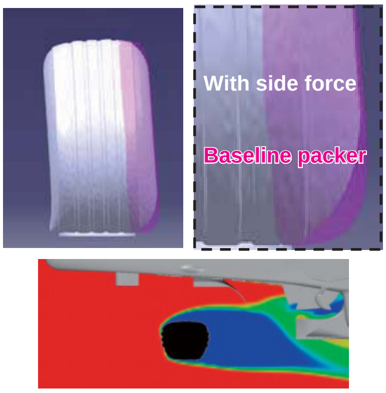


Fig. 20　Result of baseline packer

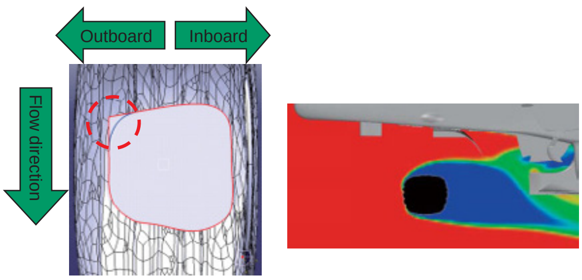
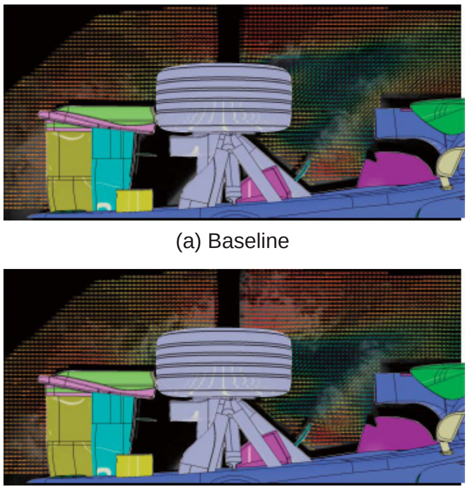


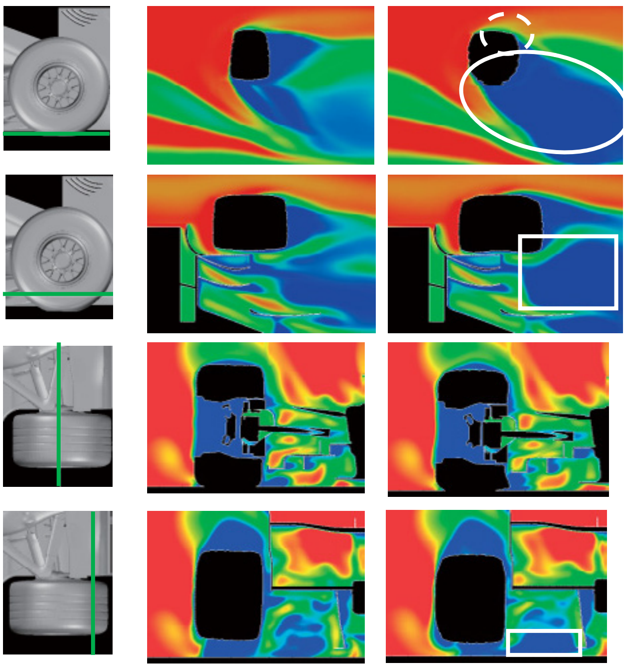
Fig. 21　Result of edged contact patch corner



(a) Baseline

(b) 10mm squashed

Fig. 22　PIV velocity vector



(a) Baseline　(b) Fy: 6500 N

Fig. 23　Tire wake comparison (total pressure)

pressure distribution close to the diffuser for tires with no side force and a side force of 6500 N. When the side force is applied, the separation area of the inner side of the tire becomes large (shown by the solid circle in the figure), simultaneously, as was the case for the front tire, the separation point shifts to the rear (shown by the broken circle in the figure), and the wake of the tire flows to the center of the vehicle (shown by the solid square in the figure). The fact that this becomes a blockage and results in the decline of the diffuser flow, with a consequent decline in the vehicle's downforce, can be judged from the change in the static pressure underneath the vehicle when the tire goes from a state of no side force to one in which side force is applied (Fig. 24).

4.1.5. Verification of effect using full-scale wind tunnel tests

The results obtained from CFD and model-scale wind tunnel tests were verified using an actual vehicle in the wind tunnel. Due to the system limitations of the wind tunnel, it is impossible to apply any loads to the tires other than the weight of the vehicle and the downforce. Therefore, changes in the downforce produced by changes in wind speed were used to vary the amount of squash of the tires in the vertical direction, the effect of which was analyzed. At this time, RH was adjusted by varying the length of the push rods.

Figure 25 shows the amount of squashing of the front tires and the rate of change of normalized downforce against wind speed. The downforce at a wind speed of 30 m/s was employed as the standard for the rate of change of downforce. Normalized downforce declines with the squashing of the tires as the downforce increases with increasing wind speed. This matches the tendency of results from CFD and the model-scale wind tunnel tests, as discussed in Section 4.1.3. In addition, PIV confirmed that this change in the normalized downforce occurred with the shift of the separation point on the outboard side of the tire.

Figure 26 shows the shape of a front tire when the downforces generated at vehicle speeds of 40 m/s and 70 m/s were statically loaded on it. The results show that the increase in load relaxes the apex of the outboard side of the contact patch, and increases the degree of deformation of the bottom of the tire.

However, in addition to the effect of the deformation of the tires, the effects of deflection of bodywork due to wind and variations in the position of the suspension arms are also included in the changes in the downforce, and these effects, however small they may be, need to be taken into account.

4.1.6. Effect of differences between tire manufacturers

The shape of the deflection of the side of the tire and the shape of the contact patch also differ between tire manufacturers, and the effects of these parameters therefore also differ. For example, if tires manufactured by maker A and maker B which display an identical amount of squashing are compared using CFD, the vehicle downforce when maker B's tires are used will be approximately 5% lower. As Fig. 27 shows, this is because the separation point on the outboard of maker B's front tires shifts rearward, and the wakes from the inner sides of the maker's rear tires are large. This tendency matches the changes in magnitude of the normalized downforce in the full-scale wind tunnel tests

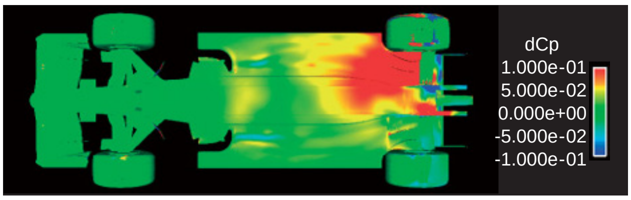


Fig. 24 Deformed tire effect (delta Cp)

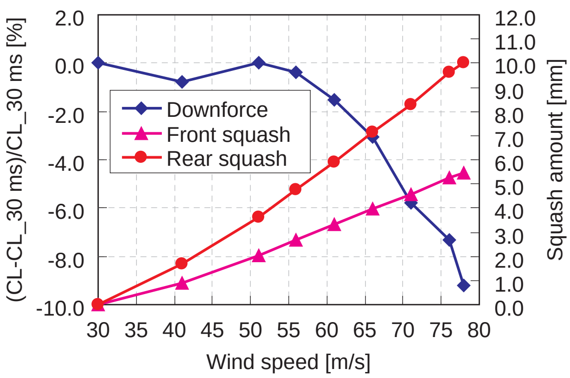


Fig. 25 Squash and downforce change

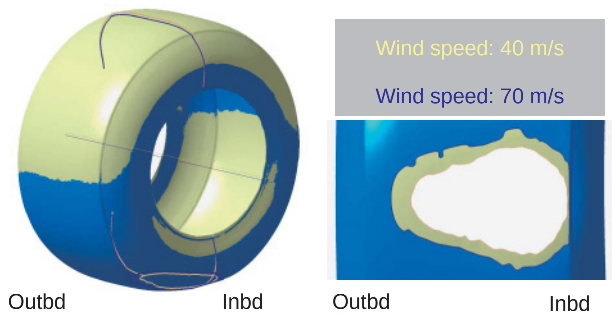


Fig. 26 Snapshot of scanned tire

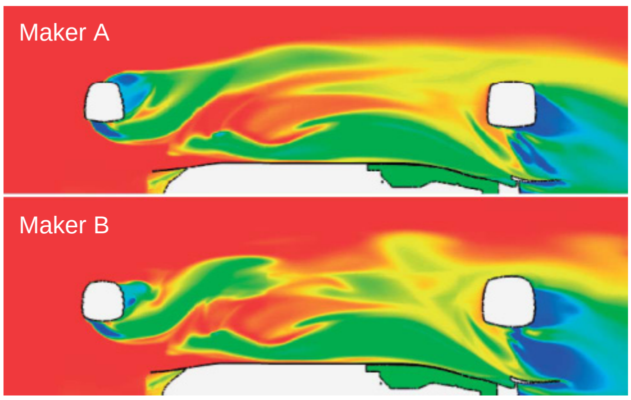


Fig. 27 Total pressure under floor

(Fig. 28). However, because the shape of the contact patch also changes with the camber and other parameters, a conclusion cannot be drawn until these other factors are taken into consideration.

4.1.7. Methods of reproducing tire deflection in wind tunnel

In order to obtain stable aerodynamic performance on the track, it was necessary to increase robustness against tire deformation. This made it necessary to reproduce worst-case scenarios in wind tunnel development. It was considered that using the pushrods to apply vertical loads to the tire, or, as shown in Fig. 29, using a roller to push the side wall from the inner side of the tire, would be effective methods of doing so.

4.2. Transient Aerodynamic Characteristics during Deceleration

Data obtained during track test had indicated the possibility that normalized downforce declined when the vehicle was decelerating. In particular, it was possible that the normalized downforce was declining because of the development of a boundary layer at the RW due to deceleration and the fact that the RW wake overtook it. An actual RW was therefore used in slow-motion tests in a water towing tank, and the occurrence or non-occurrence of separation and the changes in load when the vehicle was decelerating were analyzed.

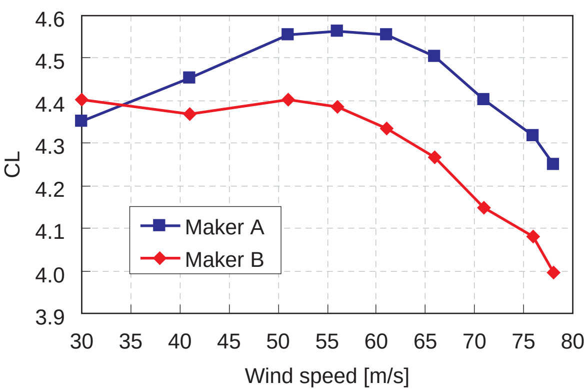


Fig. 28 Normalized downforce comparison

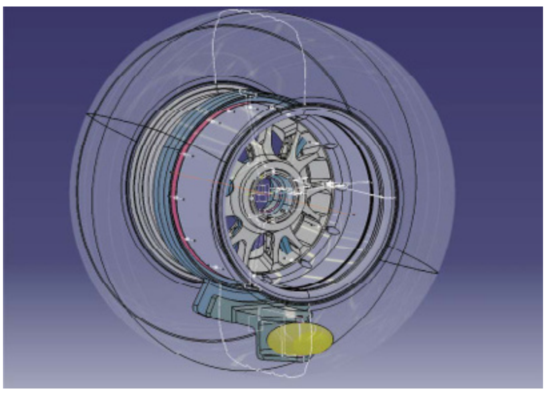


Fig. 29 Method of deforming rubber tire in wind tunnel

4.2.1. Test conditions

First, in order to make non-aerodynamic factors such as deformation of the RW under loads correspond to conditions when the vehicle was actually running on a race track, the aerodynamic loads in a towing tank and that on a track were coordinated. If the forces in air and in water are coordinated, i.e., the dynamic pressure is made constant,

$$q = \frac{1}{2}\rho U^2 = const$$

where ρ: density and U: speed,

then at an atmospheric temperature of 25˚C and a water temperature of 15˚C, the speed in water will be

$$U_{water} = \sqrt{\frac{\rho_{air}}{\rho_{water}}} U_{air} = 0.034 U_{air}$$

i.e., the speed in water will be one-thirtieth of that in air. The Reynolds number at this time is given by

$$\mathrm{Re}_{water} = \frac{U_{water}\nu_{air}}{U_{air}\nu_{water}} \mathrm{Re}_{air} = 0.47\ \mathrm{Re}_{air}$$

where ν: dynamic viscosity coefficient. The Reynolds number in water is half its value in air, but considering changes in vehicle speed and the results of model-scale wind tunnel tests, this was in an allowable range.

When the Froude number is made constant,

$$Fr = \frac{U^2}{gL} = const$$

where g: acceleration and L: reference length, the acceleration in water is expressed by

$$g_{water} = \frac{U^2_{water}}{U^2_{air}} g_{air} = 0.0012 g_{air}$$

i.e., acceleration in water is 1/1000 of that in air.

Given this, it is possible to conduct slow-motion tests in water, and transient aerodynamic loads during deceleration, which are challenging to measure when the vehicle is actually running on a race track due to error factors, can be measured with a high degree of accuracy. During this program of tests, deceleration tests from 0.005-0.05 G and fixed-speed tests from 1.02-2.94 m/s (corresponding to 106-307 km/h in air) were conducted.

4.2.2. Towing tank and test rig

The towing tank employed in the tests was 200 m long, 10 m wide, and 5 m deep (Fig. 30). A six-

Fig. 30 Towing tank

component load cell and a mounting rig were attached to the towing train, and the RW was positioned upside-down in 1 m of water. To help prevent waves from being generated on the surface of the water, an acrylic plate was positioned close to the surface (Fig. 31).

4.2.3. Test results

In order to investigate the existence of transient aerodynamic characteristics during deceleration, the RW downforce when decelerating from 2.94 m/s (corresponding to 307 km/h in air) at 0.005 G (corresponding to 4.2 G in air) was compared with values for driving at a constant speed (Fig. 32). The same tests were conducted with braking G-force varied down to 0.05 G. No transient characteristics during deceleration were observed during these tests, verifying that the RW was fulfilling its function as a unit according to design targets even when the vehicle was decelerating.

The result indicated that towing tank tests were an effective method of verifying performance on a track

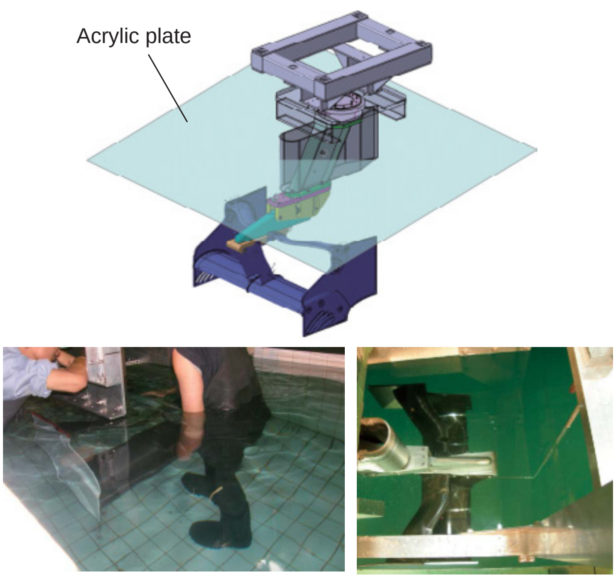


Fig. 31 RW assembly

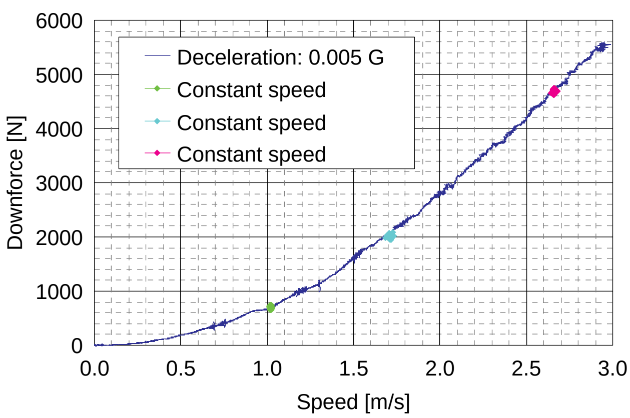


Fig. 32 Hydrodynamic force comparison between deceleration and constant speed modes

when, for example, dealing with higher-load RW or studying aeroelasticity. In addition, the towing tank was an effective means of reproducing new fluid phenomena, such as when studying the possibility of a decline in the normalized RW downforce during deceleration due to the rear-tire wakes overtaking the RW.

5. Afterword

This paper has discussed aerodynamics development indices used until 2008, and new indices such as the analysis of the effect of tire deformation on aerodynamic loads and the analysis of transient aerodynamic characteristics during deceleration. These efforts deepened understanding of the air flow around the vehicle actually running on a race track.

Acknowledgments

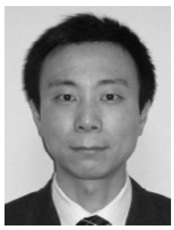
The authors wish to take this opportunity to offer their sincere thanks to all the staff members of the former BAR and HRF1 for their many years of assistance in aerodynamics developments, and also to the staff of IHI Corporation who generously offered their advice and assistance in the towing tank tests.

■ Author ■



Atsushi OGAWA   Shujiro YANO   Susumu MASHIO





Takashi TAKIGUCHI   Shinsuke NAKAMURA   Mitsuru SHINGAI

Aerodynamics Analysis of Formula One Vehicles

Atsushi OGAWA* Susumu MASHIO* Daisuke NAKAMURA*
Yasutaka MASUMITSU* Masayuki MINAGAWA* Yusuke NAKAI*

ABSTRACT

Formula One vehicles are fitted with a variety of aerodynamic devices. This produces complex mutual interference in the air flows around the vehicles, generating highly nonlinear flows. The clarification of these aerodynamic phenomena helps to enable efficient optimization of aerodynamic devices. This paper will provide some examples of findings regarding the air flows around Formula One vehicles obtained using wind tunnels and CFD.

1. Introduction

The shapes of the aero parts employed on Honda Formula One vehicles were optimized using 50% model wind tunnel tests and CFD[1]. The chief consideration during this process was the tradeoff between downforce, the front-rear balance of downforce, and drag. Mutual interference between the air flows around these aero parts produced a strong non-linearity in the air flows around the vehicles, and in practice it was not possible to clarify the mechanism of all aerodynamic phenomena. However, the use of CFD to analyze the core aerodynamic phenomena did generate new findings concerning the air flows around Formula One vehicles. This paper will discuss some of these findings.

2. The Entire Air Flow around the Vehicle

2.1. Ground Effects

This section will provide a simple discussion of two-dimensional and three-dimensional ground effects. The chief two-dimensional ground effect is an increase in dynamic pressure and a decrease in upwash (a change in the angle of the wake in an upward direction) due to mirror images of the wings (Fig. 1). Because the upwash of the wing is the angle of attack for aero parts behind it, it plays a particularly important role in aerodynamics involving multiple bodies, as in Formula One vehicles. A reduction in upwash and a reduction in induced drag due to mirror images of tip vortices can be indicated as three-dimensional ground effects (Fig. 2).

* Automobile R&D Center

2.2. The Entire Flow around the Vehicle

For the reasons discussed in Section 2.1., in addition to generating downforce close to the ground, it is also important to actively generate lift, which may initially appear disadvantageous, in areas more distant from the ground. In parts in which the influence of ground effects is minimal, a significant downwash can be generated by a small lift, and this can be used to increase the downforce of the rear parts. To do so, lift-generating parts called “bunny ears” and “fox ears” (see Chapter

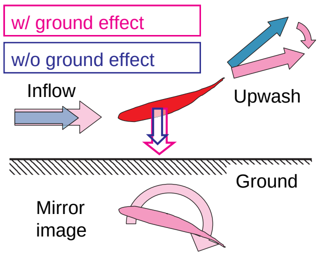


Fig. 1 2D ground effects

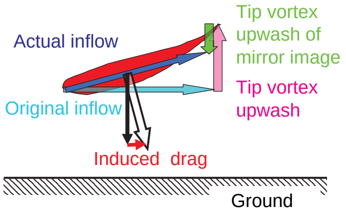


Fig. 2 3D ground effects

4) were fitted on the upper part of the vehicle in order to direct the upwash from the front wing (FW) downwards and boost the performance of the floor, the diffuser, and the rear wing (RW) (Fig. 3). The front suspension arm and the nose also play a role in generating downwash. This can be seen from the static pressure distribution on the upper and lower surfaces of the vehicle (Fig. 4).

This section has offered a rough picture of the air flow around the vehicle, but the actual flow is composed of countless small and large flow phenomena. The following pages will present a discussion of some of these phenomena.

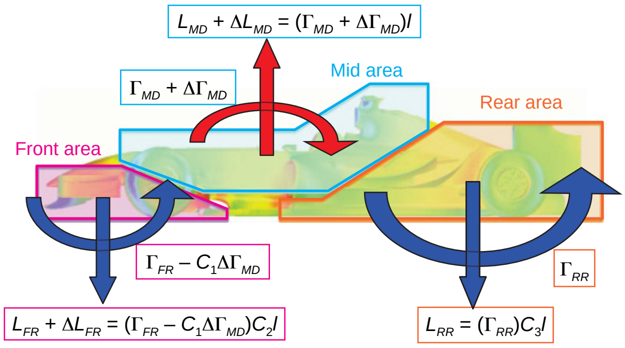


Fig. 3 Ideal chassis circulation

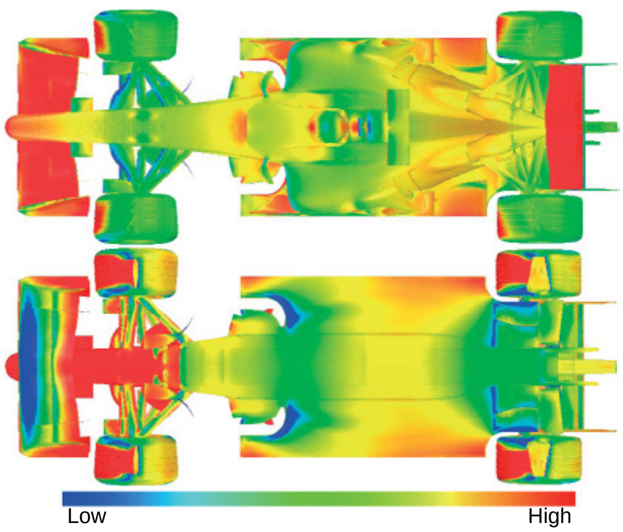


Fig. 4 Surface static pressure

3. Front Wings (FW)

The downforce produced by the FW (Fig. 5) represents approximately 20-25% of the downforce of the entire vehicle. In addition, the aerodynamic balance (CoP) of the vehicle is extremely sensitive to the FW downforce, and it therefore plays a role in the adjustment of the CoP of the entire vehicle, through alteration of the flap angle.

Because the FW is positioned upstream, the upwash and tip vortex it produces have a significant effect on flows at the rear of the vehicle, and they also display a high degree of sensitivity to changes in vehicle orientation produced by steer, pitch, and other factors. These characteristics, in addition to the fact that the range of adjustment of the CoP must be considered, make the FW the device on which most time is expended in wind tunnel development.

3.1. Effect of FW Downforce and Consequent Upwash

The ground effect of the FW has a beneficial effect on the lift-drag ratio, and also controls upwash to a certain extent. However, if the downforce generated by the FW is too strong, the limit of the lift generated by the suspension arm and other parts behind the FW can be reached, and upwash from the FW can reduce the angle of attack on the devices at the rear of the vehicle, and consequently reduce the downforce of the vehicle. In addition, the wake from the FW and the suspension arm can also reduce the dynamic pressure close to the RW, resulting in a decline in the RW downforce (Fig. 6). Because of this, it was necessary to control the downforce of the FW to a specific value or below, rather than maximizing it. The surplus that this produces in the regulation box[1] was used for a variety of purposes, including the reduction of steer sensitivity.

In concrete terms, the height and shape of the FW were adjusted in the Y direction in order to help improve the upwash distribution and the steer sensitivity. It was necessary to optimize the wing tips in coordination with the front wing end plate (FWEP), which will be discussed below.

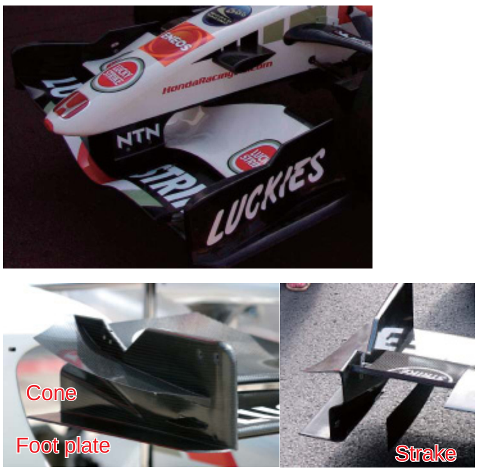


Fig. 5 Front wing

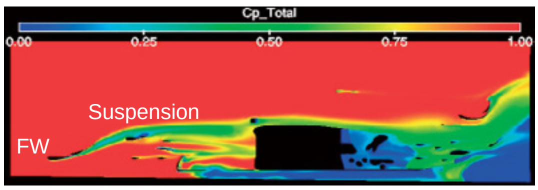


Fig. 6 Suspension stall (CpT, Y=300 mm)

3.2. FW Tip Vortex and Tires

Another issue in FW development was the treatment of the tip vortex shed by the FWEP. Until 2008, the FWEP vortex was shed to the lower sections of the inboard faces of the tires. A separation vortex generating a significant total pressure loss was shed from the leading edges of the lower sections of the inboard faces of the tires (Fig. 7). One measure considered was to merge the separation vortex and the FWEP vortex; the merged vortices would remain in the same position (the YZ position)as they go downstream, which would prevent the vortices from going into the inboard side of the vehicle and help to prevent the loss of dynamic pressure underneath the vehicle floor (Fig. 8). It was necessary for a stable FWEP vortex to be shed at an appropriate position and at a strength at which vortex breakdown would not occur.

In addition, the effect of steer could not be ignored in FW development. Steer accompanies yaw angles in the flows, and FW was optimized to help ensure that it would not result in large variations between steer case and no-steer case in the relationship between the positions of the separation vortex shed from the inner lower section of the windward tire and the FWEP vortex.

The FWEP tip vortex could be controlled by adding appendages to the FWEP, or using a flat plate called a foot at the bottom of the FWEP and a semi-conical plate called a cone at the rear-end of the FWEP. It was also possible to adjust the strength and position of the tip vortex by means of the longitudinal vortex produced by the strake, a perpendicular plate positioned in the center of the FW (Fig. 5).

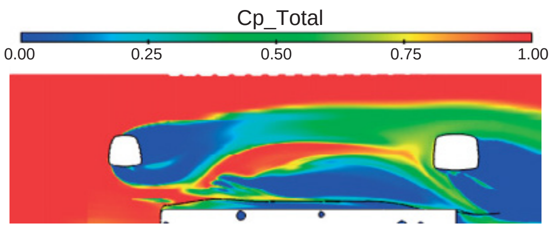


Fig. 7 Tire LE separation wake (CpT, Z=0 mm)

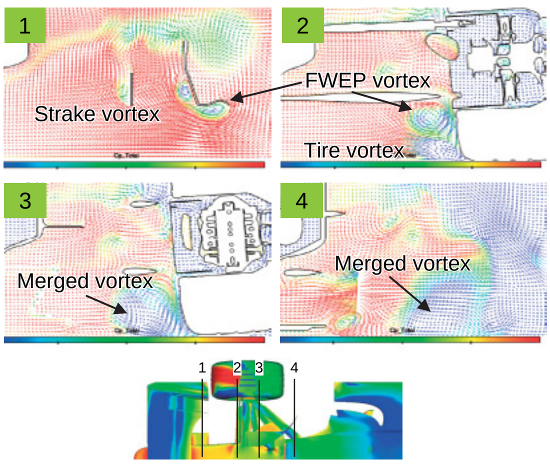


Fig. 8 FWEP and tire vortices (CpT, velocity vector)

In addition to this, a variety of other methods were employed, including raising the mounting position of the outboard sides of the lower front suspension arm in order to help prevent changes in air flow around the front suspension due to changes in the vehicle ride-height from affecting the position of the FWEP vortex.

4. Chassis Upper Devices

The chassis upper refers to the area extending from the nose and the upper section of the monocoque to its sides. While there are virtually no stipulations in the regulations regarding limits on the vehicles in the direction of height, the height of the front rollhoop is the effective maximum height based on considerations of visibility for the driver and safety around the cockpit.

Chassis upper devices mainly contribute to increasing downforce under the floor of the vehicle and from rear components such as the RW by means of the downwash generated by lift.

4.1. Front Suspension Arm

The front suspension arm is the part that plays the greatest role in deflecting the upwash flow from the FW downwards by means of lift (Fig. 9). If no front suspension arm was used, control of the position of the barge-board vortex (see section 5.2.) would cease to function entirely.

However, as the suspension is not allowed to be used as an aero part, there are restrictions on the aspect ratio of the cross-sectional thickness to the chord length of the suspension members and their angle of attack. Because of this, the upper section of the suspension arm can be stalled by the upwash from the FW. The cross section of the suspension arm and the suspension geometry were therefore adjusted, among other measures, in order to maximize the aerodynamic effect.

4.2. Bunny Ears and Fox Ears

Bunny ears that attach to the upper section of the nose (Fig. 10) are another device that creates a downwash. However, the bunny ears have more effect in increasing the angle of attack to the RW than in increasing the suction underneath the vehicle. Consequently, while they increase downforce, the lift-drag ratio of the vehicle, being dependent on that of the RW, does not improve. For this reason, bunny ears can be indicated as a device that is particularly effective on

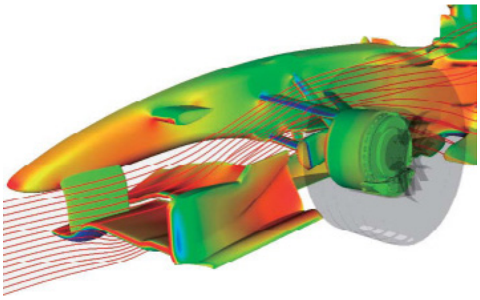


Fig. 9 FW upwash deflection by suspension arm

high-downforce circuits. In addition, because there are few aero parts on the upper section of the chassis, the position of the tip vortex from the bunny ears remains stable to vehicle orientations and disturbances, which makes the bunny ears effective in crosswinds (Fig. 11). The use of bunny ears resulted in an increase of approximately 2% in maximum downforce.

Fox ears (Fig. 12) are fin-shaped devices mounted in front of the mirrors. Like bunny ears, fox ears increase the angle of attack to the RW, in addition to which they contribute to increased downwash to the vehicle floor. This downwash also affects the position of the longitudinal vortex of the barge board. The Formula One regulations state that aero parts in this area can only be positioned above parts fitted to the floor of the vehicle (the shadow rule). This added the limitation that the fox ears must not extend past the projection on the XY plane of the water wings (”WW” below; Fig. 13) directly below them. In addition, while the effect of the fox ears would increase the higher the outer tips were positioned, the potential obstruction of the driver’s field of vision significantly limited the degree of freedom of the shape of the device.

Fig. 10 Bunny ears

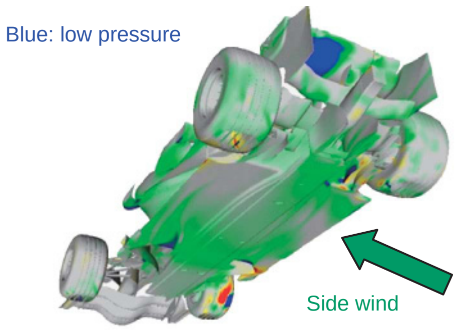


Fig. 11 Effect of bunny ears (delta Cp)

Fig. 12 Fox ear

5. Chassis Lower Devices

The chassis lower area refers to the area from the front axle to the front ends of the side pods (Fig. 13). There are many regulation stipulations regarding minimum heights and maximum widths in this area. The shadow rule also applies across almost the entire area, and there are many regulations concerning the shapes of parts. In addition to the increase in downwash by means of lift, as achieved by chassis upper devices, downwash is also generated by longitudinal vortices. By this means, the mass flow under the floor is increased, and downforce is generated even by devices in front of the floor. Because of this, it is necessary to consider interference between almost all devices from the FW to the diffuser, and the level of contribution of this area to vehicle performance is high, making it important in terms of aerodynamic performance.

5.1. Barge Board Longitudinal Vortex Control

The barge board (Fig. 14) functions to generate a downwash in front of the floor of the vehicle by controlling the longitudinal vortices that it produces. The shape of the barge board varies between teams, but its role in each case is generally identical.

The cambered barge board produces upper and lower tip vortices (Fig. 15). Directly behind the barge board, the flow induced between the vortices is directed outboard of the vehicle. As it moves downstream, the upper vortex moves outboard and downwards by the lower vortex, the vanes, the suspension, the side pods,

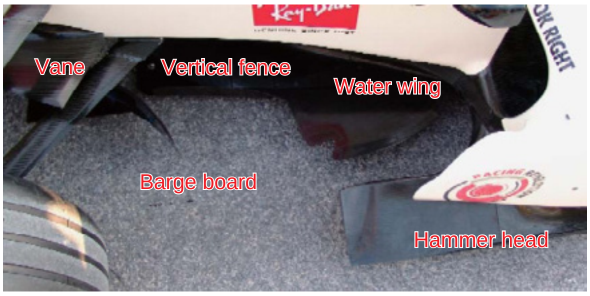


Fig. 13 Devices on chassis lower section

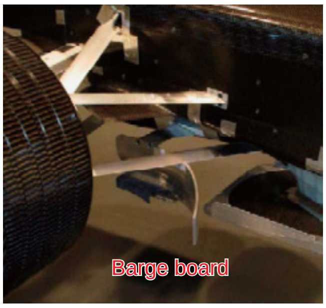


Fig. 14 Barge board

and other parts. The lower vortex proceeds downstream, increasing in intensity as it induces longitudinal vortices at the vertical fence and the WW. Because of this, and also due to ground effects, the lower vortex, shed from the barge board, remains in the same YZ position as was shed from the barge board. In this way, a displacement in the Y direction is produced in the upper and lower vortices, and the flow induced between the vortices is directed downwards. This is a downwash produced by the barge-board longitudinal vortices. This downwash increases the angle of attack towards the underfloor, and increases suction at the leading edge of the floor. In addition, while the effect is small, downforce is also increased by the suction of the lower vortex itself that flows under the floor of the vehicle. However, excess suction at the leading edge of the floor and total pressure loss at the center of the lower vortex can promote growth of the boundary layer on the floor and result in a decrease in diffuser performance. Because of this, the sequence of aero parts to be optimized and setting priorities in optimization are important issues.

The barge board does not produce any other effects as significant as the downwash effect produced by the longitudinal vortices, but it does play another role. Figure 16 shows a comparison of streamlines with and without a barge board. The outward flow created by the barge board pushes the separation wake of the leading edge of the inboard lower section of the front tire outwards, helping to prevent a decrease in diffuser performance. However, very little change is observed in the large separation flow at the back of the tires whether or not a barge board is used. Along with the fact that the separation wake from the leading edges of the tire interacts with the FW tip vortex, as indicated before, these results show how strongly the barge board interacts with front and rear aero parts.

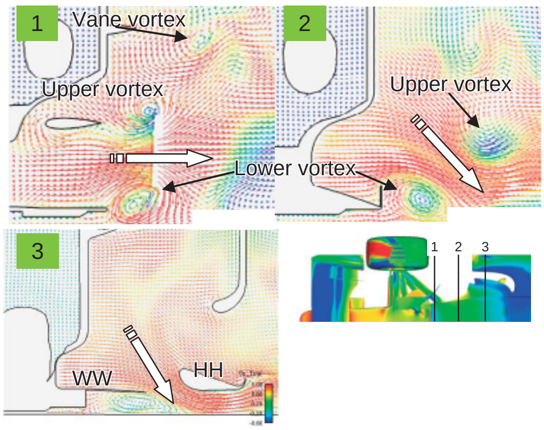


Fig. 15 BBD vortices

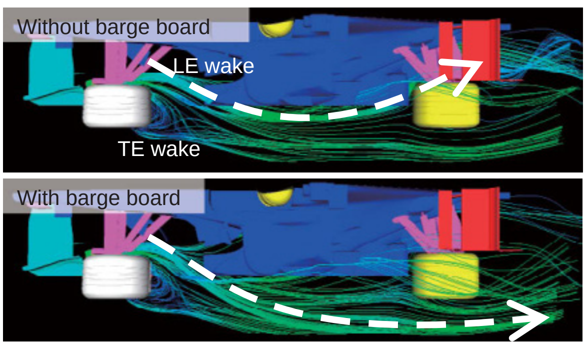


Fig. 16 Tire wake direction

5.2. Development of O-nose Fence

Adding to the effect of the barge board in controlling longitudinal vortices, the O-nose fences (ONF) (Fig. 17) increase downforce by means of the suction they produce.

The ONF generates strong suction below the nose by means of a Venturi effect produced between the left and right fences extending from the nose towards the ground (Fig. 18). In addition, the bottom edges of the fences are close enough to the ground to seal the flow through the gap between the ground and the fences, which increases this effect further. However, if strong suction is generated at the lower section of the nose, an upwash is also generated as a consequent reaction, and this can result in a decrease in underfloor performance. Further, as in the case of the barge board, as a result of the strong suction generated by the fences, the separation vortices from the leading edges of the inboard lower sections of the tires are caught up in the lower vortices of the fences, and this can also result in a decrease in underfloor performance.

To resolve these issues, the downwash was strengthened through the use of a high-lift vane, a strong longitudinal vortex was generated through the use of biplanar ONF, the use of locally high camber profiles on the ONF, and the addition of a shadow

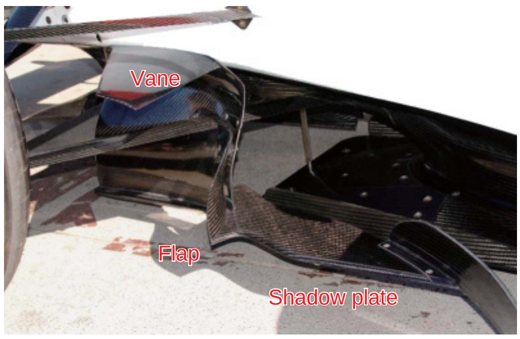


Fig. 17 O-nose fence

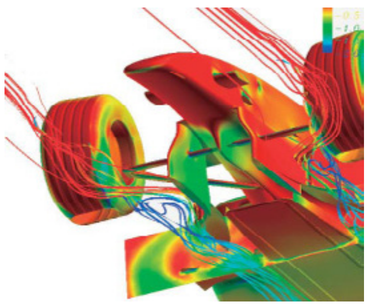


Fig. 18 ONF suction

plate, and the position in which the upper vortex was shed was controlled by adjusting the relationship between the ONF flap height and the position of the vanes. These measures helped to control the generation of upwash. The optimal placement of the cambers also alleviated the drawing-in of the separation vortices from the leading edges of the tires.

The application of these measures helped to enable the ONF to generate the same level of downwash as the barge board with no loss of suction at the lower section of the nose (Fig. 19). As a result, the ONF used at the time increased the lift-drag ratio of the vehicle by 1.5%.

5.3. Effective Use of Downwash of HH and WW

Hammer heads (HH) are parts attached to the vehicle to maximize the use of the downwash of the barge board and other parts. To obtain an effective aspect ratio from the hammer heads, they are extended forwards in the X direction, rather than the Y direction, in which regulations stipulate the maximum vehicle width. The fact that the downwash flows outboard is another reason that the HH, with their leading edges positioned towards inboard, are effective. In addition, gurneys and other parts are attached to the upper section of the trailing edge in order to increase circulation.

The role of the WW, touched on above, will now be discussed in a little more detail. In passing the lower section of the WW, the lower vortex generated by the barge board produces a pressure difference between the upper and lower sections of the WW, which generates the tip vortex from the WW and increases the intensity of the barge board lower vortex. The WW also play another important role. As Fig. 20 shows, the flows at the leading edges of the HH are accelerated and the suction at the HH is increased by narrowing the gap between the WW and the HH. However, if too much downforce is generated, the downforce of the rear aero parts will be reduced as a result of upwash and the growth of underfloor boundary layers.

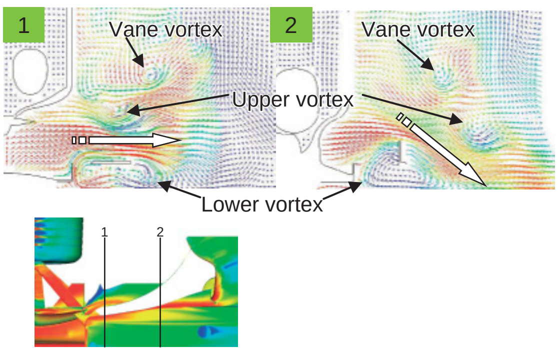


Fig. 19 ONF vortices

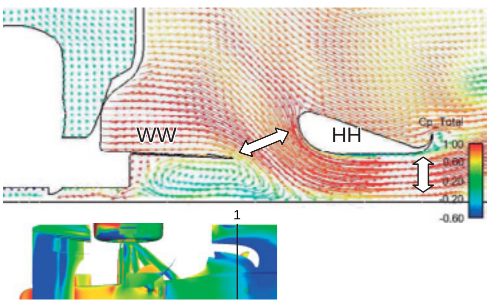


Fig. 20 Channel flow

5.4. Downwash produced by SPLEF

The side-pod leading-edge flickups (SPLEF) are vertical plates with L-shaped YZ sections that are positioned on the exterior of the side pods (Fig. 21). The role of the SPLEF is to intensify the downwash to the HH by providing a blockage and producing circulation. The use of the SPLEF increases the downforce of the HH, but they also generate high levels of induced drag and pressure drag acting on them. However, it was possible to increase the lift-drag ratio through the application of treatments to the leading edges of the vertical plates and the optimization of their shape.

5.5. Ideal Flow

Figure 22 shows the surface streamlines from the nose to the chassis. The figure shows that the use of chassis upper and lower aero parts has resulted in the FW upwash being rapidly directed towards the underfloor – the ideal flow for a Formula One vehicle, as described in section 2.2.

Fig. 21 SPLEF

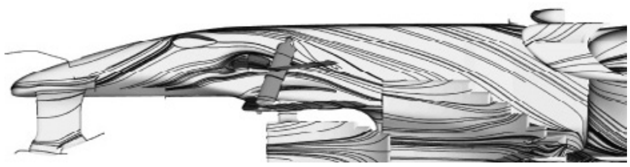


Fig. 22 Surface streamlines

6. Bodywork

The bodywork refers to the areas around the side pods (SP) and the cowl.

6.1. Effect of SP Undercut

From 2000, most teams began using a curved shape called an undercut for the lower sections of the anterior halves of the SP [Fig. 23(b)].

The SP undercuts can be indicated as having three main effects. The first of these is an underfloor seal effect. The circle in Figure 24(b) shows a hypothetical side wall connecting the side edges of the floor and the ground vertically. Without such a side wall, the flows from the leading edges of the SP flow out from under the floor and flow back under the floor and towards the diffuser, but part of the flows also flow out towards the rear tires [Fig. 24(a)]. By contrast, with side walls in place, the underfloor flows are sealed in and flow in straight lines to the diffuser. This accelerates the underfloor flows, boosting suction and increasing downforce [Fig. 24(b)]. However, because the flows towards the rear tires are also blocked, the suction at the leading edges of the SP tends to become weak. In other words, if only the middle and the rear section of the floor was sealed, the maximum benefit of the seal effect could be obtained.

The SP undercuts help to enable this effect to be realized within the scope of the regulations. First, as a result of the undercuts, the influence of the downwash from the front half of the vehicle is extended to the center of the floor (Fig. 25). There, the downwash plays the role of side walls, producing a floor seal effect and increasing underfloor suction (Fig. 26). An increase in the suction at the sides of the SP also accelerates the downwash, further intensifying the seal effect.

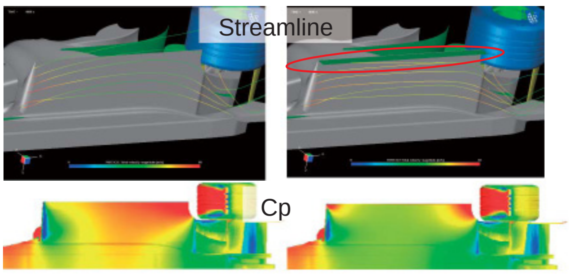
The front of the floor, where no seal effect is necessary, could be used to extend the HH, and it was possible to increase suction on the lower sections of the HH through the use of devices such as gurneys on the trailing edges. This was the second effect of the undercuts.

Depending on the type of barge board employed, the barge-board upper vortex might directly pass the undercuts (Fig. 27). This upper vortex also produces a seal effect, but has a greater effect in optimizing the suction on the lower section of the HH in the direction of the span, by means of the position at which it passes the HH. This is the third effect of the undercuts. However, the effects are not uniform, due to the fact that the intensity and height of the upper vortex differ with different types of barge boards, and its optimum span-wise position to pass the HH also differs. The large barge boards employed by almost all teams generally function to promote the effects of the undercuts.

The use of undercuts rather than linear side pods produced a 1.5% increase in downforce in model-scale wind tunnel tests. The undercuts also helped to increase the underfloor suction in line with predictions. It is also confirmed that the undercut shape was effective even when a barge board was not used.

(a) Flat SP (b) Undercut SP

Fig. 23 Side pod variations



(a) Without side fence (b) With side fence

Fig. 24 Floor seal effect

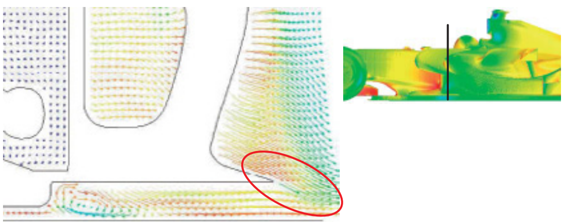


Fig. 25 Downwash flow

6.2. Analysis of Chimney Exhaust Air Flow

The flows that pass the radiators and oil coolers located inside the SP are exhausted through openings in the sides and rear end of the cowl. Cooling performance is adjusted by changing the area of the side openings, but the exhaust flows have a considerable effect on aerodynamic performance of the vehicle.

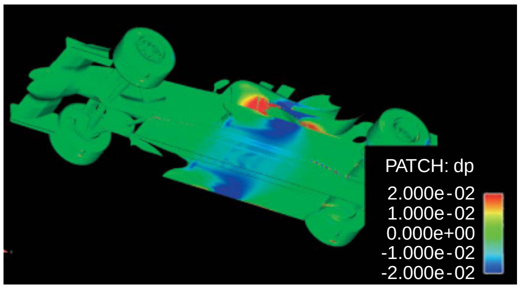


Fig. 26 Undercut SP effect (delta Cp)

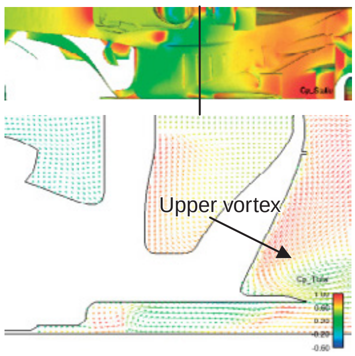
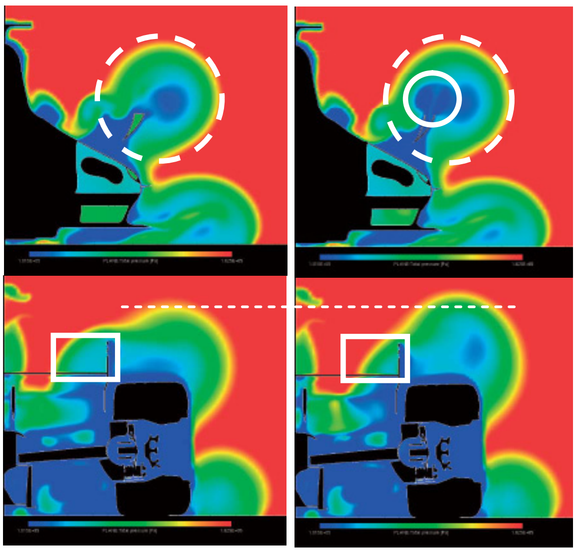


Fig. 27 BBD upper vortex path

From the beginning of the 2000s, chimney-shaped exhaust holes (termed “chimneys”) began to be employed (Fig. 28). Cooling performance was adjusted by means of the openings at the top of the chimneys. Wind tunnel tests using a model showed that fully opening the openings from a fully closed state increased vehicle downforce by 4% and vehicle drag by 5%. The exhaust flow affects the RW load, and half of these increases were due to the change in the load on the RW.

Figure 29 shows the total pressure distribution when the openings of the chimneys are fully closed and when they are fully open. The areas surrounded by the broken circles are wakes from the back of a front tire, and include areas of high-pressure loss (Cpt=0.3). This wake passes above the openings of the chimneys, and part of the wake flows close to the tips of the RW (shown surrounded by the solid squares in the figure). When the chimney openings are fully open, the wake from the front tire is shifted upwards and towards the outboard of the vehicle by the exhaust flow from the chimneys (shown surrounded by the solid circles in the figure). By this means, the area of pressure loss at the RW is reduced in size, and the load on the RW is increased. Because this change in load is due to a change in dynamic pressure, there is almost no change in the lift-drag ratio of the RW.

Fig. 28 Chimney and upper flick-up



(a) Closed (b) Open

Fig. 29 Total pressure distribution in section around chimney trailing edge and RW leading edge

It was also necessary to consider the increase in downforce and induced drag due to the upwash from the chimneys, and the increase in pressure drag due to the increase in the cooling flow. In addition, because the chimneys generated suction on the sides of the SP, like the SP undercuts they produced an underfloor seal effect (Fig. 30).

CFD was employed to study the effect of the blockage, and it was determined that the direction of the wakes from the front tires was affected by the size of the blockage, and the degree of change in the load on the RW due to the opening of the chimneys was affected by the blockage. It was necessary to give sufficient attention to this point in wind tunnel tests.

7. Diffuser

The Formula One regulations strictly determine the dimensions of the diffuser (Fig. 31). Within the scope of these regulations, the suction of the diffuser and the underfloor area was increased by maximizing the effective sectional area and minimizing the static pressure at the exit.

7.1. Maximization of Effective Sectional Area of Diffuser Exit

Separation in the areas in which pressure is recovered, the in-flow from the upper section at the diffuser tips, and the entry of the leading-edge separation vortices of the rear tires can be indicated as factors that reduce the effective sectional area of the diffuser exit [Fig. 32(a)].

Pressure recovery is optimized by the kick-up shape, the section shape, and the fence to produce as uniform

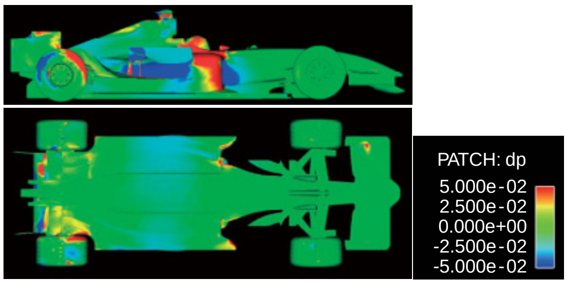


Fig. 30 Chimney effect (delta Cp)

Fig. 31 Diffuser

a pressure distribution as possible in the Y direction without generating separation locally (Fig. 33). The flow from the upper section and the entry of the tire-separation vortices are controlled to a certain extent chiefly by the shape of the foot [Fig. 32(b)].

7.2. Minimization of Static Pressure of Exits

Figure 34 shows the static pressure distribution at the diffuser exit. The static pressure at the diffuser exit is affected by the position of the suspension members and the lower rear wings. The static pressure at the exit can be reduced, and the diffuser flow-rate increased, by adjusting these positions and shapes and optimizing the suction peak location of the upper rear wings.

7.3. Effect of Front Half of Vehicle

The diffuser, positioned at the rear of the vehicle, is affected by the flows from the front half of the vehicle. For example, if the suction at the inlet of the floor is increased, or the lower vortex of the barge board is intensified, boundary-layer thickness will increase under the floor, and the separation toughness of the diffuser will decline. Because of this, the choice of which area aerodynamic development proceeds from, and at what timing, have a significant effect on the aerodynamic characteristics of the vehicle.

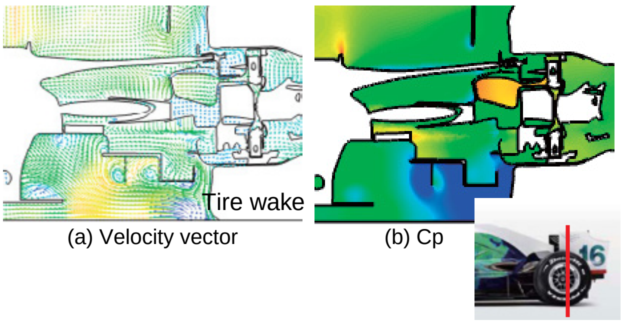


Fig. 32 Flow around diffuser

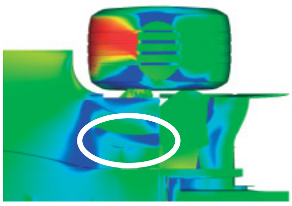


Fig. 33 Diffuser Cp

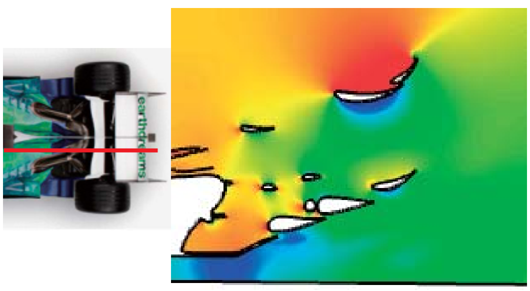


Fig. 34 Diffuser exit static pressure

7.4. Aerodynamic Characteristics produced by Changes in Ride Height

The area ratio of the entrance and exit of the diffuser change with the ride height of the vehicle, and the relative position of the diffuser in relation to the suspension also changes. Because of these changes, the downforce generated also changes.

The downforce characteristic produced by ride height is determined based on a variety of considerations, including the CoP characteristic at low- and high-speed cornering and during braking. In most cases, the CoP during braking is shifted to the rear of the vehicle for the sake of braking stability. This means that aerodynamic development is conducted to produce an increase in the dimensionless rear downforce as the rear ride height increases due to braking (Fig. 35, broken line).

In order to realize this characteristic, vehicle design attempts to ensure that at a low rear ride height there is partial separation from the diffuser and the downforce is reduced. However, caution is necessary, because if the level of separation is too great, hysteresis may prevent the downforce from being recovered when the vehicle is braking (Fig. 35, solid line).

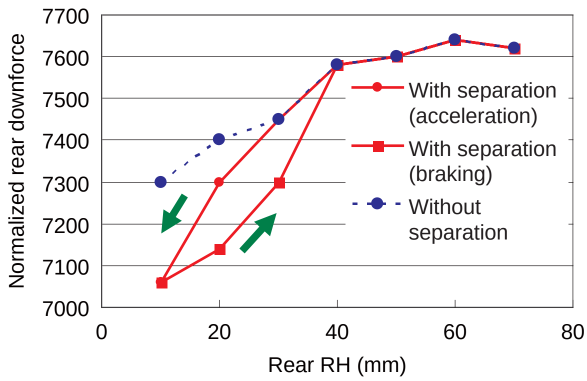


Fig. 35 Ride height aero characteristics

8. Rear Wings

The RW are positioned in an area specified by regulations, behind the rear axle and 300 mm higher in the Z direction than the reference flat section of the vehicle bottom. From 2004, the regulations allowed two elements in the upper section and one element in the lower section of the RW regulation box.

Because the RW have low aspect ratios, an upwash distribution is formed by the tip vortices in the direction of flow (Fig. 36), and massive separation does not occur even when an airfoil with a high camber is used.

Induced drag represents the major component of RW drag, and an increase in drag necessarily follows from an increase in downforce. In addition, the downforce and drag generated by the RW represents a considerable percentage of the downforce and drag for the vehicle as a whole, and is therefore used in the adjustment of the balance between maximum speed and cornering performance. This is to say that it is necessary to use several RW generating different levels of downforce in order to cover the speed characteristics of all circuits. The purpose of RW development is to help increase the lift-drag ratio for the vehicle as a whole by developing, among other considerations, efficient airfoils, load distribution in the Y direction, and RW end plates for each of these various downforce levels.

8.1. RW with Raised Tips

Figure 37 shows the distribution of the angle of attack when the upper RW is not in place, as obtained by CFD. These results show that there is a significant upwash at the tips of the RW.

Figure 38 shows a YZ section of the total pressure distribution close to the tips of the RW. The tips of the RW are exposed to the wake from the front half of the vehicle, and the total pressure declines. Because of this, the load on the tips declines. An upwash is generated close to Y 300 mm, and a downwash close to the wing tips, alleviating this sudden change in load in the Y direction (Fig. 39). This phenomenon can also be seen from the fact that at the tips the pressure distribution in the direction of flow (Fig. 40) shows characteristic of high angles of attack such as a strong peak at the leading edge (Y = 450 mm) though the angle of attack forms an upwash at the tips when no upper RW is placed.

The reduction of the camber of the tips in order to reduce the load could therefore be expected to reduce pressure drag and increase the lift-drag ratio. Other than this, optimization was conducted in the RW development with consideration of changes in induced drag.

8.2. Increasing Robustness by means of Pillar Shape

In most cases, pillars are used to support the centers of RW, and have the function of reducing the weight of the entire structure. However, the air flow undergoes a sudden acceleration between the left and right pillars, necessitating strong pressure recovery. In rearward-

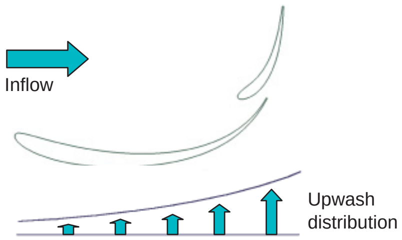


Fig. 36 Upwash by tip vortices

Fig. 37 Inflow angle for RW

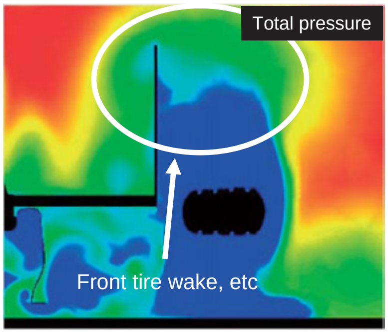


Fig. 38 Cp total around RW

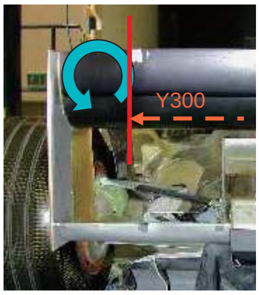


Fig. 39 Secondary vortex due to RW load distribution

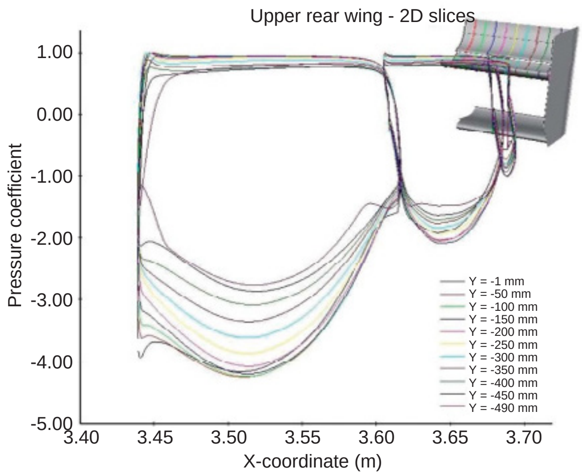


Fig. 40 Sectional Cp distribution

Fig. 41 RW with raised tip

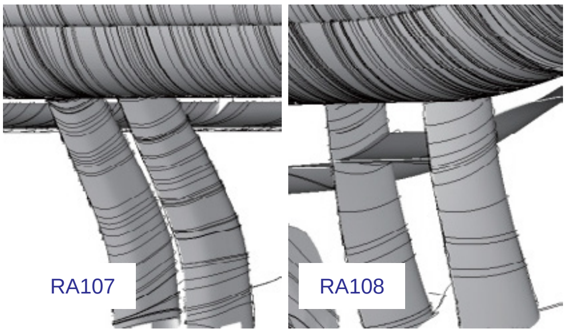


Fig. 42 Boundary layer shear direction on pillars

leaning pillars like those used in the RA 107 (Fig. 42), the spanwise suction distribution produced by the rear inclination directs the flow in the boundary-layer upwards, extending the distance at which boundary layers grows; at the same time, a crossflow occurs between the boundary layer and the outer flow. As the boundary-layer of the pillars merges with that of RW, these are thought to be the factors behind a decline in the separation stability of the RW.

The following changes were made in order to resolve the instability produced by the pillars in the wings developed for the RA 108. First, the space between the pillars was increased to the regulation limit and the sectional shape of the pillars was modified in order to ease pressure recovery. The pillars were also moved to the leading edges of the wings in order to separate the pressure recovery areas of the wings and the pillars. In addition, small wings were positioned between the pillars in order to ease pressure recovery on the RW. Finally, the rearward inclination of the pillars was minimized in order to control the upward inclination of the pillar boundary layer flow.

9. Afterword

The air flows around multiple bodies such as the chassis of Formula One vehicles display a strong non-linearity, and it is not possible in practice to understand the detailed mechanisms of all the aerodynamic phenomena involved. However, aerodynamics developments were conducted efficiently by the use of CFD for qualitative analysis of the core aerodynamic phenomena, backed up by quantitative data obtained in wind tunnel tests. These methods also enabled the accuracy of predicting the aerodynamic performance of the vehicle when it is actually running on a race track to be increased.

10. Acknowledgments

The authors would like to take this opportunity to offer their sincere thanks to all the staff members of the former BAR and HRF1 for their many years of shared developments in aerodynamics, and to the other members of the Honda team and staff members of other companies who generously assisted in a wide variety of ways in development projects.

Reference

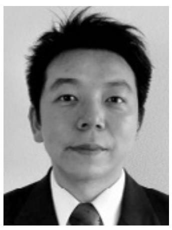
(1) Ogawa, A., Yano, S., Mashio, S., Takiguchi, T., Nakamura, S., Shingai, M.: Development Methodologies for Formula One Aerodynamics, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 142-151

■ Author ■

Atsushi OGAWA

Susumu MASHIO

Daisuke NAKAMURA



Yasutaka MASUMITSU

Masayuki MINAGAWA

Yusuke NAKAI

Technologies for Enhancement of Dynamic Performance of Formula One Vehicle

Hideaki SHIBUE* Kazuhiro TANEDA* Yasutaka KITAKI*

ABSTRACT

No matter how superb the performance of the powerplant, unless that power can be transmitted efficiently and effectively through the tires to the track surface, a race vehicle will not have a competitive edge. The most recent technologies for the enhancement of the dynamic performance of Formula One vehicles are developed with a focus on maximizing tire performance. The tires used in Formula One are designed for good performance only within an extremely narrow range of conditions in terms of parameters such as tire contact state and tire temperature, in order to enhance the performance of the tires to the limit. Therefore, understanding and controlling these conditions is an important issue in the development of enhancement technologies for the dynamic performance of Formula One vehicles. This paper will discuss suspension design and vehicle setup, two factors that affect tire performance, and the development of a tire model to function as their theoretical basis.

1. Introduction

Development programs to enhance the dynamic performance of Formula One vehicles are conducted in two stages. The first is the design stage for vehicle performance. At this stage, the vehicle concept and targets for the various elements of dynamic performance are established based on performance analyses of Honda vehicles and other teams' vehicles, and vehicle components enabling these targets to be realized are designed.

The second stage is vehicle setup on an actual circuit. The dynamic performance of Formula One vehicles varies with the layout of the circuit, track conditions, and atmospheric conditions, and a vehicle will therefore not always display identical performance. Accordingly, at this stage, element parts with varying specifications and mechanisms that enable vehicle characteristics to be varied are prepared, and the vehicle's dynamic performance in actual circuit driving is optimized by means of their application and coordination.

While vehicle design determines the potential dynamic performance of the vehicle, setup on the circuit enables the vehicle to reach that potential in response to circuit conditions and race strategy. The use of these two stages enables the vehicle to display a high level of dynamic performance in a race.

The paper will discuss the concepts employed in suspension design and setup, factors that affect tire performance, and will consider the development of a tire model to serve as their theoretical basis.

* Automobile R&D Center

2. Suspension Design

The suspension plays an important role in maximizing the performance potential of the tires. In Honda's third Formula One era, suspension development was conducted making maximal use of techniques fostered in the development of mass production vehicles. In particular, specifications were set for the scrub radius, caster trails, and other aspects of king pin geometry using identical concepts to those employed in mass production vehicles, and their usefulness was verified.

Elements specific to Formula One are a tire characteristic in which the tires perform well under an extremely narrow range of conditions, and an aerodynamic characteristic in which vehicle behavior that is determined by the suspension plays an important role. Based on these considerations, development was conducted as follows:

(1) Optimization of initial camber, camber gain, and camber change with steering angle in accordance with tire characteristics
(2) Development of tire air temperature control technique for stabilization of tire pressure
(3) Development of load transfer control for adjustment of mechanical balance, in order to enhance tire warm-up performance
(4) Development of geometry enabling maximization of aerodynamic performance

As one example aspect of suspension development, this paper will discuss the development of a front pushrod on upright (FPROU) suspension that is designed to maximize tire performance by means of control of wheel load for all four wheels.

2.1. Overview of FPROU

The Formula One steering characteristic basically uses understeer (US) to increase stability during high-speed cornering and oversteer (OS) to enhance turn-in performance during low-speed cornering. Suspension design exploited the nonlinearity of the angles of rotation of the rocker arms in relation to the vertical motion of the pushrods to induce a high level of nonlinearity in the wheel rate of the front wheels (a rising rate). This increased front roll stiffness during high-speed cornering, when a strong downforce is acting, thus enabling use of US.

During Honda's third Formula One era, the company was successful in enhancing turnability during low-speed cornering by mounting the front pushrods on the uprights rather than the lower wishbones in order to produce a characteristic that varied the mechanical balance (the front-rear allocation of lateral load transfer during turning) in relation to the steering angle (Fig. 1).

2.2. Load Transfer Mechanism during Turning

When the vehicle is statically steered, the front inner wheel will normally be subjected to a downward load, due chiefly to the effect of the angle of the casters. Because the motion in the front roll direction that this induces is restricted by the rear suspension, the wheel loads for the four wheels are transferred diagonally. It is known that because the load transfer produced by this geometric motion is added to the load transfer produced by the front-rear allocation of roll stiffness and the anti-force geometry when the vehicle is turning, load transfer can be controlled by controlling the trajectory of contact patch lift when the steering wheel is turned. In conventional suspensions, increasing the caster angle will reduce the load transfer at the front of the vehicle.

This section will focus on the inner turning wheels in order to consider the changes produced when the pushrods are mounted on the uprights. Because there is virtually no change in damper stroke when the steering wheel is turned, pivot N of the pushrod on the rocker side (Fig. 1) can be considered to be fixed. This means that pivot M of the pushrod on the upright side is confined on the sphere surface with pivot N at its center. When pivot M is positioned to the rear of the kingpin

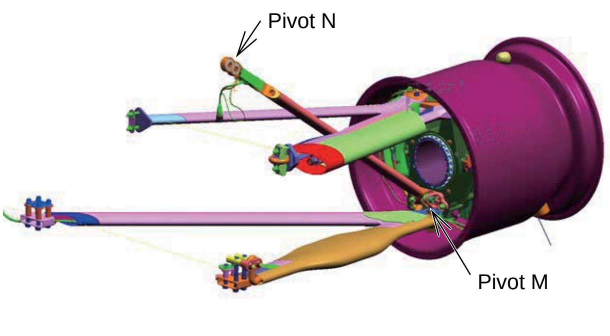


Fig. 1 Front pushrod on upright suspension

(Fig. 2), pivot M rotates around the axis of the kingpin and shifts towards the center of the vehicle when the steering wheel is turned. Because the pushrod is at an anhedral angle, it is necessary for the upright to shift in a downward direction. In the same way, if the outside wheel is considered, the upright shifts in an upward direction.

Figure 3 shows the amount of contact patch lift against toe angle. When the M pivot is offset to the rear, the same effect can be obtained as when the caster angle is increased. As a result, there is a comparative reduction in front load transfer when the vehicle is turning. By contrast, when the M pivot is offset to the front, the same effect can be obtained as when the caster angle is reduced.

The exploitation of the mechanism described above enabled control of the mechanical balance that is dependent on toe angle without restriction by geometric considerations such as the position of the kingpin axis, and played an important role in helping to inhibit US during low-speed cornering.

In addition, when the M pivot was offset in the left-right direction (y) using the same mechanism, the upright rose together with the inner and outer turning wheels when the steering wheel was turned, lowering the height of the vehicle (Fig. 4). This effect was particularly marked in the large steering angle range.

2.3. Dynamic Performance Simulation

Figure 5 shows the results of prediction of changes in mechanical balance in a vehicle motion simulation using ADAMS. When a positive offset is introduced, i.e.,

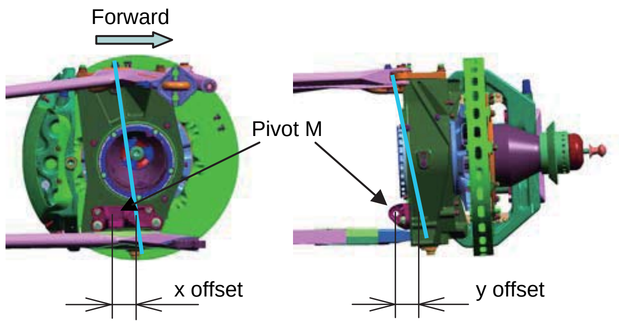


Fig. 2 Definition of pushrod offset

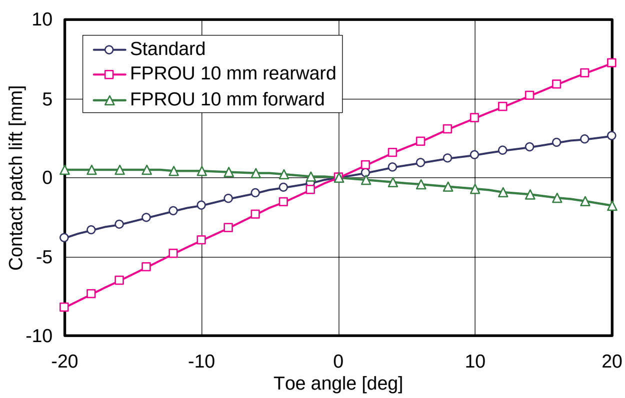


Fig. 3 Change in lift against toe angle

when the M pivot is offset to the rear of the kingpin axis, the mechanical balance shifts to the rear, and the steering characteristic tends towards OS. The degree of change in the mechanical balance increases in proportion to the level of offset.

2.4. Track Tests and Application in Races

Track tests were conducted on the suspension system designed and manufactured on the basis of the simulations in September 2005 on the Jerez circuit. The tests demonstrated the superiority of the new suspension system, as it matched the targets set for it in helping to inhibit US during low-speed cornering and consistently bettering lap times against the base vehicle. Based on these results, the suspension system was employed in races from the Japan Grand Prix in 2005 onwards.

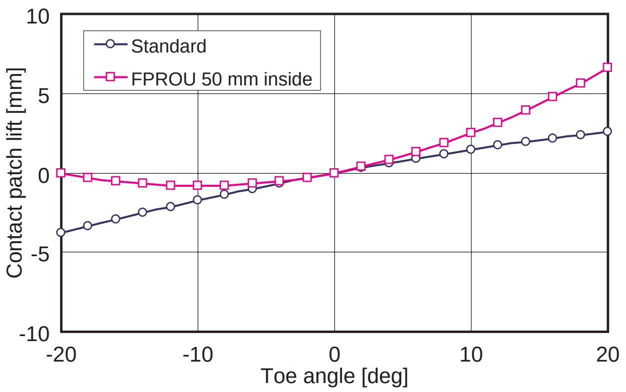


Fig. 4 Change in lift against toe angle (y-direction offset)

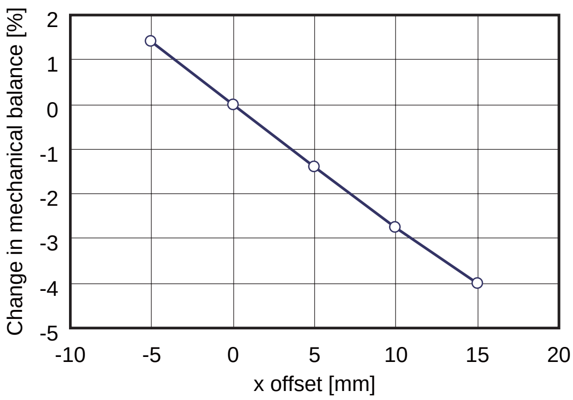


Fig. 5 Change in mechanical balance with x-direction offset

3. Concept of Vehicle Setup

This chapter will focus on vehicle setup, and will consider the concept for the setup of the 2006 RA106 vehicle, which became the basis for later vehicle setup.

Having taken second place in the 2004 World Constructors' Championship, Honda commenced the 2005 race season determined to win the title, but the 2005 vehicle displayed low braking stability. The drivers demanded increased stability during braking, and the hydraulic control was modified to greatly inhibit the differential motion of the differential gears (ultimately, spools were fitted to mechanically connect the left and right wheels). Due to this increased US during turn-in, the mechanical balance was setup to shift towards the rear wheels in order to enhance turn-in performance. However, this increased the load on the outer rear wheels during cornering, generating sudden snap OS resulting in the rear wheels slipping when the vehicle accelerated coming out of a corner. This made the vehicle's cornering traction performance fall behind that of other vehicles (Fig. 6).

In addition, the increased load on the rear tires brought about a degradation of the tire compound, and stability and traction performance declined with each lap. In addition, the setup produced a vicious circle in which a decline in cornering speed also reduced the temperature of the front tires, further increasing US and thus resulting in a decline in dynamic performance.

3.1. Method of Formulation of Setup for RA106

The main issue with regard to the setup of Honda's 2005 Formula One vehicle was that engineers had become process-focused and shortsighted in their quest for local optimal solutions, resulting in the inability to exploit the potential for the dynamic performance that should be expected from the vehicle.

Accordingly, the formulation of a concept to enable vehicle setup to be conducted in a strategic fashion was focused on in the development of the RA106. This meant a rigorous process in which the performance elements that should be enhanced in setup were clarified, methods of achieving these targets were quantitatively analyzed, effects and levels of sensitivity were predicted, taking in pros and cons, and these predictions were verified in track tests. The process aimed, by establishing the orientation for the setup on the desk and using track tests for verification, to avoid falling into the trap of localized optimization as had been the case when the setup was successively altered on the basis of the results of individual tests.

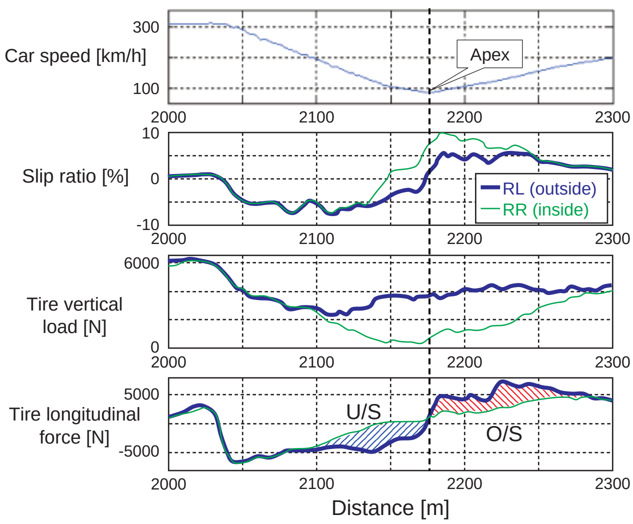


Fig. 6 Cornering conditions in 2005

In addition, setting milestones in the formulation of the setup and designing and implementing track test plans on this basis helped to enable a reliable verification process.

3.2. Concept of Setup of RA106

The dynamic performance of Honda's 2005 Formula One vehicle was analyzed, and the following performance elements were selected as targets for the RA106, based on considerations of maximizing tire performance by optimizing the total setup of the vehicle:

(1) Enhancement of cornering traction performance
(2) Reduction of degradation of rear tires
(3) Achievement of increased grip force by increasing temperature of front tires

3.3. Methods of Achieving Targets

A simulation using a tire model was employed in an analysis to enable engineers to determine how to achieve the targets listed above using the following elements of vehicle setup, which determine dynamic performance:

- Front axle weight distribution (W/D)
- Front axle downforce distribution (CoP)
- Front axle mechanical balance (M/B)
- Lateral distribution of rear braking and driving force (Diff)

Figure 7 shows the results of an analysis of changes in traction performance when cornering in each speed range. The forward shift of M/B has the greatest effect in enhancing traction performance during low-speed cornering, with W/D having the next greatest effect. The CoP makes a high contribution in the high-speed range. Figure 8 shows the results of an analysis of changes in traction performance when each of the elements of the setup is varied. In the case of M/B, traction performance increases virtually linearly up to a shift of approximately 25% forward. Because increased traction performance from low speeds upwards would make a significant contribution to enhanced lap times, forward M/B was made the main element in the achievement of enhanced traction performance.

With forward M/B, the overall steering balance would become US. Figure 9 shows estimates of the amount of adjustment of W/D and CoP necessary to achieving this balance. Despite the fact that sensitivity differs with cornering speed, each parameter displayed a largely linear response in relation to the degree of change in M/B. Figure 10 shows the results of an analysis of the effect of the lateral distribution of longitudinal forces on the rear wheels produced by the differential gear on the steering balance. During low-speed cornering, the US generated by forward M/B could be compensated by differential control, but this did not have sufficient effect during high-speed cornering, and it would therefore be necessary to compensate for US in combination with another element of the setup. It was judged that the combination of forward M/B and rearward W/D would contribute to enhanced traction performance, and this was therefore selected as the best strategy.

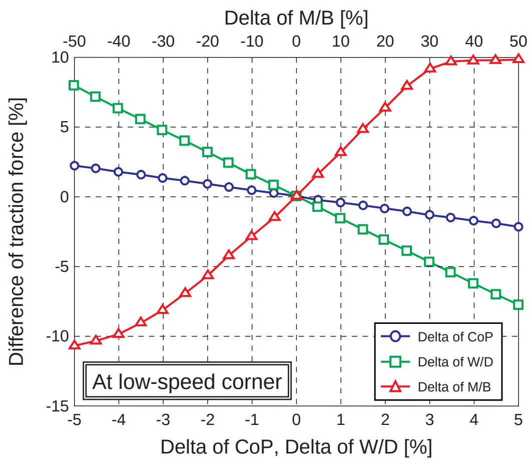


Fig. 8　Traction change with setup sweep

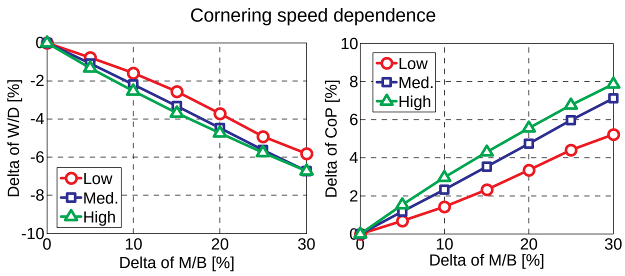


Fig. 9　Relationships in setups for achievement of equivalent steer balance

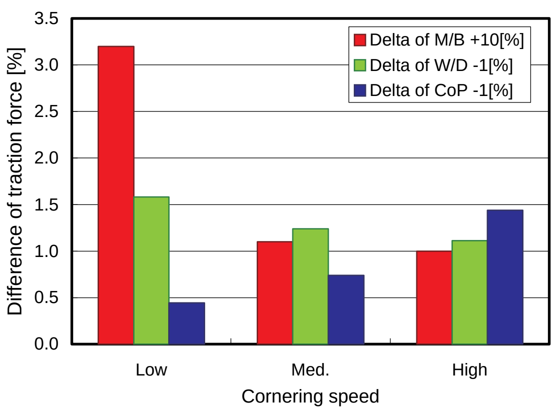


Fig. 7　Traction enhancement with setup changes

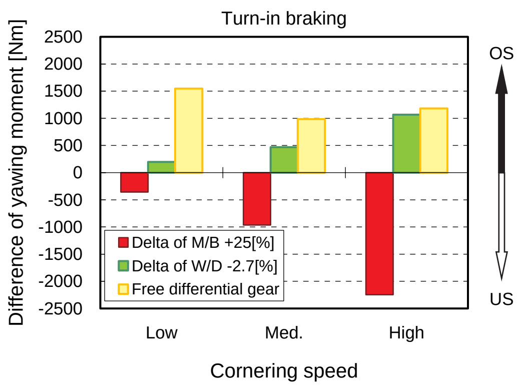


Fig. 10　Effect of differential gear on steer balance

Figure 11 shows the results of an analysis of the change in tire surface temperature during cornering when the M/B was varied. It was estimated that forward M/B could cause the temperature of the front outside tire, which is dominant during cornering, to increase, and that the tire's grip force would increase as its temperature approached the necessary temperature range. In addition, because the temperature of the rear outside tire would decrease, this method could also be expected to control the degradation of the tire compounds due to excessive temperatures.

Based on the results of the analyses discussed above, the proposed setup for the RA106 was formulated as follows:

(1) Forward mechanical balance
(2) Rearward weight distribution
(3) Differential gear control during cornering

3.4. Verification of Effects

The effects of the RA106 setup that had been formulated on the desk were verified in track tests conducted on a variety of courses from December 2005 to February 2006.

Figure 12 shows a comparison of traction force during low-speed cornering in track tests conducted on the Jerez circuit. For this test, the M/B of the RA106 was increased by 6.4% and its W/D reduced by 1.5% against the 2005 vehicle in the setup. As a result, the average traction force of the RA106, averaged over 10 laps, increased by approximately 10%. Figure 13 shows a comparison of section time for cornering traction during low-speed cornering, with the RA106 setup reducing the vehicle's average time per corner by 0.05 sec. As a result, the lap time of the RA106 was reduced by an average of 0.74 sec per lap (Fig. 14). These results indicated that the cornering traction performance of the RA106 had been enhanced.

Figure 15 shows the results of an analysis of tire degradation. Data for which track conditions and atmospheric conditions could be regarded as identical was isolated from data collected in multiple track tests of the 2005 vehicle and the RA106 conducted for the same periods of time. Lap times for each lap were averaged from this data. These average lap times were taken to be representative values for change in lap time, and the degree of degradation in lap times was compared for the vehicles. Taking into consideration the difference in the weight of the vehicles, a simulation was used to calculate the sensitivity of lap time to vehicle weight,

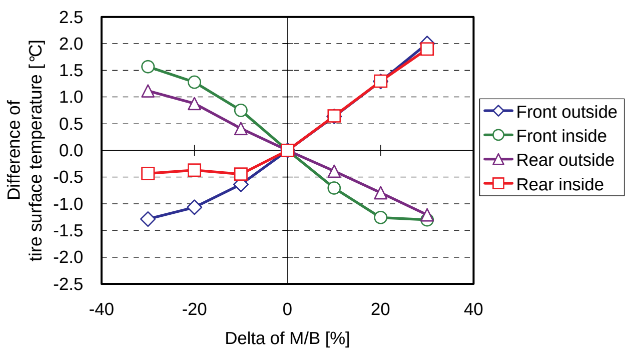


Fig. 11 Estimation of change in tire surface temperature

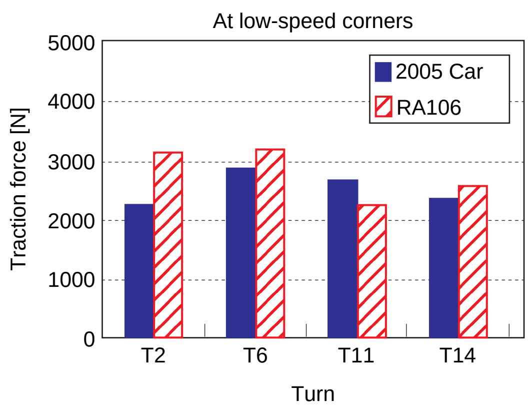


Fig. 12 Comparison of traction force

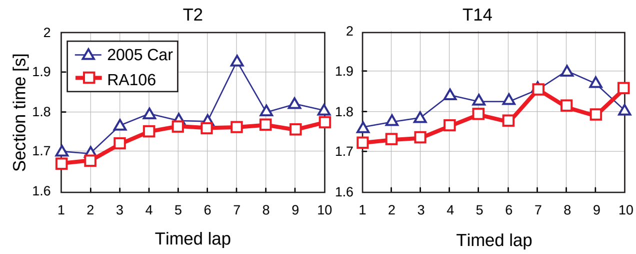


Fig. 13 Section time during corner acceleration

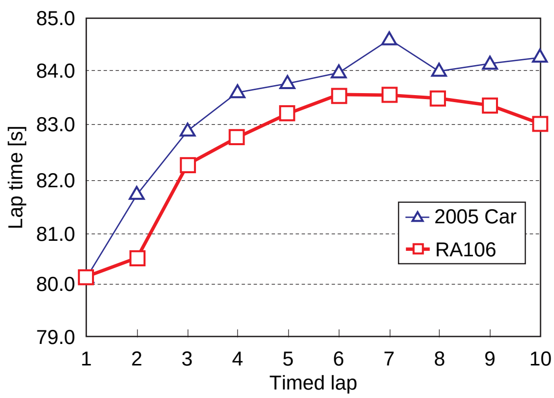


Fig. 14 Comparison of lap time

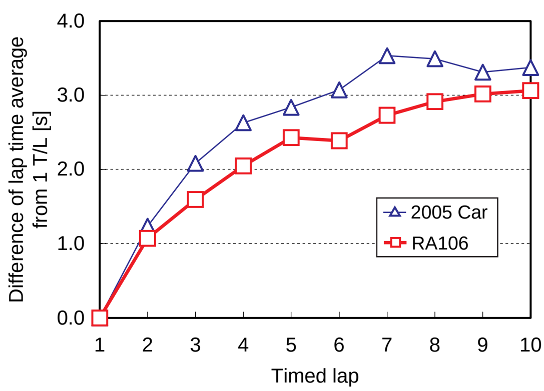


Fig. 15 Analysis of tire degradation from perspective of lap time

and the lap times were converted to represent vehicles of the same weight. The results of the comparison showed that the level of deterioration in lap time was reduced by an average of approximately 0.48 sec for the RA106, indicating that the degradation of the tires had been controlled.

Figure 16 shows the results of a comparison of front tire temperature conducted in the same way, using data from track tests with largely identical track and atmospheric conditions. The tests were conducted on the Jerez circuit, which features a large number of right turns. The temperature of the left front tire, which more frequently worked as the outer tire, increased by an average of approximately 16°C, while the temperature of the right front tire increased by approximately 5°C.

The results of these analyses confirmed that the RA106 setup had performed as speculated, increasing cornering traction performance, controlling tire degradation, and increasing grip force by raising the temperature of the front tires.

3.5. Summary of Concept of Vehicle Setup

The elements of the RA106 setup which would lead to performance enhancements were determined through analysis of the 2005 vehicle, and a simulation employing a tire model was used to study a setup that would actualize these elements. The effects of the setup were verified in track tests, confirming that the setup accorded with the goals established for it. Following this verification of the effects of the RA106 setup in tests conducted during winter, the setup was used as the base for all vehicle setups from 2006, and contributed to Honda's victory in the 2006 Hungary Grand Prix.

While vehicle setup had previously relied to a great extent on comments from drivers and the experience of the engineers involved, these results demonstrated that a logical approach, based on a tire model, was effective. The necessity for the development of a more sophisticated tire model capable of predicting performance under all conditions therefore increased.

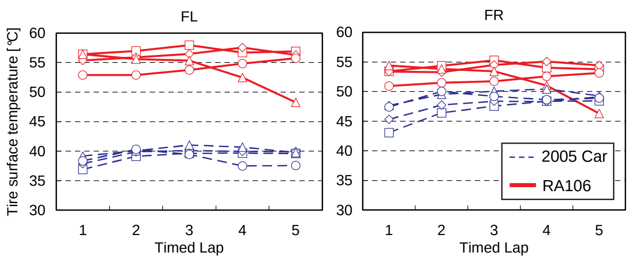


Fig. 16 Front tire surface temperature

4. Development of Tire Model with Coupled Calculation of Force and Temperature

Tire performance has a considerable effect on the dynamic performance of Formula One vehicles. The achievement of enhanced dynamic performance necessitates a quantitative understanding of tire performance, and vehicle design and setup that maximize tire performance under all conditions. Regulations stipulate that the setup be performed by Saturday morning in race week, and it is therefore necessary to perform the setup with consideration of factors including changes in environmental conditions during the qualifying sessions on Saturday afternoon and the finals on Sunday. This necessitates a model capable of calculating at a sufficient rate of accuracy the force generated by the tires under a variety of driving conditions, including load, alignment, and track surface temperature. The performance of Formula One vehicle tires is in particular affected by the surface temperature and the internal temperature of the tires, and this necessitates the use of a model that considers the effect of temperature, and is able to perform coupled calculations of force and temperature in response to changes in tire input due to environmental temperature and driving conditions.

The ability to simultaneously analyze tire force and temperature would represent a significant advantage not only to considerations of vehicle setup and tire performance, but also to considerations including race strategy, such as warm-up performance when a safety car is on the track, and degradation due to heat.

The Magic Formula is one well-known tire model, but this model does not include a temperature element, making it unable to calculate the effect of temperature changes resulting from driving conditions on tire force. Tire suppliers also provide tire models, but their insides are concealed, in practice inhibiting any increase in the accuracy of the models or modification for use by the race team.

It was judged that in order to excel over other teams in the area of vehicle dynamic performance, it would be necessary to possess, as an in-house developed technology, a tire model able to perform coupled calculations of force and temperature, and to use this tire model as the basis for a technology enabling prediction and analysis of dynamic performance.

4.1. Structure of the Model

4.1.1. Model concept

Considering the use of the model in vehicle setup at the circuit in addition to vehicle design, the development of a model with a good balance between calculation accuracy and speed was established as a target. The elements with the greatest effect on the accuracy of calculations of tire forces were isolated and modeled to enable this target to be met.

Tire forces are chiefly determined by the structural deformation of the tire and the friction characteristic between the road surface and the tire. The friction coefficient of the tires is changed significantly by environmental conditions and driving conditions, including the road surface roughness, dust on the road surface, tire surface temperature, tire slip speed, rubber wear, and thermal degradation. Accordingly, characteristics such as structural deformation and heat transfer, which could be modeled on the basis of theoretical concepts and the results of bench tests, were separated in the model's structure from elements such as the friction coefficient, which are dependent on actual driving conditions, with the parameters for the latter being identified from actual vehicle data. This helped to enable the creation of a more realistic and more accurate model.

4.1.2. Force model

The model treated deformation of the tire contact patch as divided into a belt section and a tread rubber section.

Belt deformation was approximated by expression as a quadratic function in relation to the position of the tire contact patch in the longitudinal direction. The deformation obtained in this manner was corrected using the tire side force, self-aligning torque, internal pressure, and longitudinal force.

With regard to the deformation of the tread rubber section, the adhesive contact area (the area of elastic deformation) of the contact patch was calculated from the relationship between the maximum deformation of the tread rubber, as determined by the elastic modulus, the static friction coefficient, and the contact surface pressure, as well as the necessary deformation, as determined by the tire slip angle, slip ratio, and degree of belt deformation. The rest of the contact patch was considered a sliding contact area (Fig. 17).

The forces on the tire are determined in the adhesive contact area by the elastic modulus of the rubber and the degree of deformation, and in the sliding contact area by the slip friction coefficient and load. At this stage, a model was formulated in which the elastic modulus was defined as bulk temperature functions (bulk temperature will be discussed below), the static friction coefficient as surface temperature functions and contact surface pressure functions, which is to be corrected based on its distribution in the direction of the tread. The slip friction coefficient was further defined as tire slip speed functions in the model.

Using this method, the tire force in the adhesive and sliding contact areas were calculated, and their synthesis enabled calculation of the final tire force.

4.1.3. Thermal (temperature) model

The cross-sectional structure of the tire was formularized as a one dimensional node model divided into three layers, a tread surface layer, tread bulk rubber layer, and carcass/tread belt rubber layer, with each layer at a single temperature.

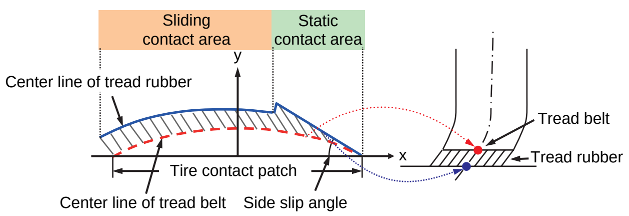


Fig. 17 Deformation model for tire contact patch

In addition, the work performed by the tire was input as heat chiefly in the tread surface layer for the sliding contact area and in the tread bulk rubber layer for the adhesive contact area. The rolling resistance of the tire was treated as heat generated in the tread bulk rubber layer.

Heat transfer between the three layers, convection heat transfer to the air, and heat conduction to the road surface at the contact area are all calculated to enable calculation of the temperature of each layer. Heat transfer by radiation was considered to have represented 2% or less of the total figure in comparison to heat convection and conduction, and was therefore omitted from the model (Fig. 18).

4.2. Identification of Parameters

4.2.1. Method employed

The difference between the calculation results for force and moment from the model, obtained using time series data from track tests for parameters including wheel load, slip angle, slip ratio, and camber angle, and the target data were treated as objective functions, and parameters were identified using an optimization method to minimize this difference. Parameters that changed with the degradation of the tires, such as maximum friction coefficient, were identified for each lap, while parameters that did not change, such as the temperature characteristic, were identified as a single value for all the track test data. Measurement results from sensors and the results of calculations using a combination of sensor results and bench test results or theoretical values (in the case of parameters including wheel load and slip angle) were used as the data necessary for parameter identification.

The following two methods of identification were employed, depending on the target.

(1) Method using the results of 6-component wheel force measurements

The results of 6-force measurements (longitudinal force, side force, self-aligning torque) were used as targets. Because the accuracy of identification was increased and the level of variation reduced the greater the diversity of combinations of parameters such as wheel load and alignment, data for both the left and right tires were employed in the parameter identification.

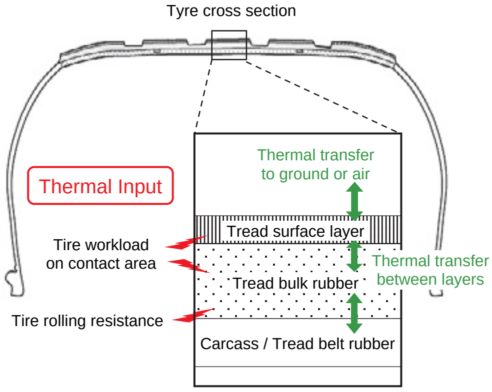


Fig. 18 Tire thermal model

(2) Method using vehicle's dynamic states

Vehicle's dynamic states (longitudinal acceleration, lateral acceleration, and yaw moment) measured using sensors were employed as targets. The vehicle's dynamic states were calculated using a variety of data including calculation results from the tire model, vehicle specifications, and figures for air resistance and tire rolling resistance, and parameters were identified simultaneously for the front and rear tires that enabled these dynamic states to be matched with the target dynamic states.

4.2.2. Optimization method

Broadly speaking, two types of optimization methods are available. Gradient methods use the gradient of the objective function (the error in relation to the target) against the design variables (the optimization parameters) to enable optimization. In stochastic methods, design variables are varied at a specific probability to produce a variety of figures that are potential optimum solutions, and the optimum solution is selected from among this group. The project discussed here used an evolution strategy, which is a type of stochastic method.

4.3. Results of Model Verification

Because results are affected by the measurement accuracy of the data used in the model calculations and the target data for comparison, it can be challenging in the verification of a model that uses actual vehicle data to determine whether a specific issue originates in the model or in the data. In addition, because the actual values of the tire parameters are unknown, quantitative verification of the results of parameter identification also represents a challenge.

Artificial target data was therefore created by constructing a vehicle model from body, suspension, steering, and aerodynamics models and the tire model, conducting a one-lap circuit simulation, and by adding noise corresponding to actual driving conditions to the obtained vehicle dynamics data. This artificial data was then used to verify the model. This method enabled issues of measurement accuracy to be set aside, and because the tire parameters that should be obtained as a result of parameter identification were already known, it was possible to verify the tire model and its parameter identification section.

Figure 19 shows a time series comparison of the average error in vehicle's dynamic states (longitudinal force, lateral force, yaw moment) for one lap in the target data and the model, using a specific identification calculation. Because the evolution strategy is a type of stochastic method, the identification calculations have been conducted 10 times for the purposes of comparison. Error for all the results is approximately 1.2%, and as the time series graph shows, sufficient accuracy was obtained for the analysis of vehicle dynamics. However, despite the fact that the figure for error in vehicle's dynamic states was largely identical in both sets of results, a phenomenon could be observed in which synchronized variations in multiple parameters were observed. The combination of different parameters to produce an identical output from the model is termed a modal characteristic, and is a phenomenon that is often observed in multi-parameter models. Increasing the diversity of the target data is an effective method of controlling the modal characteristic in analysis results, and this study provided insights into the degree of diversity necessary.

Based on the verification of the accuracy of the model described above, results for tire force and temperature were compared with results from track tests.

Figure 20 shows a comparison of measurements using a tire force meter and calculation results from the tire model for the fifth lap of five laps around the Barcelona circuit. The parameters for the tire model were identified using data from the third lap. Side force is calculated accurately across the entire range. Calculation results for the peak value of longitudinal force during braking are somewhat low, but results for turn-in correspond well. The slip ratio, one of the input parameters for the model, changes rapidly when the vehicle is braking, and it is therefore not easy to be confident of its accuracy. The effect of error due to input data accuracy was therefore considered to be greater in producing this result than the effect of the accuracy of the model.

Figure 21 shows a comparison of measurements and model calculation results for the temperature of the tire surface and bulk layers. During the period shown, the vehicle makes a pit stop (t=220 sec), and then reenters the track (t=280 sec). The model was able to reproduce the drop in the internal temperature of the tire due to the cessation of internal heat generation and the increase in the temperature of the tire surface due to the decline in heat dissipation into the air after stopping, in addition

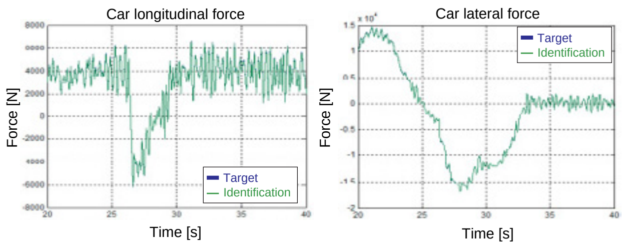


Fig. 19 Identification result (car dynamics error)

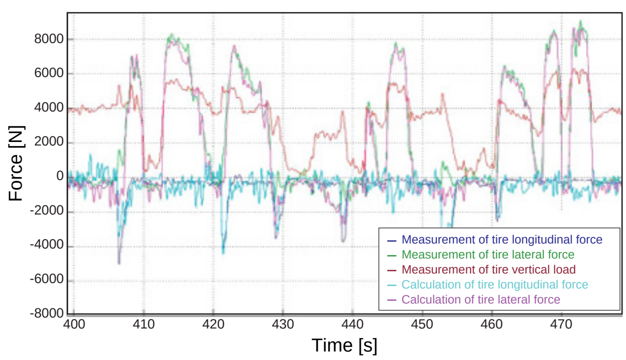


Fig. 20 Comparison of tire forces (front)

to the gradual increase in temperature until a state of thermal equilibrium is reached after the vehicle starts moving again. The model calculated the internal temperature of the tire at a somewhat high level during lap driving, but the level of accuracy can be considered sufficient for evaluating warm-up performance.

4.4. Summary of Tire Model Development

The project discussed in this paper developed a tire model that is able to perform coupled calculations of tire force and tire temperature that affects tire force. The use of a model configuration, in which parameter identification was performed for elements affected by driving conditions based on track data, enabled the realization of a level of calculation accuracy and speed satisfying a broad range of demands, from vehicle design to vehicle setup.

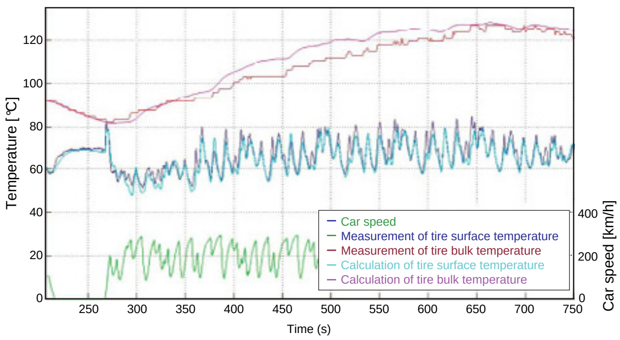


Fig. 21 Comparison of tire temperatures (rear)

5. Conclusion

As demonstrated by the technology introduced in this paper, vehicle dynamics simulation technologies play a significant role in developments relating to the dynamic performance of Formula One vehicles. In addition to the benefits of simulations in reducing costs and increasing development efficiency, the very process of observing phenomena, analyzing their mechanisms, and formulating models to develop a simulation technology in itself leads to dramatic enhancements in dynamic performance.

More recent developments, seeking further performance increases and enhancements in development efficiency, are progressing to the development and use of a driving simulator, which represents an evolution of vehicle dynamics simulation technology. The removal of the various external factors associated with track tests on a circuit enabled high quality test results to be obtained. The driving simulator was applied not only in evaluating the performance of development items, but also in the formulation of vehicle development concepts and the creation of indices for drive feeling in relation to dynamic performance. It is expected that these technologies will be actively employed in the development of mass production vehicles.

■ Author ■


Hideaki SHIBUE


Kazuhiro TANEDA


Yasutaka KITAKI

Weight Reduction and Stiffness Enhancement Technology in Formula One Chassis Development

Hajime WATANABE* Jun IWAI* Naoki HIROMASA*
Terukazu EBISAWA* Kazuhiro TANEDA*

ABSTRACT

When developing Formula One chassis, reducing weight while still satisfying the functions and requirements of each part was an issue for chassis design. The authors took into account the degree of contribution to circuit lap time, and found a balance between functions and weight reduction during chassis development. The method used consisted of reviewing materials and structures, and the weight of chassis parts was successfully reduced.

This paper introduces various examples of weight reduction.

1. Introduction

In chassis development, reducing weight and enhancing stiffness while satisfying the functions and requirements of each part is an issue that automobile design engineers are constantly working at in their daily development work, and Formula One chassis development is no exception.

Most racing car parts are related with the functions of accelerating, cornering, and stopping. Therefore, development placed particular attention on safety, and care was taken to develop parts that would not become factors resulting in retirement from a race.

In particular, parts that are directly linked to safety are given a Class A rank, and stress analysis is performed in the design stage to check that the prescribed safety factor is met before design release and manufacturing. Only the parts of which strength and durability have been confirmed by bench tests, can be introduced to circuit tests. Finally, parts for which safety has been confirmed in the running state in track tests are introduced to actual races.

In this manner, much effort was given to maintaining safety and quality, even though one characteristic of Formula One chassis development is that time resources are always limited. Normal annual development consists mainly of parts development that aims to reduce weight by a few grams while securing the same stiffness as the previous model, or parts development that aims to enhance stiffness by a few percent while maintaining the same weight.

However, Honda aimed for weight reduction that far exceeded the bounds of this normal development, by introducing new materials and structures. In addition, emphasis was placed on reducing the weight of unsprung parts and parts located far from the center of gravity or in high positions, taking into account the map shown in Fig. 1 and other factors, from the standpoint of reducing

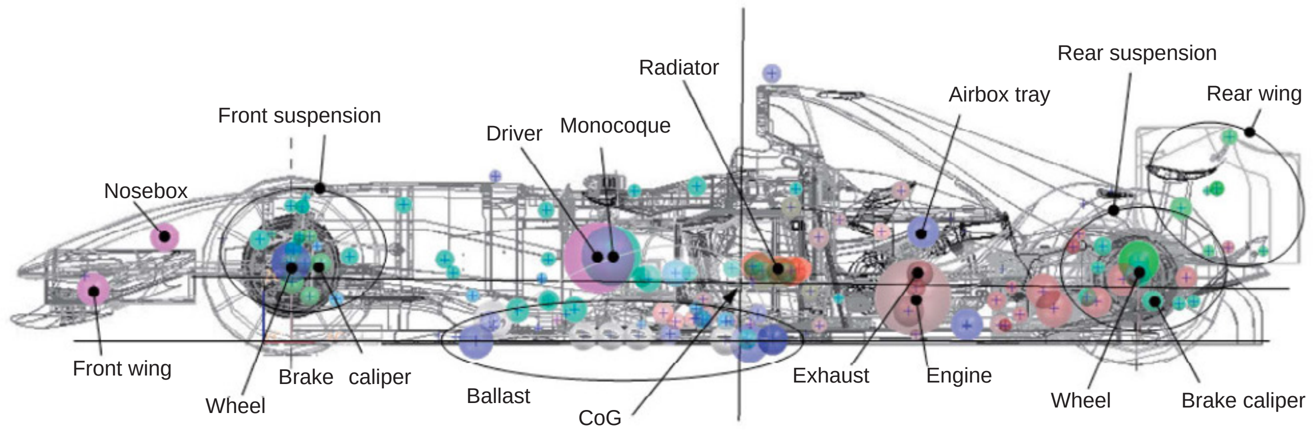


Fig. 1 CG_x versus CG_z plot for each component

* Automobile R&D Center

the moment of inertia. Furthermore, the contribution of chassis stiffness to performance was considered using evaluation standards that differed from those conventionally used for Formula One, and development also attempted to enhance the stiffness of key chassis locations including the engine.

2. Weight Reduction and Stiffness Enhancement of CFRP Parts

CFRP composites have high specific strength and specific stiffness, and are widely utilized in current Formula One chassis. Approximately 80% of chassis parts are made of this material, and most body parts in particular are made of CFRP composites.

Efforts to reduce weight and enhance stiffness in the body area focused almost exclusively to make the most out of the characteristics of CFRP composites. This was done by development of material or by applying new construction to the parts using small-lot production technology.

Weight reduction was investigated focusing on parts located far from the center of gravity, that is to say the nosebox, rear impact structure, and rollhoop, from the standpoint of reducing the moment of inertia. These parts are for crash safety, and must absorb collision energy by crumpling themselves. That requirement is stipulated in the regulations prescribed by FIA in such terms as the amount of energy absorbed, initial velocity, and rate of deceleration. We had to meet this collision requirement and reduce the weight at the same time while keeping the aerodynamically required shape and size, durability with respect to force inputs during circuit running, among other requirements. To realize this, a structure designed to increase the amount of collision energy absorption and the management of the manner how the part crumples during a collision was required.

CFRP parts are created by manufacturing a master model from a block of easily-machinable resin, and then producing a reverse mold by laminating pre-impregnated composites (prepreg), which are used to make molds, over this master model. The product materials are then laminated inside this mold, and formed in an autoclave. The manufacturing process involves a lot of manual work, so complex shapes and structures can be created.

However, being able to manufacture parts easily in a short time means that more time can be allotted to parts design, and more proposals can be investigated and tested. This can make a decisive difference when launching a new chassis in the beginning of each season, or when implementing a large-scale update partway through a season. Therefore, shortening the part manufacturing time is also an issue when aiming to reduce weight using new structures.

Typical examples of developing CFRP parts are described below.

2.1. Development of the Nosebox

The nosebox formerly used a sandwich structure (CFRP, aluminum honeycomb, CFRP) such as shown in Fig. 2 (a) to secure shape stiffness with respect to front collisions. However, this meant that the energy absorption ratio per unit weight was small.

Therefore, the honeycomb was eliminated and a pillared structure with pipe sections in each corner was used as shown in Fig. 2 (b). This increased the energy absorption ratio while maintaining the shape stiffness, and achieved a weight reduction of 13% compared to the conventional nosebox.

In terms of manufacturing technology as well, eliminating the honeycomb enabled one-piece molding, which reduced the process time by 30% compared to the conventional nosebox. However, establishing predictive design technology took time, so parts with the conventional structure had to be made before manufacturing parts with the new structure. Changes to the external shape occurred repeatedly before all the parts with the new structure were completed, and therefore, these parts could not be applied to races.

2.2. Development of the Rear Impact Structure

The rear impact structure is a part that had a high energy absorption ratio from the beginning. However, when the external shape was stipulated by Formula One regulations, an even higher energy absorption ratio was required to prevent an increase in the weight.

CFRP absorbs energy by crumpling, so the more fractured the structure after a collision, the higher the energy absorption ratio. Focusing on this characteristic, a thin-walled structure with a window frame shaped cross-section [Fig. 3 (b)] was used instead of the conventional thick-walled structure [Fig. 3 (a)], to

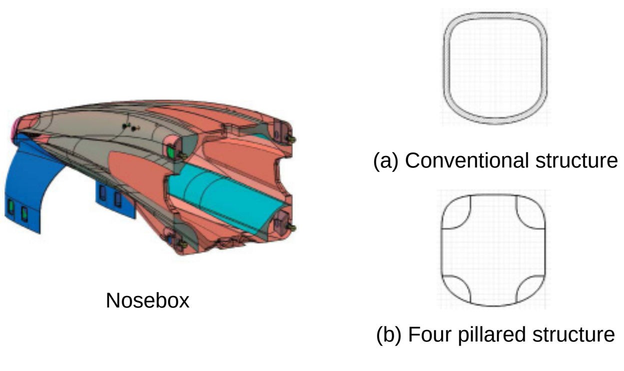


Fig. 2 Nosebox

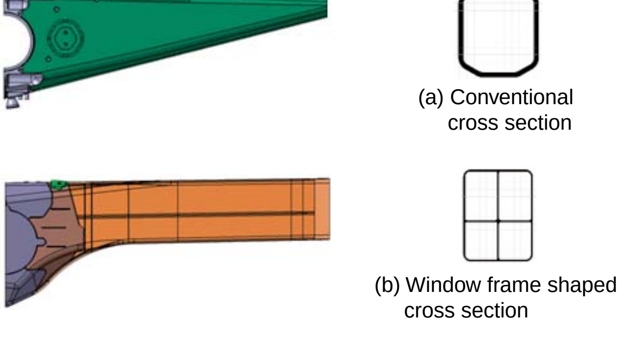


Fig. 3 Rear impact structure

achieve further fracturing of the components in the event of a collision.

This structure was created by laminating the window frame shape materials over a silicon mandrel, placing an outer mold containing the laminated outer shell materials over, and performing one-piece internal pressure molding.

The new structure achieved a 28% weight reduction, and was applied to races while continuing to evolve the design to more aerodynamic shapes every year.

2.3. Development of the RollHoop

The rollhoop must protect the driver's head in the event of rollover, and secure a sufficient air intake area for the engine. In addition, an aerodynamic external shape with respect to the rear of the chassis is also demanded.

A pentagonal shape as viewed from the front was introduced to reduce thickness and weight while still satisfying the above requirements. Within this structure, the roof lines of the pentagonal shape were set against to the load direction as stipulated by Formula One regulations. A thinner rollhoop that meets the requirements was successfully developed by increasing the load resistance of these portions to reduce deformation (Fig. 4).

This reduced the weight by 18% compared to the conventional rollhoop, and we were able to apply this new rollhoop to races. In addition, the shape of the connection with the monocoque was also reviewed when switching the monocoque, which achieved another 16% weight reduction.

As engine development advanced and it became possible to reduce the aperture area, the next step was thought to be to increase the sectional area of the rollhoop to further reduce weight. The new design received loads with a planer surface, and weight was reduced by making use of the shape stiffness of the larger section, using a lightweight core (Rohacell)(Fig. 5). Prospects for lamination were obtained through simple investigation using past results, and the maximum load was predicted by fracture propagation analysis using a nonlinear analysis tool. In this way, the new rollhoop was designed. A structure that does not fail even when subjected to the regulation load was achieved by efficiently distributing the load over the entire structure.

This realized a weight reduction of approximately 20%, and the new rollhoop was applied to races.

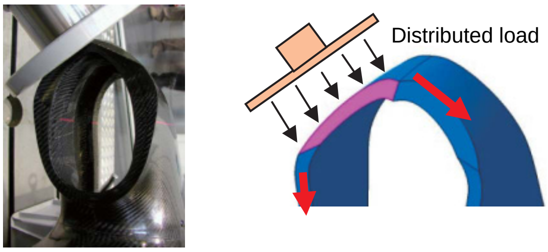


Fig. 4 Pentagonal rollhoop

2.4. Development of the Engine Cover

The weight of exterior cover parts such as the engine cover (Fig. 6) must be reduced to the greatest extent possible while realizing aerodynamic functions. Therefore, as long as the part stiffness is kept by the shape, a sandwich structure with the minimum layers is used.

Weight was further reduced by making the CFRP materials that comprise the inner and outer shells of this sandwich structure as thin as possible. This was achieved by using a new material developed jointly with the material supplier, and by applying lightweight resins. This new material was an open fiber material made by spreading carbon fiber yarn into a flat, thin layer, making the fibers into a plain fabric, and then soaking the fabric in resin to create a prepreg. This realized a molded thickness half or less than that of normal materials.

As a result, the engine cover weight was successfully reduced by 60% in the qualifying specifications. To achieve durability, applied areas are modified, and the part with open fiber material was put in to the race specifications.

This material was also used to laminate various other external cover parts, and contributed greatly to reducing the weight of non-structural chassis parts.

2.5. Development of the Monocoque

Clarifying how chassis stiffness contributes to vehicle dynamics performance was a major issue. An attempt was made to enhance vehicle dynamics performance by increasing the stiffness of the joint between the power train and the chassis, and by using the airbox mounted on top of the engine as a stiffener, but an increase in performance could not be confirmed.

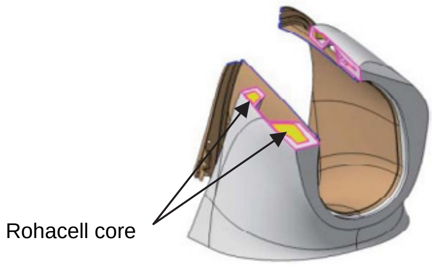


Fig. 5 Rollhoop structure

Fig. 6 Engine cover

Therefore, it was hypothesized that the degree of contribution of stiffness to vehicle dynamics performance is influenced more by the local stiffness of the suspension mounts than by the overall stiffness, and the monocoque weight was reduced with the sole restriction of equivalent local stiffness.

Figure 7 shows the weight reduction methods.

- Reinforcing materials were eliminated by allowing a 20% drop in the overall torsional stiffness.
- Exclusively developed side panel materials were applied.

The specifications were investigated using the local stiffness evaluation method employed in production car development, and the suspension mounts were reinforced as shown in Fig. 8. This showed that the weight could be reduced by 6.5% and the center of gravity lowered by 2 mm compared to the conventional specifications.

Track tests were performed to compare the new specifications with the conventional specifications. The results confirmed that even when the overall torsional stiffness is lowered, as long as the suspension mount stiffness can be secured, there is almost no effect on steering stability. Therefore, a monocoque that reflects these specifications was applied to races the following year.

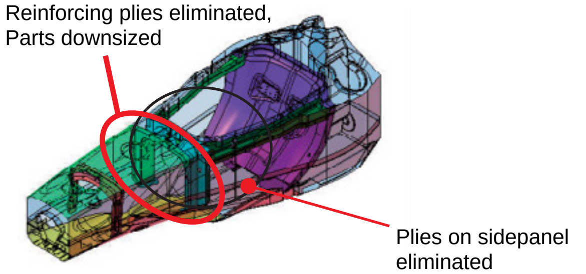


Fig. 7 Weight reduction method

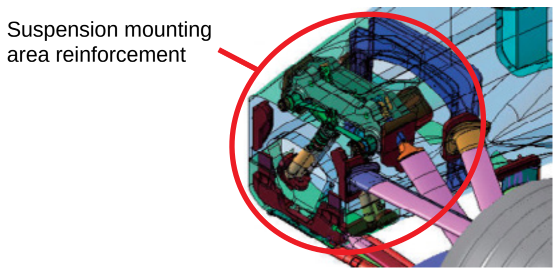


Fig. 8 Reinforcement method

3. Weight Reduction and Stiffness Enhancement of Functional Parts

Weight reduction and stiffness enhancement of functional parts such as the suspension, steering, axles, brakes, fuel system, radiators, and exhaust pipes is described below.

Development of these parts pursued part performance that enhances vehicle dynamics performance. This included maximizing tire performance by optimizing the suspension geometry, controlling ground contact load by tuning the spring and damper systems, optimizing the power steering assist characteristics, optimizing the steering variable gear ratio, developing brake cooling technology that uses dynamos, reducing fuel sloshing, and so on.

On the other hand, the Formula One car weight is prescribed, so it was also important to reduce part weights to help lower the center of gravity and increase the freedom of longitudinal weight distribution using ballasts. The authors constantly set goals to promote the simultaneous achievement of both performance and reduced weight.

Various examples of developed parts are described below.

3.1. Development of Suspension Parts

3.1.1. Development of anti-roll bars

Figures 9 and 10 show schematic diagrams of the suspension systems. The anti-roll bar is a spring that acts only during rolling, and is a key part that controls the rolling angle when cornering, and also determines the longitudinal balance of rolling stiffness. In the conventional structure, the rocker arm acts on the torsion bar via links as shown in Fig. 9, and the mechanism is such that in the reverse phase (roll), the right and left

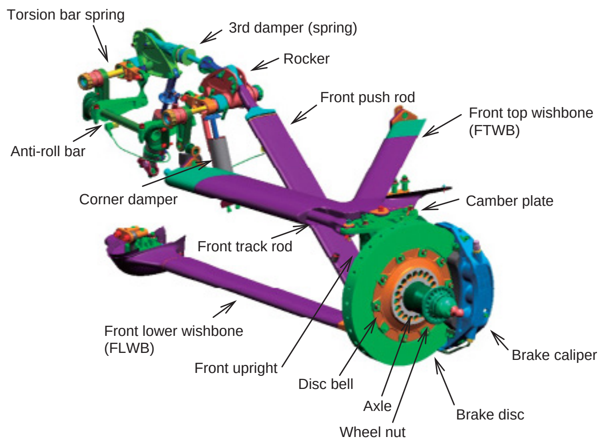


Fig. 9 Front suspension system (2003)

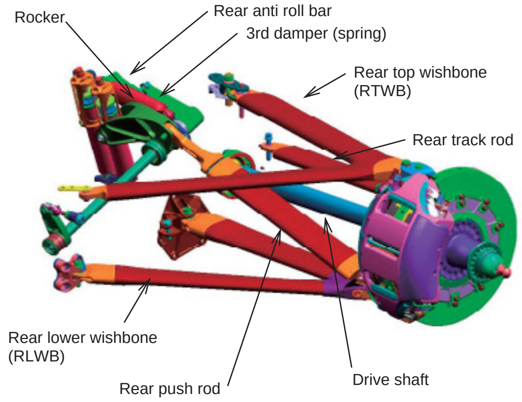


Fig. 10 Rear suspension system (2003)

link generate reaction force in the direction that works to operate the torsion bar. In contrast to this, the structure was changed to link the right and left flat springs with a double end spherical joint as shown in Fig. 11. In this structure, bending force is generated on the flat springs, which deform in the primary bending mode, only in the reverse phase, enabling the same work as that of the conventional structure. In addition, simplification of the structure achieved both weight reduction and enhanced space efficiency.

3.1.2. Development of wheels

The performances demanded of wheels are lightness in weight to minimize the unspring weight and the loss of turning inertia energy, and high stiffness with respect to the input from tire. However, the ideal balance between weight reduction and high stiffness was not clear, so development was first performed by selecting means of stiffness enhancement by modifying the shape and replacing materials with a minimum impact on weight.

For each suspension part, the degree of contribution to camber stiffness was investigated using bench tests after that, and wheel deformation was found to have an effect. Increasing the camber stiffness and enhancing the camber relative to the ground, even at the expense of weight, was judged to be advantageous for lap time, so development was performed with the target of increasing the camber stiffness of the standalone wheels by 80%. A topology optimization method was used to investigate the spokes, and the spokes were changed from the shape shown in Fig. 12 (a) to that shown in Fig. 12 (b). This achieved a 56% increase in the stiffness to weight ratio, which reached the target with a weight increase of approximately 10%. The new wheel design was confirmed in track tests to enhance the rear lateral force and rear stability, and was therefore introduced to races.

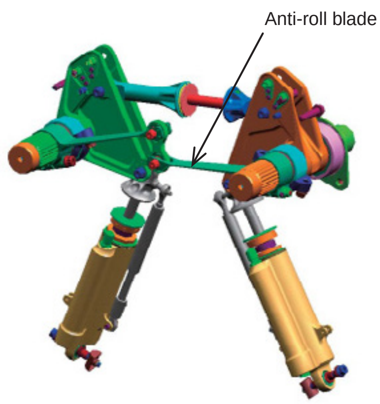


Fig. 11 Front anti-roll blade

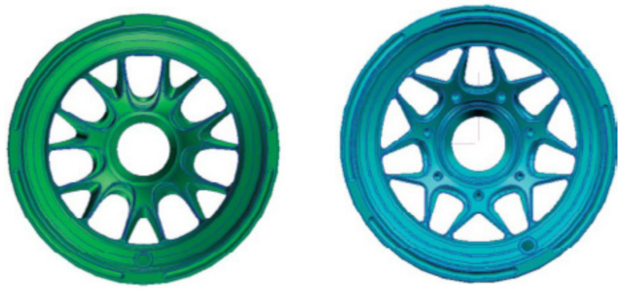


Fig. 12 Wheel spoke topology optimization

3.2. Development of Brake Parts

Brake development focused on the issue of how to achieve a high level of balance between the initial braking force, braking force stability, and drag reduction. One method attempted was to reduce the caliper body weight. Formula One regulations stipulate that the caliper body material must have a Young’s modulus of 80 GPa or less, so this essentially limits the material to aluminum alloy. Among various aluminum alloys, A2099 was selected, which features good durability (high fatigue strength characteristics), low density, high stiffness, high specific strength, and heat resistance strength. Hydraulic stiffness is important for the caliper body to secure a good pedal feeling, so the center bridge shape was reviewed in consideration of the stiffness balance, and unnecessary material was eliminated from around the cylinder, which succeeded in reducing the weight by 6% (Fig. 13).

3.3. Development of the Water Radiator

Water radiators and oil radiators are mounted in Formula One cars. Water radiators radiate the heat from engine cooling water in the same manner as those in production cars, and are keeping the temperature ranges at high engine efficiency. The cooling water is pressurized to raise the boiling point and to prevent cavitation. Current Formula One regulations stipulate to install a pressure relief valve that opens at 3.75 bars, and this valve is supplied from FIA.

The pressure upper limit is stipulated, and the liquid side heat radiating area has only a small effect, so development of the radiators was performed first for the water radiator side with the goal of weight reduction (Fig. 14). Development was initially started for the qualifying specifications so that priority could be given to weight reduction, even at the expense of durability. Application to the qualifying specifications started from 2001, and the design was repeatedly modified for race use thereafter, and introduced to all races from 2006. Even though the pressure is stipulated, the radiator is still

Fig. 13 Brake caliper

subject to approximately four times the pressure in production cars, so the outside of the stacked tubes is subject to the greatest deformation, and the stress increases in this area. In the initial type water radiator, focus was placed on the heat transfer surface area of the water side, which has a lower contribution to heat radiation performance than the heat transfer surface area of the air cooled side. The inner fins were eliminated, and the outer frame was utilized to suppress bulging in each tube due to the water pressure and the force from the engine. The tube thickness was increased in locations with local increases in stress, and measures were also taken such as reducing the fin thickness, while taking into account pressure resistance. The size was reduced compared to the original base specifications, and performance was enhanced by approximately 9%, which successfully reduced the weight by 37%. However, this structure had its limits when considering a chassis package with enhanced aerodynamic performance, so exclusive tube was developed thereafter. Louver fins were also employed with the aim of increasing toughness against chipping, and together with a greater degree of shape freedom and enhanced durability, achieved further weight reduction of approximately 7% and enhanced heat radiation performance of approximately 4.4%. In addition, boundary layer formation on the water side was considered, and separate specifications that further increase heat radiation performance by approximately 8% was also ready to be applied. Copper, which has a high thermal conductivity, was also investigated as the material, but aluminum was used from the standpoint of weight.

3.4. Development of Titanium Exhaust Pipes

The requirements for exhaust pipe materials are strength, resistance to oxidation, and fatigue strength, when subjected to high temperatures, and a low coefficient of linear thermal expansion. Exhaust does not have a uniform temperature distribution due to the exhaust flow, and resonance also occurs at some engine speeds. Therefore, durability was analyzed and the exhaust pipes were designed in consideration of these effects. That said, exhaust reaches a maximum temperature of 1100 °C, so the materials that can be used are limited, and Inconel is generally used.

A titanium alloy with a low specific weight (density 4.4 g/cm^3) approximately half that of Iconnel (density 8.5 g/cm^3) was used to develop titanium exhaust pipes with the aim of a weight reduction of approximately 30% (Fig. 15). Titanium oxidizes at high temperatures, so development of an anti-oxidation treatment was a key element. The anti-oxidation surface treatment applied was NCC treatment, which consists of first plating parts with aluminum, and then burying the parts in aluminum and alumina powder. This surface treatment generates a uniform Ti-Al alloy layer on the surface of the Ti base material, and oxygen reacts preferentially with aluminum at high temperatures, which enabled the strength of the base material to be maintained. The wear resistance of joint insertion parts must also be increased, so nickel-chromium treatment was applied as surface treatment according to the location. In addition, the exhaust temperature distribution differs according to the exhaust flow and other factors, so also considering the manufacturing efficiency, various Ti materials were used.

Durability of 600 km or more was confirmed by track tests, but not all requirements were satisfied, so the titanium exhaust pipes were not introduced to races.

3.5. Development of Carbon/Carbon (CC) Exhaust Pipes

Like the titanium exhaust pipes described above, which were developed mainly with the goal of weight reduction by material substitution, carbon (CC, density 1.6 g/cm^3) exhaust pipes were also developed to further reduce weight. Normal carbon materials have carbon fibers in resin matrix. However, CC is a carbon fiber composite that uses carbon as a matrix. That is to say, CC exhaust pipes are exhaust pipes made of charcoal. Development started with a target durability of 25 laps, aiming for use in qualifying rounds.

The first step was the collector and tail parts, but the ultimate goal was a complete set of CC exhaust pipes including the primary, and a weight reduction target of approximately 39% was set. In addition, conventional exhaust pipes are made by welding together Iconnel plate materials. This made it difficult to set a thinner plate thickness, and manufacturability and durability issues meant that excess materials were unavoidable. In contrast, CC exhaust pipes are manufactured using molds, so the minimum thicknesses can be set regardless of welding toughness, and the intermediate sections also

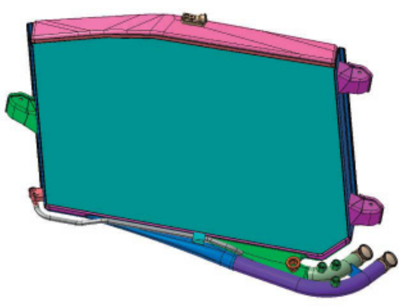


Fig. 14 Water radiator

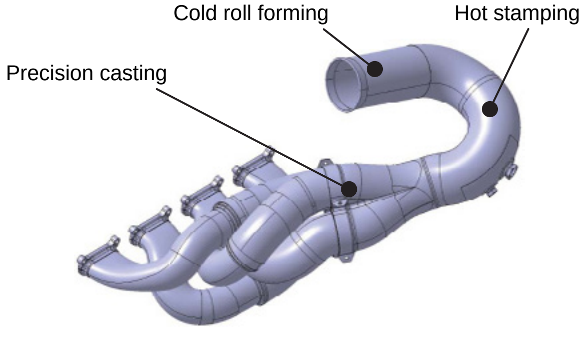


Fig. 15 Titanium exhaust

have a high degree of shape freedom. The final specifications aimed for a compact layout that would be able to enhance chassis aerodynamic characteristics.

In the first step with only the collector and tail parts made of CC, gaps opened at high engine speeds between the CC, which has an extremely low coefficient of linear thermal expansion, and the Iconnel materials used for the primary. This produced the issue that the air-fuel ratio did not stabilize, so a gasket was also developed to fill these gaps that appear at high temperatures. Pure carbon has high heat resistance but low oxidation resistance, and may burn up if measures are not taken, so development of a surface coating to prevent oxidation was a key element. Ultimately, resistance to surface oxidation for 3 hours at 900 °C was successfully secured using a four-layer coating, and durability of 30 laps was confirmed in engine dyno tests, which exceeded the initial target. Efforts were also made at the same time to shorten the manufacturing time, but application only to the qualifying specifications became impossible, so the development was halted.

4. Conclusion

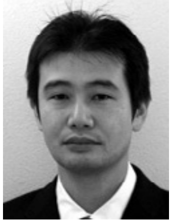
Formula One regulation stipulates a minimum weight of 605 kg, including the driver. Therefore, the weight reduction does not contribute to reducing the absolute weight of the car, but to enhance vehicle dynamics performance, which means increasing chassis competitiveness. This is accomplished by replacing the reduced weight with a ballast to lower the center of gravity, concentrate the weight in the center of the chassis, and expand the longitudinal weight distribution adjustment range.

In addition, detailed regulations also stipulate the materials allowed to use, so there is a limit to weight reduction that can be achieved by simple material replacement, which made us to work on approaches such as structural changes based on an understanding of the required functions and performance were needed.

(1) Reducing the weight of parts located above the car's center of gravity is particularly effective from the standpoint of lowering the center of gravity. This was achieved with the rollhoop, engine cover, water radiator, exhaust pipe, and other parts.
(2) Reducing the weight of parts that overhang from the front and rear tires is effective from the standpoint of enhancing turn-in performance (reducing the moment of inertia around the Z axis (Izz)). This was achieved with the nosebox, rear impact structure and other parts.
(3) Reducing the weight of the suspension parts and other unsprung parts is effective from the standpoint of enhancing vehicle dynamics performance, and targeted parts such as the brake calipers, anti-roll bar, and wheels. In addition, not only reducing the weight of these parts, but also stiffness and function enhancements were achieved.
(4) The monocoque weight was also reduced from the standpoint of optimizing the stiffness.

In this manner, the development was performed to reduce the Formula One chassis weight, and achieved a certain degree of effects. The development directions and issues in the future was to continue ceaseless development to reduce weight, focusing on higher contributions to reducing circuit lap times, and using ideas conceived during the course of development for further enhancement of functions.

■ Author ■


Hajime WATANABE


Jun IWAI


Naoki HIROMASA


Terukazu EBISAWA


Kazuhiro TANEDA

Measurement and Analysis Techniques of Formula One Chassis Development

Hideaki SHIBUE* Tomokazu SUZUKI* Mamoru URAKI*
Akihiro IDA* Atsushi TSUBOUCHI* Yasutaka KITAKI*

ABSTRACT

Measuring and analyzing chassis data are the basics of Formula One vehicle dynamics development. The acquired information is used in every stage of development, from design concept to race management. This paper explains examples of data analysis techniques including competitor performance analysis, dynamic performance indices used for chassis development and various techniques of measurement systems and sensors used in Formula One.

1. Introduction

The decisive difference of developing a Formula One car against a passenger car is that the engineers cannot drive and feel the cars themselves. Therefore, various high quality onboard data and analysis tools are needed to accurately understand the vehicle dynamics issues in order to improve car performances. Some of them are standardized to maximize tire performance during each run. These data are obtained through real-time processing using a telemetry system and being analyzed automatically so that the engineers can determine car setup for the next run whilst the car is still on the track. Simulating tools had been developed as well to predict and understand the phenomenon more in detail.

This paper introduces examples of the cutting edge measurement systems and sensors, competitor performance analysis technique including simulation tools and the latest analysis techniques of the dynamic performance indices.

2. Measurement Systems

2.1. Overview of Measurement Systems

The tasks of gathering data while running on a circuit track, analyzing that data, and implementing countermeasures form an important cycle in Formula One vehicle dynamics development. This cycle should be repeated accurately within a short time, so advanced measurement systems are built into Formula One chassis from the design stage. In this sense, a Formula One chassis could itself be considered a measurement apparatus. Even during races when performance is given the highest priority, some 60 different types of sensors with close to 100 channels are mounted on the chassis. During tests when greater emphasis is placed on data acquisition, even more sensors are mounted with as many as 170 channels. In addition, video cameras and recorders are mounted to record moving images, and driver operations, racing lines, infrared images of tire temperatures, air flows, and other images are recorded in sync with the measurement data according to the application.

Figure 1 shows an overview of the measurement and analysis system. The sensor signals are first processed inside the onboard ECU and recorded in the data logger. In addition, some performance evaluation indices are also calculated inside the ECU in real time, and recorded together with the sensor data. Some of this data is sent by real-time telemetry to the pit, where it is processed instantly and continuously on a server (vTagServer:

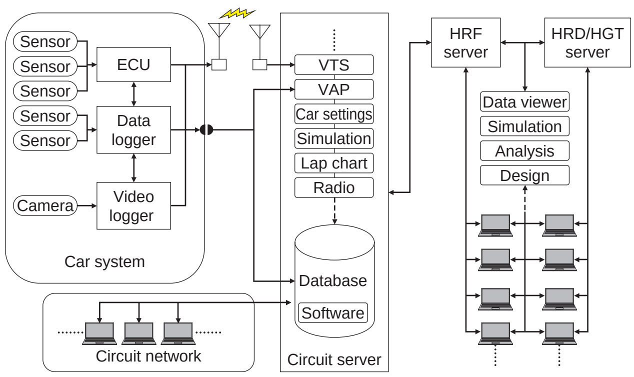


Fig. 1 Measurement and analysis system

* Automobile R&D Center

VTS). Performance indicies are also calculated simultaneously at this time, enabling engineers to obtain analysis results and investigate the next setup changes before the car returns to the pit. This real-time data is distributed over a network, enabling simultaneous use not only in the pit, but by many members in the team factory and other locations with Internet connections.

When the car returns to the garage and the data downloading starts, the automatic data analysis system (VAP: Vehicle Analysis Package) transforms the data immediately to physical values within minutes. This data consists of approximately 2500 items that record the car status at sampling rates of up to 10 kHz, and the data volume can reach up to several GB per run. In addition to running data, virtually all information related to running is also managed centrally on the server. This information includes the lap times, setup information, driver comments and wireless audio for each car, weather and track condition information, work instructions and communication history between members, trouble information, and simulation analysis results.

The data is also transferred to the team factory (HRF) and Automobile R&D Center Tochigi (HGT) servers, enabling engineers in various fields to promote development while sharing data on the same information infrastructure, without the limitations of place or environment.

Including both races and running tests, Formula One running is performed for approximately 125 days per year (2008 results). Running data for up to 200 laps per day may be recorded, so measures to increase and maintain the accuracy and reliability of this enormous amount of measurement data are demanded. On the other hand, the chassis needs to be as slim, lightweight and have as low a center of gravity as possible, so the weight and space that can be used are limited. In addition, the engine that is rigidly connected to the chassis produces vibration over 3000 G, and the exhaust heat over 1000 °C. Therefore, efforts are made to develop original sensors and new measurement methods to support the needs of high accuracy, compact size, light weight, and high durability. Specific examples are introduced below.

2.2. Strain Gauged Wheel (SGW)

A six tire forces measuring wheel system (SGW) has been developed for Formula One excessive use. The requirements were to give accuracy of 1%, compact and easy to mount. This SGW uses a wireless system that is lighter weight and has less aerodynamic effect than the conventional wired system (Fig. 2). This reduced the aerodynamic effect of mounting, shortened the mounting preparation time, and enabled the acquisition of useful tire data with limited track testing (Fig. 3).

The force generated on the tire is measured using a strain sensor-type load cell configured by four pairs of bridge circuits (4 gauge method) built into the wheel. In addition, a measurement value processing box, a rotation angle calculation box, a battery box, and a transmitter box that wirelessly transmits the data to a matrix calculation box (MCB) installed on the chassis, are mounted on the wheel disc surface. Matrix calculations and rotation angle and temperature compensation processing are performed inside the MCB on the data received by the chassis side to obtain six-component tire force data that is transmitted via CAN to the data logger on the chassis.

Loads of up to 30 kN and a bending moment of up to 4000 Nm can be measured under an operating environment with a maximum speed of 360 km/h and maximum temperature of 120 °C, which covers tire force measurement in the limit running state of a Formula One car. In particular, heat from around the brakes, which gets as hot as 700 °C or more, affects the temperature characteristics of electronic components including the strain sensors, and may result in a drop in measurement accuracy. Therefore, temperature compensation logic and a corresponding calibration method were established to address the issue of heat-induced changes in characteristics.

2.3. Tire Bulk Temperature Sensor (TBTS) and Tire Internal Surface Temperature Sensor (TIST)

Formula One tire performance is exercised by precisely controlling the temperature, load, slip, wear, and the like. Of these, the tire internal (structural portion) temperature is one of the most important control elements. Usually the tire surface temperatures are

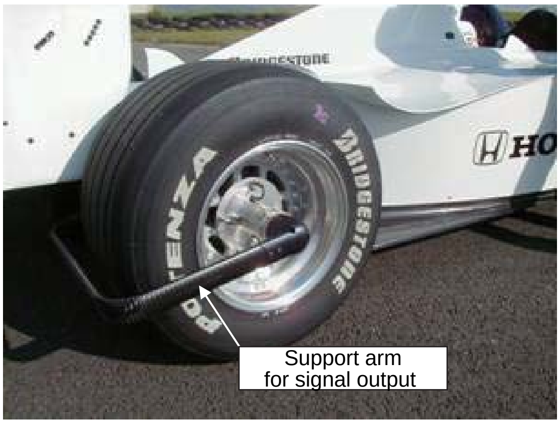


Fig. 2 Conventional SGW

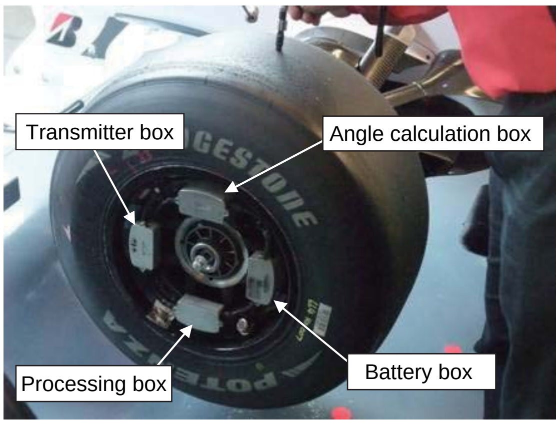


Fig. 3 Wireless SGW

measured during the run, and the internal bulk temperature inside the garage using probes. A hole can be made in part of the tread rubber and a thermometric element embedded to measure the temperature of passenger car and truck tires while running. However, Formula One tires have thin tread rubber just several mm thick, and the high loads and high-speed rotation make it a challenge to apply a similar method. Therefore, two types of sensors were developed: TBTS that embeds elements inside the tread rubber during tire manufacture (Fig. 4), and TIST that affixes elements to the tire internal surface (Fig. 5). A compact multi-channel telemetry system was also developed at the same time to acquire the data from the rotating tires, and this enabled measurement at racing speeds.

2.4. Multi-point Tire Temperature Sensor and Onboard Thermal Camera

To obtain continuous performance from Formula One tires, it is also important to maintain a proper pressure distribution within the tire contact patch. The load on the tire due to this pressure distribution can be estimated by measuring the tire surface temperature distribution during running, so infrared tire temperature sensors capable of simultaneously measuring multiple points were developed (Fig. 6). First, a 5-point sensor using multi-point thermopiles was developed to support grooved tires (5 ribs), and the number of points was then expanded to 8 and 16 points to increase the distribution measurement resolution.

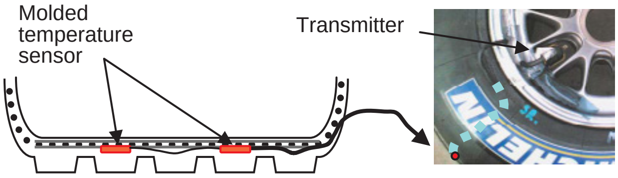


Fig. 4 Tire bulk temperature sensor (TBTS)

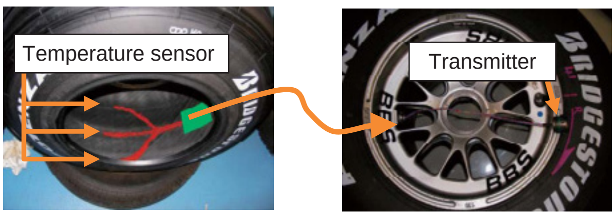


Fig. 5 Tire internal surface temperature sensor (TIST)

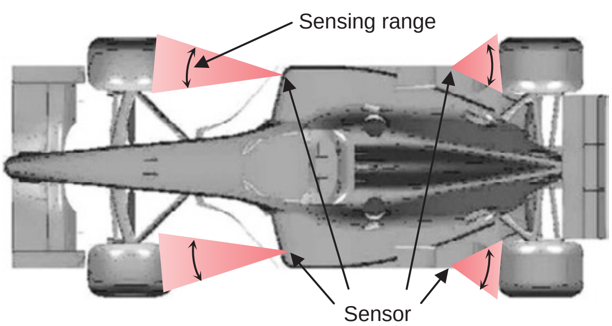


Fig. 6 Multiple tire surface temperature sensor

In parallel with this, a method that uses an onboard infrared camera to measure the tire surface temperature was also developed. A compact infrared camera was installed on the roll hoop where the video camera for TV broadcasts is normally mounted, and temperature images of all four tires were recorded using a four-sided mirror (Fig. 7). There is also the method of mounting multiple cameras aimed at each tire, but the mirror method has the merits of light weight, compact size, and high data quality due to recording all four tires at the same calibration. The recorded temperature images alone are useful information for evaluating the tire temperature distribution, but even more information can be extracted by analyzing the images using originally developed software, and converting into temperature distribution data.

2.5. Strain Gauged Suspension (SGS)

The strain gauged wheels that directly measures the force generated on the tires during running are useful in track tests that evaluate dynamic performance, but the increase in weight and the aerodynamic effects of the special wheel shape cannot be eliminated, so use is limited to tests. That is to say, a condition for the application of measuring systems to races is that there be no effect on vehicle dynamics if at all possible. Therefore, development of the SGS was promoted based on the concept of driving six tire forces from the loads on the suspension arms (Fig. 8).

A Formula One car suspension uses a double wishbone suspension type, but each arm is basically viewed as a

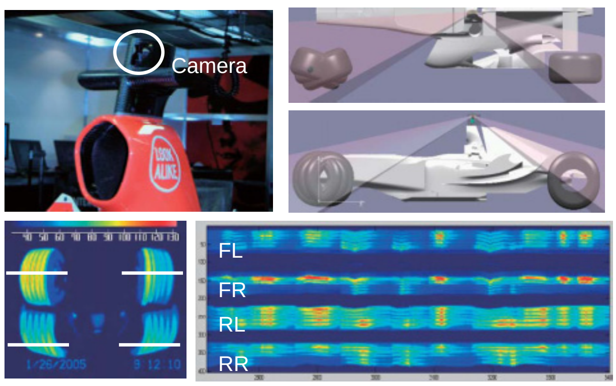


Fig. 7 Onboard thermal camera

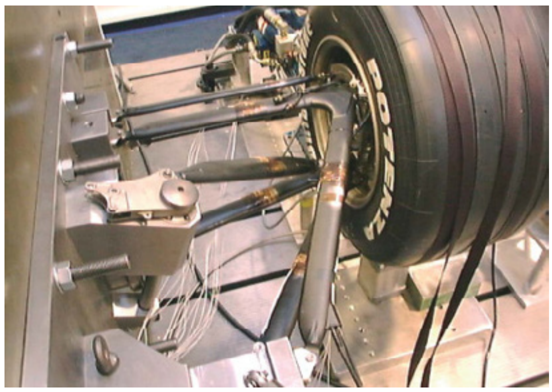


Fig. 8 Strain gauged suspension (SGS)

single straight link, and the six tire forces can be calculated from the combined force of each axle force including the push rod. That is to say, the six tire forces are obtained by performing finite element analysis and bench tests beforehand to derive the component force matrix that expresses the relationship between the six tire forces and the suspension arm axle forces, and then applying the inverse matrix of this component force matrix to the axle force measured during running. Realization of this system requires technologies that can handle A-shaped arms that are not straight links, take into account the bending stress due to the flexure joint, support changes in the component force matrix due to cornering and the suspension stroke, and take into account the effects of temperature, vibration, and other factors.

2.6. Wing Load Cells

The aerodynamic drag and lifts are usually calculated by using pushrod loads, but a direct wing load measurement system has been required to further understand the phenomenon around the wing elements itself. The requirements was to create a load cell that does not affect the aerodynamics, having ability to change wing elements without touching the load cell unit and off course strong enough to run on the circuit. A pillar type load cell (Fig. 9) was used for the front wings and a rear impact structure integrated load cell type (Fig. 10) on the rears.

2.7. Measurement Technology Using Test Rigs

In addition to track tests, bench tests are also often used in Formula One chassis development. Tests under stable environments with few undetermined elements enable the acquisition of detailed and precise information that cannot be obtained from track tests. In particular, the use of bench tests to obtain data that is a challenge to measure during running, such as the tire contact pressure distribution, helps to further deepen understanding of vehicle dynamics mechanisms.

The Contact Patch evaluation (CPE) system was developed as a bench test system that measures the tire contact pressure distribution in the state with longitudinal and lateral forces acting on the tire. The sensor sheet is resistant to compression but weak in the shearing direction, so it was covered by a protective plate (0.1 mm thick SUS material), and then a non-slip sheet was affixed to the top of the protective plate (Fig. 11). Use of this system in combination with the Dynamic Vehicle Simulator (DVS) or other bench tester enables quantitative evaluation of the suspension dynamic characteristics and vehicle setup from the viewpoint of proper contact between the tire and the road surface (Fig. 12).

Figure 13 shows an example of analyzing the

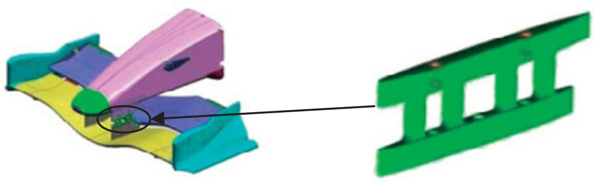


Fig. 9 Front wing load cell

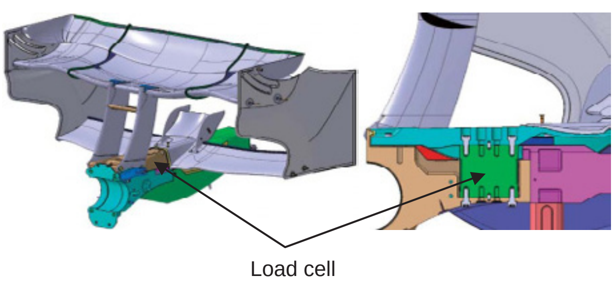


Fig. 10 Rear wing load cell

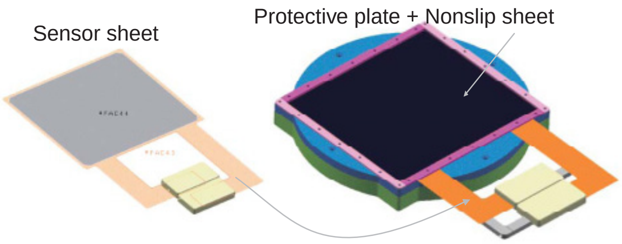


Fig. 11 CPE sensor

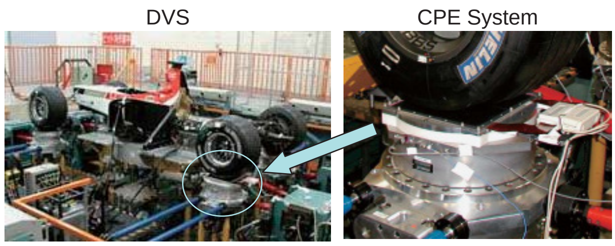


Fig. 12 CPE system on DVS

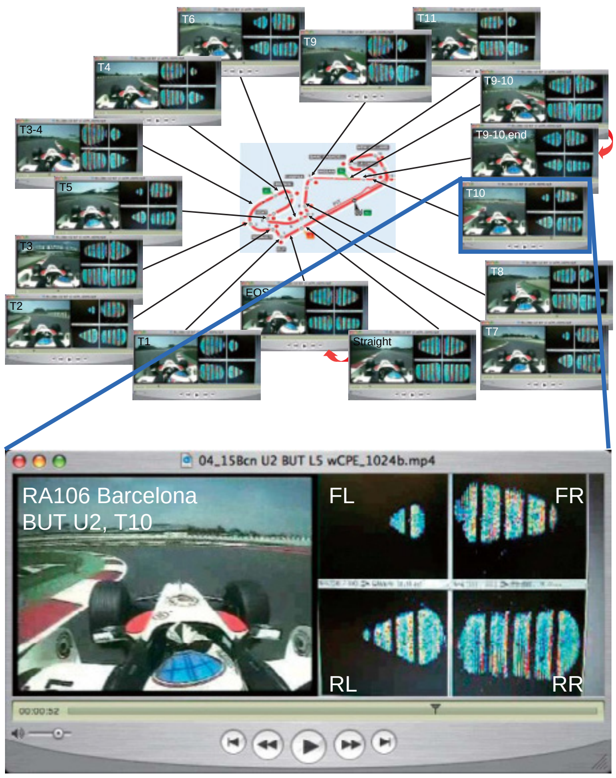


Fig. 13 Circuit simulation with CPE

constantly changing tire contact pressure distribution during running, simulated using the circuit simulation function of the DVS.

3. Competitor Performance Measurement and Analysis Technique

Understanding the primary factors for differences in lap times with competing cars during races is important for determining development directions. The ideal is to originally create the best in all technical areas, but in reality it is necessary to analyze the weak points of one's own car and formulate countermeasure policies to make efficient use of limited time and resources. Unlike production car development, it is unlikely that competing cars can be obtained for direct comparison and analysis, so original technology was developed to determine and analyze the primary factors for differences in performance without actually touching competing cars.

3.1. Acoustic Analysis

Regulation changes in 2008 added a maximum engine speed limit of 19000 rpm, but prior to that it was important to know the engine speed of competing cars as a guideline for estimating the maximum power output. During TV broadcasts of races, sound is broadcast simultaneously with the image from onboard cameras, and the engine speed can be accurately known from the frequency of the engine noise. At test tracks without TV broadcasting, the sound is recorded and speed measured simultaneously using a PC, microphone and speed gun, and the engine speed is calculated from the engine sound by compensating for the Doppler effect. This data can then be further processed to estimate the vehicle speed diagram, gear ratio, and other information (Fig. 14).

3.2. Infrared Thermal Images

The load on the tires differs for each car due to differences in the weight distribution, height of the center of gravity, suspension geometry, setup, aerodynamic characteristics, driving style, and other factors, and results in differences in the tire surface temperature distribution. Differences in the characteristics of each car can be estimated by installing thermal cameras on the course side, imaging the tire surface temperatures of each car during running, and comparing the differences in the temperature distributions (Fig. 15).

3.3. Image Analysis

Various analyses can be made based on TV broadcast images and images filmed by digital cameras or other means. The tire contact camber angle and the chassis roll angle can be estimated from images that filmed cornering (Fig. 16). Analysis of this information together with the abovementioned thermography temperature data enables an understanding of differences in the chassis characteristics, suspension characteristics, and tire load conditions.

Overlaying the running images of multiple cars filmed during circuit running enables analysis of differences in vehicle speeds, braking points, racing lines, handling characteristics and other information (Fig. 17). In addition, minute time differences over the filmed section can be calculated from these images. Time differences over the entire course can also be calculated from the onboard camera images of TV broadcasts.

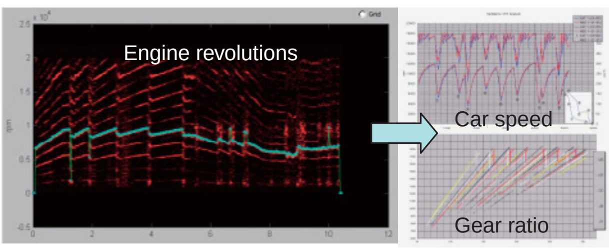


Fig. 14 Acoustic analysis

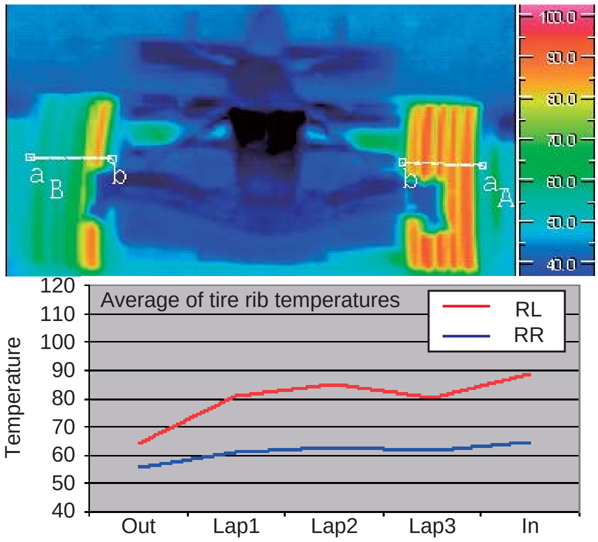


Fig. 15 Infrared thermal image

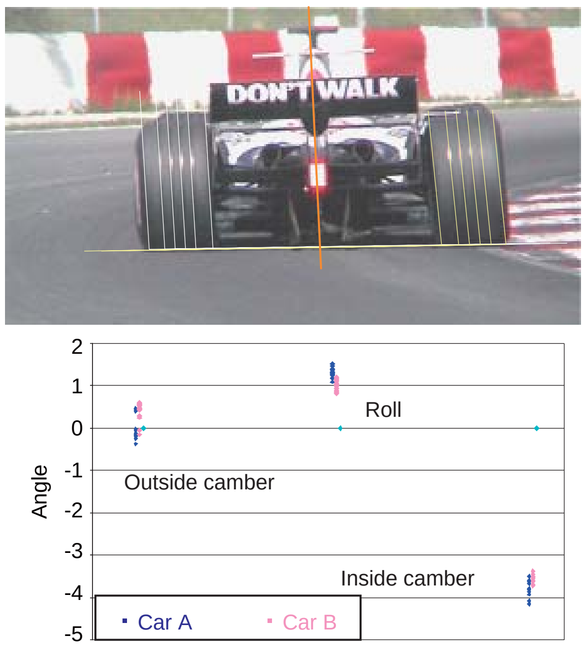


Fig. 16 Image analysis

3.4. High Speed Video Image Analysis

Various pieces of information can be obtained by using high-speed cameras that have a faster shutter speed and a higher frame rate than normal video cameras. The slip ratio (traction control performance) of the driving wheels during acceleration can be calculated from the difference in the rotation angles of the front and rear tires, and the tire surface wear conditions can also be observed (Fig. 18).

3.5. Three-dimensional Modeling

Three-dimensional models are created using multiple photos to understand the aerodynamic components, the suspension geometry and other items (Fig. 19). Comparing these models on the CAD system enables us to help validate the competitor performances.

Fig. 17 Image overlay processing

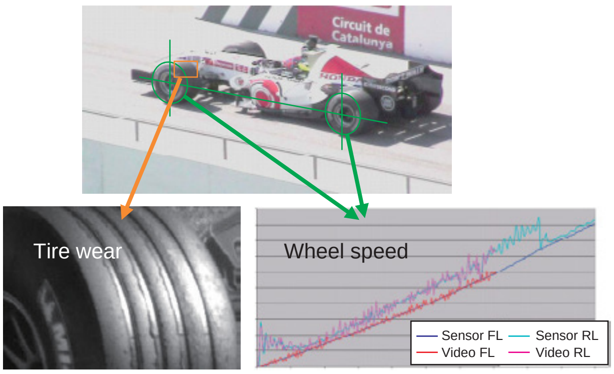


Fig. 18 High speed video analysis

Fig. 19 Modeling in three dimensions

4. Measurement Data Analysis Systems

Aerodynamic characteristics (down force and drag) and the three-component force (wheel load, longitudinal force, lateral force) acting on the tires during running are important information for analyzing Formula One dynamics. This information is used to determine the policies for aerodynamic development and tire management and the vehicle setup. However, it is a challenge to directly measure these forces, so it is required to perform high level estimation using a combination of various sensor data, suspension characteristic information obtained from rig tests and needless to say about the vehicle dynamics understanding.

During races and track tests, well over 100 different pieces of chassis performance data are required for analysis, which makes development of a system that automatically calculates vast amounts of chassis performance data vital to enable swift analysis. Therefore, an automatic analyzing system called the Vehicle Analysis Package (VAP) was developed to support performance analysis. This system used the onboard data and the suspension model information and more to say it allowed real time analysis using the telemetry system.

4.1. Overview of VAP System

The VAP system is introduced below using the example of calculating the down force, which is a main analysis item. Thus far limitations on down force calculations meant that evaluation of aerodynamic performance during running had been possible only under the limited conditions of mode running at a constant vehicle speed on a straight track. In contrast, dozens of sensor values and complex models need be used to enable calculations over all ranges, including braking, driving and turning, while running at racing speeds. Figure 20 shows an overview of the down force calculation model. Here, the down force is derived from the calculation results for the two main divisions of the wheel load and the inertia force.

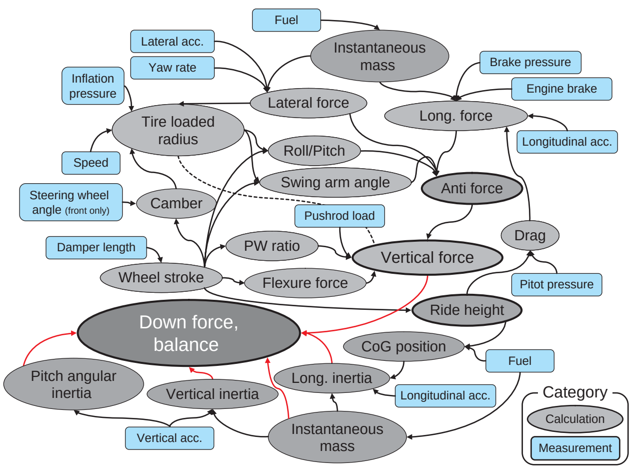


Fig. 20 Overview of down force calculation in VAP

The wheel load comprises the load on the pushrod, which bears the main load, the flexure force due to the bend arm, and the jack-up force due to the suspension geometry. Calculation of the jack-up force in particular also takes into account the suspension stroke, tire deflection, and other factors, so the wheel load is calculated using a combination of over 20 different sensor values and suspension and tire characteristics models.

The calculation of inertial force is needed in order to calculate the load transfer amount due to braking, driving and turning, and to eliminate variation in the wheel load, which is due to vertical centrifugal force resulting from the vertical bending radius of the circuit course, from the down force calculations. This means that the acceleration calculation accuracy is important, and requires accurate calculation of the pitch and roll angles and the height of the center of gravity.

The calculation accuracy is verified using a wind tunnel and bench testers such as the DVS (Fig. 21).

Figure 22 shows the results of accuracy verification using the DVS. Here, circuit simulations were performed using running data, and the VAP calculation results were compared with the down force applied as a load from the DVS. The results show a close match for both the front and rear wheels.

4.2. Introduction of Data Analysis Systems that Support Telemetry

Regulation changes implemented by the FIA from 2008 onward mandated the introduction of a common ECU made by McLaren Electronic Systems (MES). One of the functions of this ECU is a telemetry data analysis system (vTagServer: VTS).

The main function of telemetry systems thus far had been to transmit the information from onboard sensors to the pit, and the main role for this had been to monitor trouble occurring in the chassis while running. In contrast, one of the functions strongly desired from the standpoint of vehicle dynamics development was real-time analysis of dynamic performance while running. At circuits it is necessary to analyze running data, determine the next setup, and determine which running modes need to be added, all within the limited time after the vehicle returns to the pit until the next run. Therefore, real-time analysis was strongly desired by engineers engaged in vehicle dynamics development.

With the introduction of this new telemetry system, the logic of the automatic data analysis system was transplanted to the Simulink model, which achieved real-time data analysis by processing data on the VTS. In addition, the VTS can also be used as a post processing system, and development of an integrated system is proceeding with the aim of realizing a seamless environment from real-time analysis to more detailed analysis (Fig. 23).

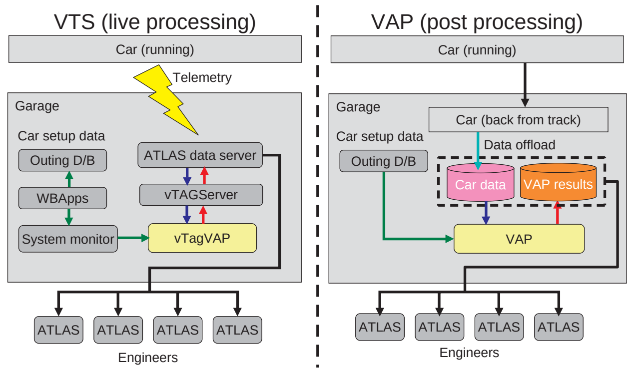


Fig. 23 Overview of live and post processing system

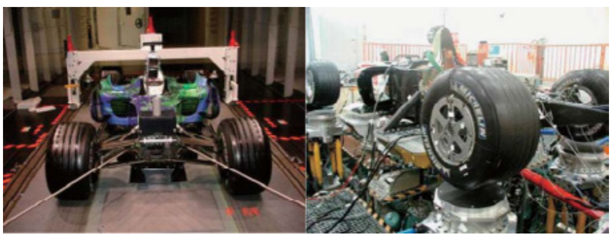


Fig. 21 Correlation test with wind tunnel and DVS

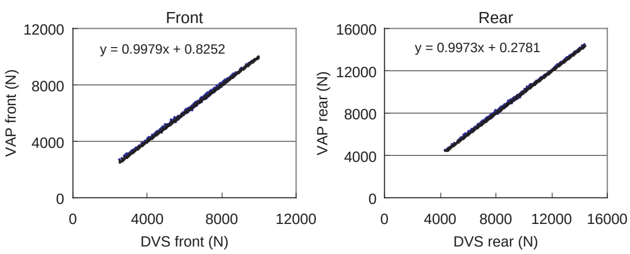


Fig. 22 Correlation test on DVS

5. Creation of Vehicle Dynamics Indices

To evaluate the analysis data as an expression for vehicle dynamics, and to use those results for vehicle design and setup at circuits, the ability to extract physical quantities that represent performance and to discuss these quantitatively and simply is required. That is to say, dynamic performance indices are important.

Without dynamic performance indices, it is a challenge to quantitatively compare multiple setups in a short time at circuits, where speed is required, and such work cannot help but rely on driver comments and the experience of the engineers. This also makes quantitative investigation a challenge in the area of vehicle design, such as when discussing vehicle dynamics targets or tradeoffs between various design elements.

5.1. Development of Braking Stability Indices

Vehicles with good braking stability not only enable shorter lap times by delaying braking, but also increase the chances of overtaking using braking. For these reasons, braking stability is one of the most important dynamic performance in Formula One. However, while it is discussed as the driver's sense of "confidence in the car while braking," there was no index that expressed the braking stability in a quantitative manner. Therefore, braking stability indices were developed as an index of dynamic performance.

5.2. Candidates for Braking Stability Indices

Ten different physical quantities thought to potentially express braking stability were hypothesized as index candidates, and each candidate was verified to determine whether it was suitable as an index. The typical index physical quantity candidates are described below (Fig. 24).

(1) Stabilizing yaw moment margin (MM)

The body slip angle was imaginarily increased from the current car state, and the driver was hypothesized to feel confidence according to the difference (margin) from the maximum value of the calculated stabilizing yaw moment.

(2) Stabilizing yaw moment variation rate (dM/ds)

The degree of sensitivity relative to the body slip angle of the stabilizing yaw moment in the current car state was hypothesized to affect confidence.

(3) Lower envelope of vertical load on rear outer wheel (Fz)

It was hypothesized that the tire gripping force is unable to track high frequency fluctuations in the wheel load, with the result that only force equivalent to the lower envelope of the wheel load can be exercised.

5.3. Verification of Indices

The premise was that "cars with higher braking stability enable stronger, more stable braking, with the result that section times should be enhanced." Running was performed using various proposed setups thought to provide different stability, the correlations between each physical quantity over the braking section and the section time were analyzed, and the physical quantities with a correlation coefficient of 0.7 or more, which indicates a strong correlation, were extracted from the results. In addition, the braking sections were divided into straight braking sections and turn braking sections, and the correlations were analyzed for each section.

The relationship between the two factors may be nonlinear, so a non-parametric analysis method called Spearman's rank correlation coefficient was used. In addition, a "no correlation" null hypothesis to be rejected was established, and hypothesis testing was performed using a 5% standard to examine whether the calculated correlation coefficients are significant. The correlation coefficient value that could be obtained only at a 5% probability when there was no correlation between the two factors was calculated, and when a correlation coefficient that exceeded this value was calculated, the null hypothesis was rejected and the factors were judged to have a significant correlation (Fig. 25).

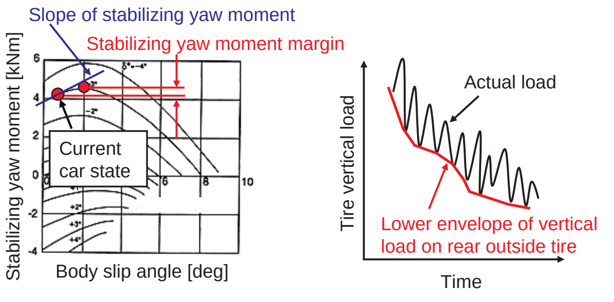


Fig. 24 Candidates for braking stability indices

5.4. Test Methods and Verification Results

Proposed setups with different transitions in the center of down force (CoP) during braking were set. Running tests were conducted on two different circuit tracks to compare a baseline setup (the CoP shifts to the front from the beginning of braking towards the corner apex point) with other setups changed so that the CoP shifts towards the rear, which was presumed to enhance braking stability (Fig. 26).

Figure 27 shows the correlation coefficient analysis results for straight braking sections. Of all the

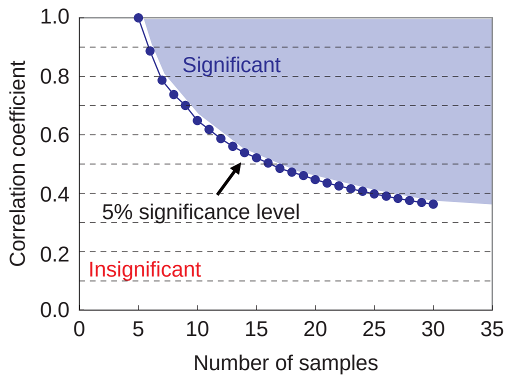


Fig. 25 Significance test for Spearman's rank correlation coefficient

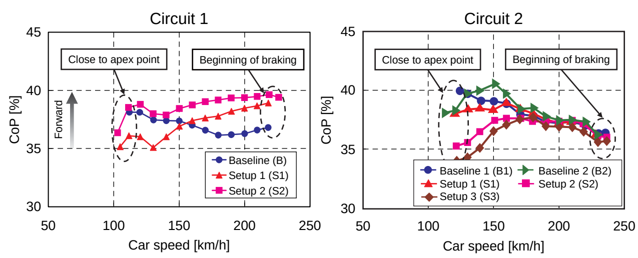


Fig. 26 Setups with different CoP transition in braking

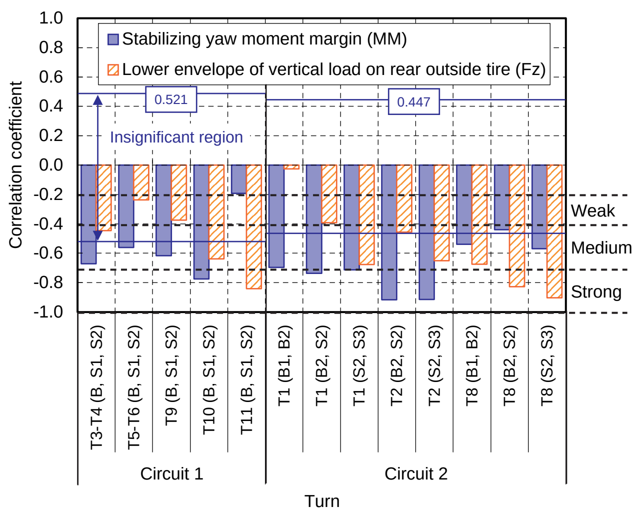


Fig. 27 Correlation analysis in straight braking

candidates, this figure shows the results of only MM and Fz, which exhibited stably high correlations. Of the MM data suitable for analysis, 80.0% of Circuit 1 samples exhibited a significant correlation, and 20.0% showed a strong correlation. For Circuit 2 these figures were 87.5% and 50.0%, respectively. This exceeded the Fz results for Circuit 1 (40.0% significant correlation, 20.0% strong correlation) and Circuit 2 (75% significant correlation, 25% strong correlation), indicating that MM is more suitable as an index in straight braking sections.

In turn braking sections, MM and dM/ds exhibited higher correlations compared to other candidates (Fig. 28). However, these correlations were not as stably high as that of MM in straight braking sections. Here, 33.3% of MM samples for Circuit 1 showed a significant correlation (33.3% strong correlation), and 66.6% (0.00%) for Circuit 2. The respective results for dM/ds were 66.6% (33.3%) for Circuit 1, and 25.0% (8.33%) for Circuit 2.

5.5. Summary of Index Creation

It is necessary to create dynamic performance indices in order to scientifically and quantitatively discuss vehicle design and setup without being dependent on engineer experience and driver senses. Indices were created for braking stability, which is one of the many items that comprises dynamic performance, and knowledge was gained regarding physical quantities that can serve as indices. Indices need also be created successively for other vehicle dynamics items.

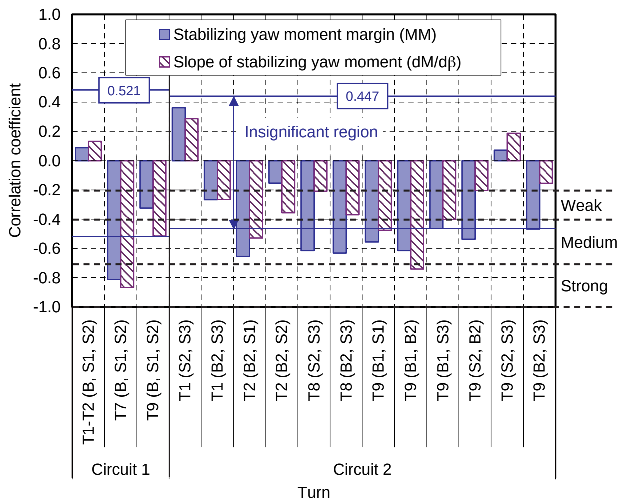


Fig. 28 Correlation analysis in turn-in braking

6. Conclusion

On a qualifying session, the lap time differences between the top teams are usually in within 0.1 s. Assuming a lap time of 100 s, this could be called a competition over differences in performance of approximately 0.1%. Therefore, the demand to have the cutting edge technology for both measurements and analysis is very high. In Formula One, the environmental conditions such as vibration heat and speed are so severe that a very high task is required. These techniques obtained in this project will be useful in passenger car development as well.

Acknowledgements

The authors wish to express their deep thanks to related parties at Tokyo Sokki Kenkyujo Co., Ltd., for their cooperation in developing the Strain Gauged Wheel, Kyowa Electronic Instruments Co., Ltd., for their cooperation in developing the wing load cells, TUV SUD Automotive GmbH for their cooperation in developing tire temperature sensors, and Bridgestone Corporation for their cooperation in developing vehicle dynamics indices.

■ Author ■

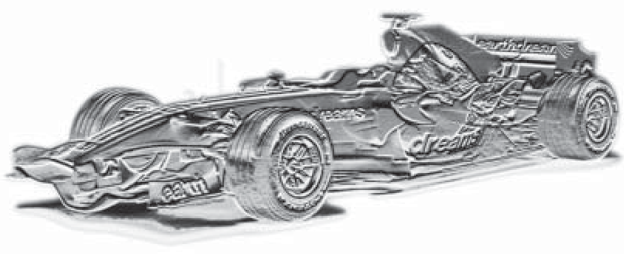


Hideaki SHIBUE Tomokazu SUZUKI Mamoru URAKI



Akihiro IDA Atsushi TSUBOUCHI Yasutaka KITAKI

Descriptions of Vehicle Control Technologies



Development of Traction Control Systems for Formula One

Kazuharu KIDERA* Yoichiro FUKAO* Tatsuya ITO*

ABSTRACT

Efforts have been made toward the development of traction control that controls the amount of tire slip at a level that exceeds the abilities of Formula One drivers and enhances turning acceleration performance, and the development of overrun control that is intended to prevent rear tire lockup by using engine torque when the brakes are being applied. Numerous control systems have been applied in racing based on modern control theories, including high-precision engine speed control and wheel speed feedback control, which is extremely challenging in Formula One, where driving at the limit is the norm.

Launch control, which is control applied in the start of a race, has also realized complete direct control of clutch transfer torque. Development has proceeded on start assist systems that can enhance start performance with reliable repeatability while continuing to satisfy regulations even after prohibition of automatic clutch control. The direct push clutch and other new technologies have been actively adopted.

1. Introduction

Traction control was extremely important to Formula One activities of the third era, which started with the Australian Grand Prix in 2000.

The Federation Internationale de l'Automobile (FIA) had basically prohibited traction control until the Spanish Grand Prix in 2001. The FIA changed its stance at that time, completely lifted the ban, and development of traction control started.

Vehicle dynamic performance continued to evolve even while restrictions were subsequently placed on tire performance. In this context, the optimal control of traction and the maximum employment of tire capabilities occupied crucial positions in the evolution of vehicle dynamic performance.

In 2008, the FIA unified control specifications by making it obligatory to install a common electronic control unit (ECU). Even after control systems that could be used as driver aids were completely eliminated, there was continuing high demand for control systems to make it easier for drivers to manage torque during cornering.

Given these circumstances, development of traction control continued largely throughout the period of third-era activity.

This article describes the history and development technology of various systems, including traction control (TC) in the acceleration range, the development of methods for setting traction that were implemented by throttle pedal input after TC was prohibited, overrun control (OC) in the deceleration range, launch control (LC) that enables fully automatic launching with the press of a single button in the launch range, and the race start system (RS) presupposing manual clutch operation that was developed after 2004 when LC was prohibited.

2. Development Goals

The purpose of traction control in Formula One racing is to assist the drivers so that they can drive the car as fast as possible with repeatability. It is necessary with racing cars in general, however, not only to bring out their performance limits, but also to sustain that unstable state. With mass-production vehicles, it is a requirement of similar control systems that, for safety reasons, they keep the vehicle inside the limit range. This difference in what the two types of system are intended to achieve is also the difference in the performance they are required to produce.

Formula One cars and mass-production vehicles also differ greatly in the hardware that is subjected to control. Formula One cars are, of course, light in weight, in the 600 kg range, and their drivetrains are built to have the lowest possible rotational inertia. Meanwhile, they have engine power, braking power, and tire grip force that are several times greater, as absolute values, than in mass-production vehicles.

For the above two reasons, Formula One cars have

* Automobile R&D Center

much more sophisticated traction control than mass-production vehicles. They demand highly precise, highly responsive systems. Generally speaking, bringing out the performance limits of a vehicle will mean making the most of the performance of its tires, which are the only points of contact with the ground. Control of Formula One power plant systems is no exception in this. While effectively transferring the limited engine power to the tires, it must never exceed the tires' limits. The control methods used can be generally classified into two types: one is the method for directly controlling wheel torque, and the other is the method for controlling the amount of tire slip. Another, indirect method for accomplishing this is to control just the engine torque or the engine speed. Drivetrain components have rotational inertia, backlash, and twist. Strictly speaking, therefore, control by wheel criteria and control by engine criteria are different, and the various methods are employed according to the application, the degree of ease or otherwise of control, and the advantages or disadvantages involved.

Figure 1 shows the history of the three systems introduced below. Figure 2 shows a configuration diagram of the control systems.

2.1. Aims of Traction Control (TC)

The aims of TC in Formula One cars are to control traction at a level surpassing the skill of the driver and to limit the amount of tire slip appropriately so as to enhance straight-line acceleration performance and turning acceleration performance. The TC system is none other than engine power control, but what it is ultimately intended to control is the amount of tire slip.

The degree of challenge represented by control, given the precision required in achieving the desired amount of slip and response, depends on the delay in generating torque by the engine, which serves as the actuator, and the existence of torque transfer delay due to backlash and twist in the component parts of the drivetrain, which is situated between the engine and the tires. TC is further required to assure robust performance with respect to disturbances from rough road surfaces, curbs, and the like.

2.2. Aims of Overrun Control (OC)

OC in Formula One cars is a system for control of engine torque during braking in order to prevent rear tire lockup and to optimize the amount of tire slip.

Active brake balance control and antilock braking system (ABS) implemented by braking systems are forbidden in Formula One racing. However, the enormous tire grip force and the maximum deceleration of as much as 5 G realized by carbon brakes result in a large amount of front and rear load shift, so that the optimum front and rear brake force balance changes from moment to moment. OC does more than simply adjust the engine braking force. When circumstances make it necessary, even during braking, it produces torque all the way to the drive side so that front and rear brake force distribution are optimized while producing performance that, although limited to the rear tires, is intended to be similar to ABS.

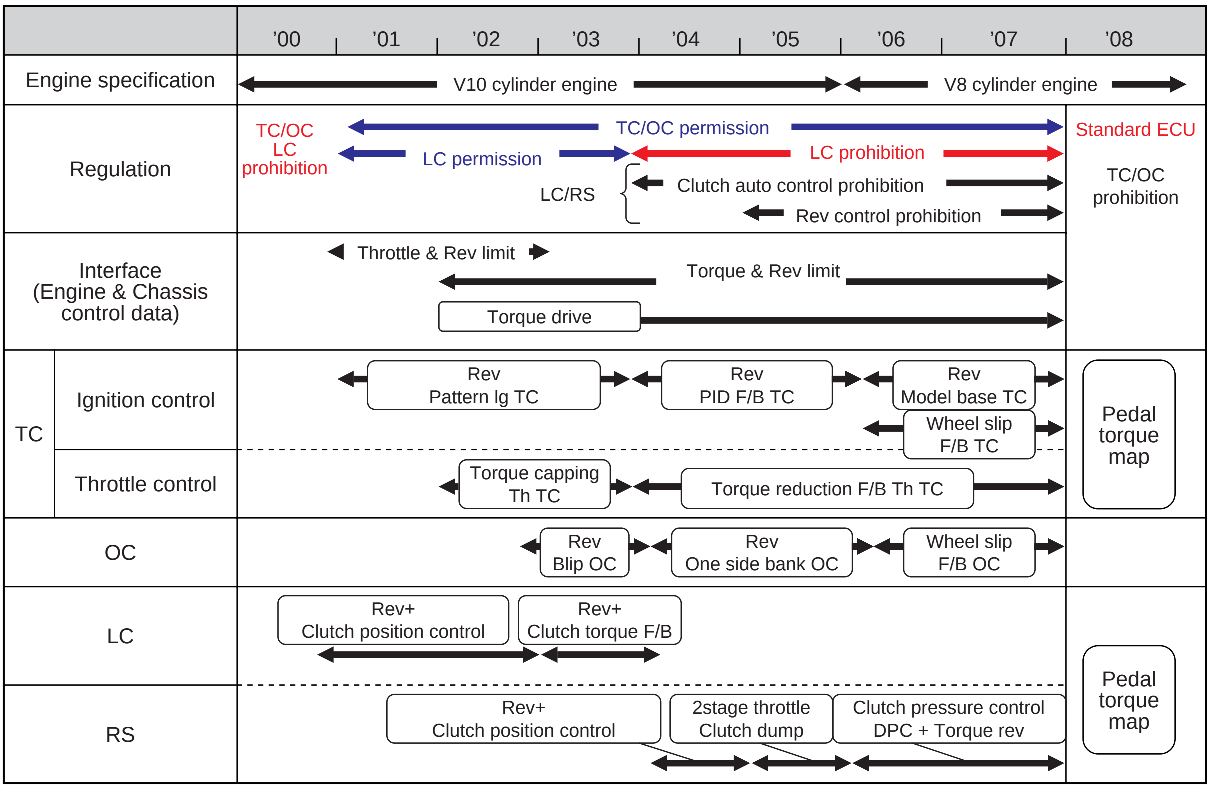


Fig. 1 History of control system development

The result is that drivers can feel safe about pressing down hard on the brake pedal even under conditions with bumps or other such disturbances. When turning in to a corner, the amount of rear tire slip can be limited so as to obtain an oversteering reduction effect.

OC has many of its challenging aspects of control in common with TC. One major difference, however, is that the primary causal source of tire slip is the braking force, while the role played by engine torque is just a fraction of that. Consequently, the margin of control available to OC is necessarily limited.

2.3. Aims of Launch Control (LC) and the Race Start System (RS)

LC is a system that automatically implements Formula One race starting by integrated control of engine torque, engine speed, and clutch engagement amount. If drivers activate the system according to a determined procedure, all that remains is for them to push a single button. LC then appropriately controls the amount of rear tire slip and realizes the maximum acceleration from zero to 100 km/h. (LC was prohibited by Formula One regulations in 2004.)

On the other hand, RS could be termed the driver's manual race start assistance system that followed the prohibition of LC. Development of standing start control within the range permitted by regulations also took place following the prohibition of LC, and this has been a battle of wits among the teams and with the FIA since 2004. The aim of RS is to realize the best standing start performance, with repeatability, by arranging it so that the driver only has to follow a predetermined standing start procedure (throttle pedal operation, clutch paddle operation) to have the proper settings for engine and clutch control be selected by engineers to match with the coefficient of friction μ between the road surface and the tires, as well as with the meteorological conditions.

The two challenges that LC and RS face in common can be summarized as control of clutch transfer torque with a high level of precision and response, and cooperative control of the engine and clutch.

3. Development of Traction Control (TC)

In 2001, the use of TC was explicitly permitted in the regulations, and it was used in racing from that point to 2007. In 2008, TC again came under complete prohibition when the FIA instituted the requirement for a common ECU. This section describes the development of TC together with the measures used to enhance drivability following the prohibition of TC.

3.1. TC Development

TC development for third-era Formula One activities can generally be classified under one or the other of the following two headings:

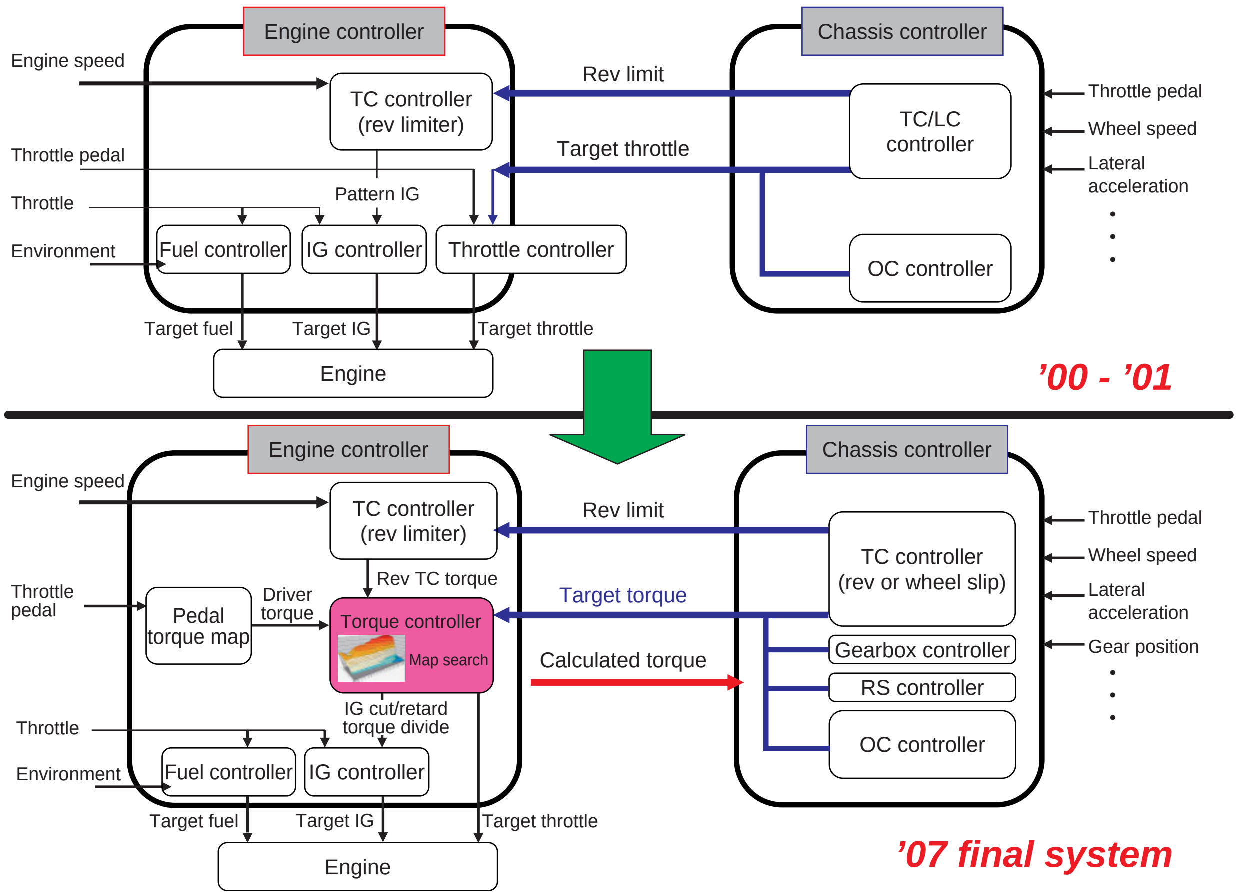


Fig. 2　Configuration of control system

- Trackability of target tire slip
- Determination of target tire slip that yields maximum acceleration performance

At the initial stage of development, the target tire slip was not very trackable, and some instances were observed of factors that disrupted vehicle behavior. Such factors included insufficient toughness with regard to disturbances such as rough road surfaces and curbs, and inadequate control due to insufficient consideration of the transfer properties involved from the engine target torque to the tire traction force. The TC development effort in the beginning and middle periods of the third era sought to resolve these matters primarily by enhancing the trackability of target tire slip, and development work focused on a controller for this purpose.

3.1.1. Enhancement of trackability of target tire slip

Methods for controlling tire slip include direct methods of using actual slip as a feedback parameter for target tire slip, and indirect methods of calculating a target engine speed from the target tire slip and reduction ratio, and then using the engine speed as a feedback (F/B) parameter (Fig. 3).

The engine speed control TC (Rev TC) that had been in use from the start of the third era corresponds to the latter method. The controller of those days used the deviation from the target engine speed as a basis for specifying a predetermined ignition pattern [Pattern Ignition (IG)]. It did not implement control using the engine torque. In the subsequent course of creating various different drivetrain control systems, engine torque was taken as the common parameter, and consolidating it in the torque interface led to enhanced precision of the control system and expansion of the degrees of freedom (introduction of torque drive system).

Rev TC subsequently had the convergence and torque linearity enhanced by the adoption of PID F/B control. The further application of modern control theory that takes engine characteristics into account enhanced convergence still more, while the optimization of the torque control amount also heightened toughness with respect to disturbance.

Figure 4 shows the actual track data for Rev TC. It can be seen how the target tire slip is tracked by controlling the engine speed. Rev TC has the advantages of not being susceptible to system delays, and of fast response. On the other hand, it does not allow independent control using rear left or rear right tire slip, and has disadvantages coping with curbs and the like.

In parallel with development of the Rev TC, development of wheel slip feedback TC (W/S F/B TC) also took place. This is a method of control that uses the tire slip as an F/B parameter. This allows traction control to use only favorable side of rear wheels or both during cornering, and it increases flexibility of the settings. It has a robustness with regard to the input of disturbances from the road surface that was established in advance through simulations, and it also enabled limitation of excessive torque reductions.

Figure 5 shows the actual track data for W/S F/B TC. The wheel torque is subject to finely tuned control with respect to changing load on the rear tires (rear inner and outer tire Fz), and the tire slip is retained in the TC slip target value.

In the third era, the above two control methods (Rev F/B TC and W/S F/B TC) were employed differently according to the circumstances. Rev TC was mainly employed with low speed gear (in racing starts and when building up speed coming out of a low speed corner) or on wet road surfaces, when the traction is large against reaction force from road surface and the F/B system requires high response and high resolution. W/S F/B TC, which allows more flexible control, was employed mainly for driving in middle and high speed gears and under conditions when disturbances such as curbs could have an influence.

Important points of TC F/B systems include not only the consideration of system delay and robustness with respect to disturbances, but also the fact that, in the interest of drivability, it limits excessive ignition control fluctuations that ignore the will of the driver, and that, in the interest of fuel economy, it limits the amount of fuel injected. Throttle TC (Th TC) implements control

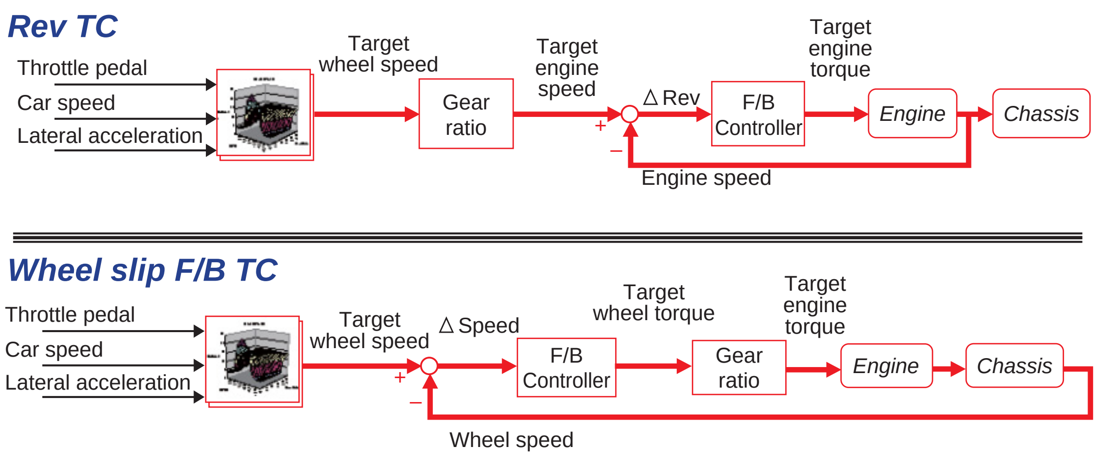


Fig. 3 Feedback control system in TC

for the above purposes. It continued to make enhancements following from Torque capping Th TC, which was close to the sequence control employed in initial development, and ultimately was able to employ control by a torque reduction F/B system that realized a fuel economy increase of approximately 3% and ignition control that provides a stable, small amount of fuel injection even during the input of disturbances.

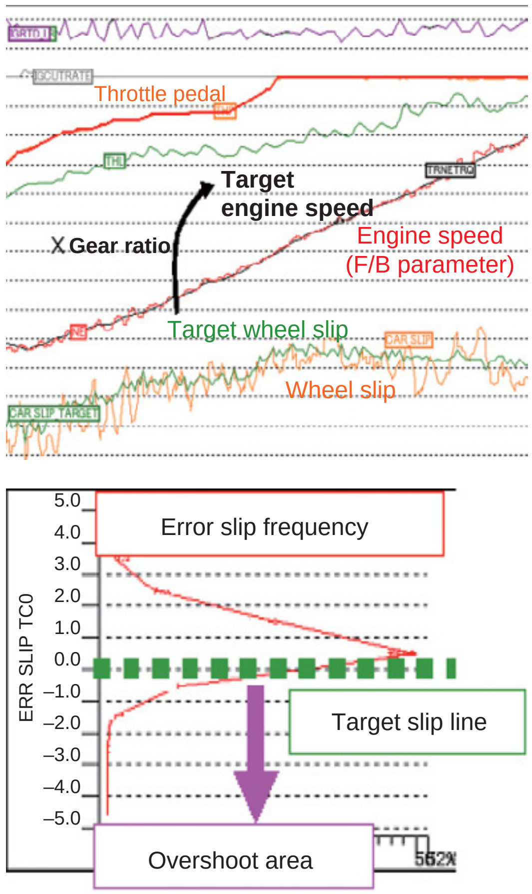


Fig. 4 Rev TC track data

3.1.2. Calculation of target tire slip

Enhancement of feedback control systems came to assure stable trackability of target slip, and TC development shifted to ascertaining optimal target slip. Starting in the summer of 2006, attempts were made to use the car model and tire model in the ECU together with data from the various types of sensors to estimate the vehicle state quantities, and then to calculate target slip through an integrated control that also included diff control (electronic limited-slip differential gear control). The method that was finally adopted, however, was to store multiple maps of target slip based on driver throttle pedal positions and vehicle lateral G forces.

The above methods adopted from the beginning of the third era had the advantage of enabling settings that responded to the feelings of drivers and reflected their will. On the other hand, it is crucial to be able to determine the settings that are appropriate for the corners, road surface, and tire conditions of each individual circuit, and that further support the skills and predilections of the driver. The challenges with respect to the time and labor involved and the method's adaptability were never fully resolved.

3.2. Development Following TC Prohibition

The change of regulations in 2008 placed control of overall vehicle systems, including the engine and gearbox, in the ECU manufactured by McLaren Electronic Systems, which would become the common FIA ECU. TC and OC were prohibited. As a result, lack of traction during cornering and poor drivability emerged as issues. It therefore became necessary to enhance both hardware and software aspects of the systems.

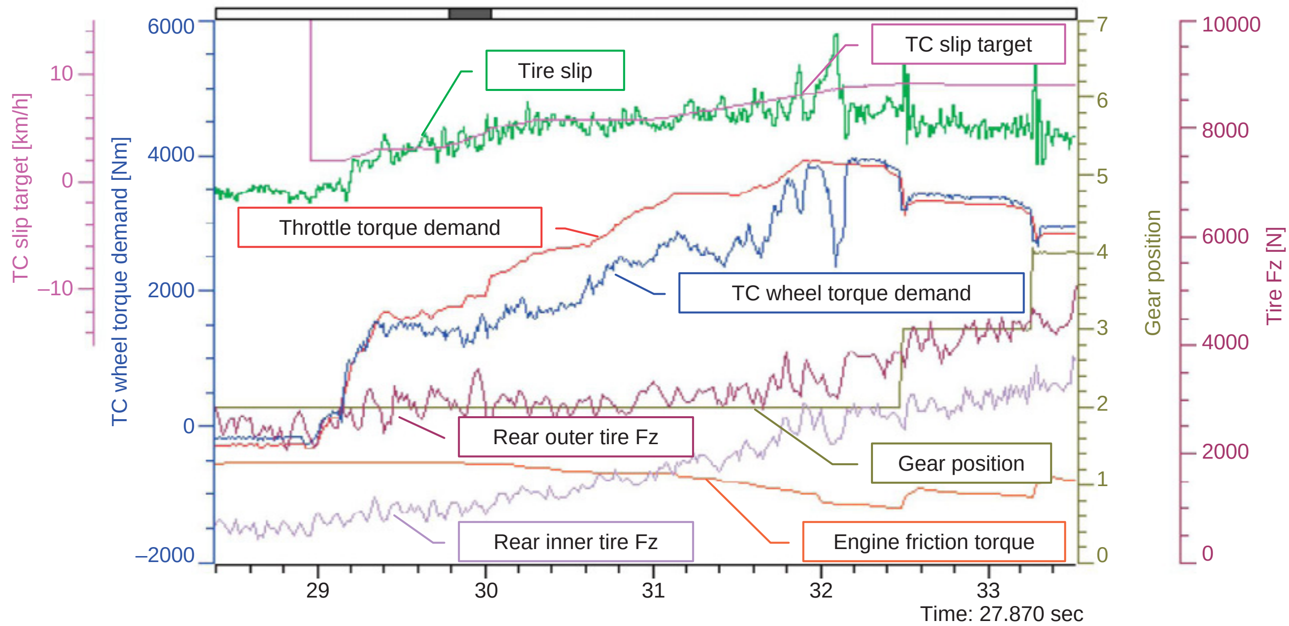


Fig. 5 Wheel slip feedback TC track data

Development of powertrain systems proceeded with a focus on torque accuracy. The reason for doing so was that when TC was in effect, torque values other than high magnitude torque errors were absorbed by the feedback system so that weaknesses tended not to come out in the open. The prohibition of TC, however, heightened the requirement for torque accuracy, including transient engine torque characteristics.

Furthermore, the method for defining the target torque against the throttle pedal could not depend only on driver feeling. It became necessary to establish setup methods to substitute for TC.

3.2.1. Torque accuracy

Important points for the powertrain from the perspective of drivability include:

(1) Ability to predict the driving torque of throttle pedal operation by the driver;
(2) Linear torque characteristics of pedal operation; and
(3) High torque response.

Maintaining the vehicle stability limit during a turn when any one of these factors is missing becomes a challenge. With regard to items (2) and (3), solutions are likely to differ according to driver characteristics and predilections.

Engine development sought to enhance the air intake and the exhaust and fuel systems. This yielded enhancement of combustion stability under low load (under rich air/fuel ratio conditions), limited variation in transient throttle opening and closing, and promoted enhancement of torque response and torque repeatability. Map measurements of various kinds made with a view to combustion stability as well as data settings made with consideration of usage conditions on the circuit also contributed to increased torque accuracy and enhanced drivability.

3.2.2. Definition of target torque

With feedback control for TC being prohibited, the setting of target torque for the throttle pedal becomes an item with significant effects on drivability. Drivers operate the throttle pedal to control the amount of tire slip when coming back up to speed out of a corner, and one thing that is crucial at this point, in addition to assuring the resolution and linearity of the driving torque against throttle pedal operation, is implementing torque reduction when excessive slipping occurs. During the 2008 season, the settings were optimized in light of the traction utilization ranges on the low-speed corners at every circuit in order to realize all of the factors noted above. At that time, circuit simulations were also carried out in advance in order to heighten the efficiency of the process leading to optimization.

3.3. Establishment of Evaluation Items and Test Bench Evaluation Methods

The ability to quantitatively analyze the influences that power plant torque behavior and vehicle settings have on TC controllability and drivability is a matter of importance in performing every kind of development. Evaluation items were therefore examined together with methods of test bench evaluation.

Since the project was frozen in mid-development, some parts of activities such as evaluation of the stability of a vehicle as a whole that relied on driver comments still remained. In powertrain evaluation, however, and particularly in evaluation of torque response characteristics and accuracy, quantitative benchmarks and test methods can be established. Benchmarks and methods can be checked in advance before running on the circuit, and front-loaded development can now be implemented.

4. Development of Overrun Control (OC)

4.1. Engine Rev Control OC

The development of OC began at the suggestion of a Honda team member in the autumn of 2001, during the BAR-Honda period. It became a full-fledged development item in 2002, and progressed to the point of being adopted for use in racing.

This was initially a system to control the handover of the lower engine speed limit (the opposite of the rev limit), in accordance with the feed forward throttle position demand and tire slip limit, from the chassis control unit (Pi-Sigma MCU) to the engine control unit (ATHENA ECU). From the autumn of 2002, as with TC, the throttle position demand handed over between the MCU and ECU was replaced by torque demand. Subsequently, there were requests from the drivers for engine braking and antilock performance, which are mutually contradictory, and in 2003, throttle torque demand was switched over to an on-demand system (Blip OC) that responded to engine speed deviation.

Up to the start of 2006, however, the drivers complained of a pushing feeling and other issues in the OC caused by a phase delay in the control. Its use was therefore limited. Changes went on being made in methods of setting target values, and further tuning continued, but the rear tire antilocking performance that was the real purpose of OC remained incomplete.

4.2. Wheel Slip Feedback OC

Wheel slip feedback TC was adopted for racing at the beginning of 2006, and after several races, a completely identical control algorithm was also applied to OC. This was not conventional engine speed control, but instead allowed the use of rear wheel speed as a source for direct feedback. Consequently, it allowed fast, finely calibrated control of engine torque with respect to disturbances from the road surface, fluctuations in brake torque, and other such factors.

Figure 6 shows a view of control in OC. The OC wheel torque demand will have risen and prevented lockup of the rear tires long before the tire slip amount reaches the limit slip amount (OC slip target), and when the slip comes under control, the torque goes down (each torque value is benchmarked to the wheel torque). The throttle is opened according to the OC torque demand with an added torque margin. This enhanced the antilock performance and received favorable comment from the drivers.

This new OC was based on wheel torque control, and therefore was easily capable of adjusting the engine braking torque simply by feed forward control. Since the necessary amount of engine braking differs according to the driving style, it is basically intended to provide an amount suited to the driver's predilections. However, considering that it tends not to be affected by changes in the gear ratio, as well as its utility in eliminating stepped changes in torque during gear shifts, it was decided to implement control by a map of wheel torque demand relative to car speed. The value given to simplicity and ease of tuning in the Formula One context was another reason for this choice. The data from the 41-second point to the vicinity of the 41.4-second point in Fig. 6 shows this OC feed forward torque demand in operation.

During braking, the engine would fire at a higher frequency, so that the rise in exhaust temperature in the exhaust pipes was initially an issue in development. This was addressed by firing only one bank when torque was adequate to the demand, and using the left and right banks in alternation every time engine braking was used.

The use of an FIA-common ECU became obligatory in the regulations starting in 2008, and TC and OC were prohibited. Driver complaints, however, centered more on their being unable to use OC than TC. This is because controlling engine torque during braking of the kind applied in OC is challenging even with the skills possessed by Formula One drivers.

5. Development of Launch Control (LC) and the Race Start System (RS)

As of the end of 2008, it was thought that the RS control method using the FIA-common ECU would continue to be used by ex-Honda Racing Formula One Team even in 2009 and beyond. The discussion here, therefore, will be limited to the LC of 2003 and earlier, and the RS used after it up to 2007.

5.1. Clutch Position Control LC

The development of LC started in 2001, the same year that the ban on TC was lifted. Up to 2002, one issue faced in LC was the controllability of the initial tire slip from immediately after the standing start (launch) until the clutch is fully engaged. This was because position control of the pull clutch of that time resulted in large changes in clamp load due to the amount of disk wear and thermal expansion that would occur even if the clutch were held at the same position in a partially engaged state. The changing clamp load meant that the transfer torque would become unstable. Originally, such changing factors should have been absorbed by the tire slip feedback control portion of the system, but the processing cycle of systems at that time was too slow to provide adequate compensation capacity.

5.2. Clutch Torque Control LC

In 2003, in order to resolve the above issues, a non-contact magnetostrictive torque sensor was installed on the input shaft located between the clutch and the gearbox, and a system for direct feedback control of clutch transfer torque was developed. This clutch torque control had practically no delay compared with TC controlling engines that experience a torque generation delay time of 10 msec or more. It therefore provided an edge in tire slip control.

The realization of direct control of transfer torque brought the capability to produce feed forward torque matched to the coefficient of friction μ between the road surface and the tire at the starting instant. Once the vehicle started moving, it could switch over to slip feedback control for the rear tires, and the amount of tire slip could be controlled so as to maximize acceleration. In order to accomplish this switchover as quickly as possible, a dedicated arithmetic processing unit was

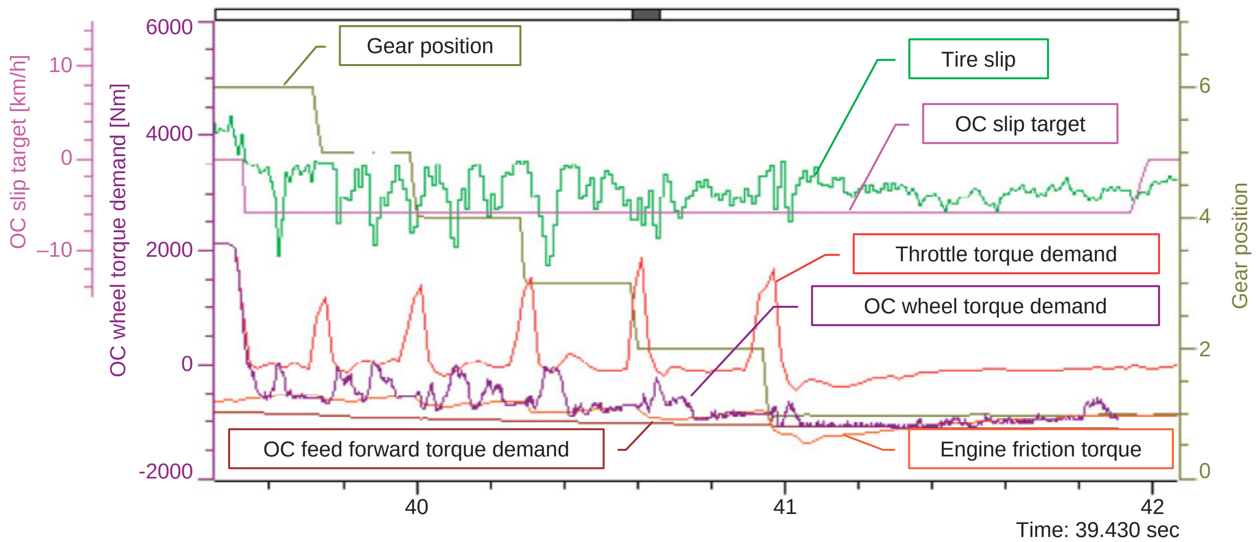


Fig. 6 Wheel slip feedback OC track data

installed to detect extremely low wheel speeds. This was done because slip control of tires when they are beginning to roll was a major contributing factor to victory from the start of a race.

With LC, the speed of the driver's reflexes also makes a significant contribution. Actual measurements were therefore made of the brake release system, button operation, paddle operation, and other such systems for each driver, and the pattern with the smallest delay was adopted. In the case of an outstanding driver, there is a reaction time of less than 0.2 second from seeing the starting signal to taking action on the controls. If the driver fails to start, then a Formula One car with acceleration performance taking it from zero to 100 km/h in under three seconds does not, in practical terms, allow for recovery even with an outstanding control system.

Another factor that determines the results of a Formula One race start is tire temperature management. The traction performance of a tire becomes optimal in the vicinity of 80-90°C, as no doubt every team was aware, and very few teams were able to manage this in an actual race start at that time. BAR-Honda finally managed to apply a tire surface temperature sensor in racing from the time of the British Grand Prix in 2003. This showed for the first time how abnormally low their tire temperatures were on the starting grid, and the team was having to play catch-up on this and related operational matters. When road surface conditions were good and tire temperatures were optimal, LC had a very real punch that could provide acceleration from zero to 100 km/h in under 2.6 seconds, and BAR-Honda was probably ahead of the others in this regard. There were other teams, however, that took the lead with a rearward weight distribution package, which was advantageous in starts, or with their knowhow on using tires, and it was regrettable that the BAR-Honda team's starting performance did not raise the team's overall strength into the top rank.

5.3. Partial Clutch Engagement RS with Pull Clutch Position Control

In 2004, the FIA entered on a program of eliminating driver aid systems, starting with the immediate prohibition of LC. The regulations required clutch control to move the clutch to a target position, or to exert a target pressure on the clutch, determined by the driver's manual operation. It was too great a challenge to explicitly state the position with regard to TC in the regulations, and so TC operation following full clutch engagement was still permitted. BAR-Honda sought to simplify the driver's clutch operations by devising a special clutch paddle map, as shown in Fig. 7, and combining it with a double paddle system in which clutch paddles were attached on either side of the reverse face of the steering wheel, as shown in Fig. 8, so that they would be in standby at different positions and would be released at different times.

The paddle on the disengagement side of the two paddles would serve as the actual paddle demand value. Therefore, the portions marked with heavier lines in Fig. 8 are the actual clutch position demands. According to this procedure, the driver shifts from the fully disengaged state to that of partial clutch engagement, and then to full engagement, enabling a sequence of operations that could be carried out rapidly and accurately. In a state of partial clutch engagement, there will be a fixed target torque in accordance with the horizontal parts of the paddle map. This makes use of the fact that at the extremely low speed in the initial phase of acceleration, the maximum driving torque of a Formula One car is close to being a certain fixed value (Fig. 10).

The ECU was equipped in advance with various internal paddle maps on which the horizontal portions represented several different torque target values. The final selection of the map to be used for the actual race start was determined several tens of seconds before the start by a control engineer who was standing by in the pit, and the driver was instructed by radio. It was done this way because the start was practiced on the actual grid at the beginning of the formation lap, and the coefficient of friction μ between the road surface and the tires was estimated from telemetry data.

Regardless of ingenious measures of this kind, the system of that time, whereby a pull clutch was subjected to position control, did not manage to realize transfer torque for the launch as intended. This situation persisted, caused by the state of disk wear, changing temperatures, and the like, much as was the case with the issues faced by LC up to 2002.

5.4. Clutch Dump RS

A super-short first gear was introduced and RS was switched over to a clutch dump procedure, starting from the Japanese Grand Prix in the closing stage of 2004,

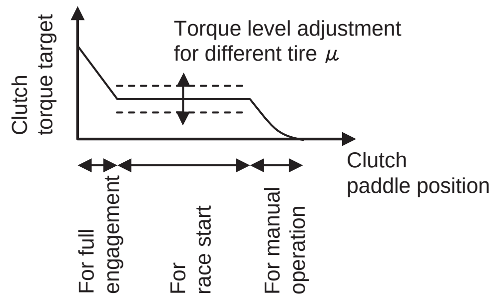


Fig. 7 Clutch paddle map

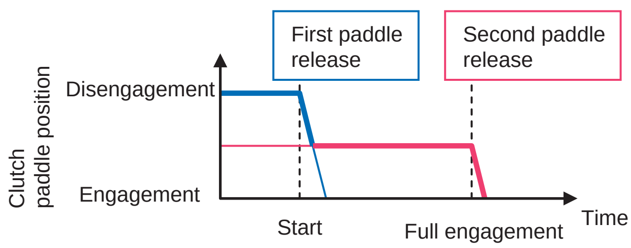


Fig. 8 Double paddle procedure

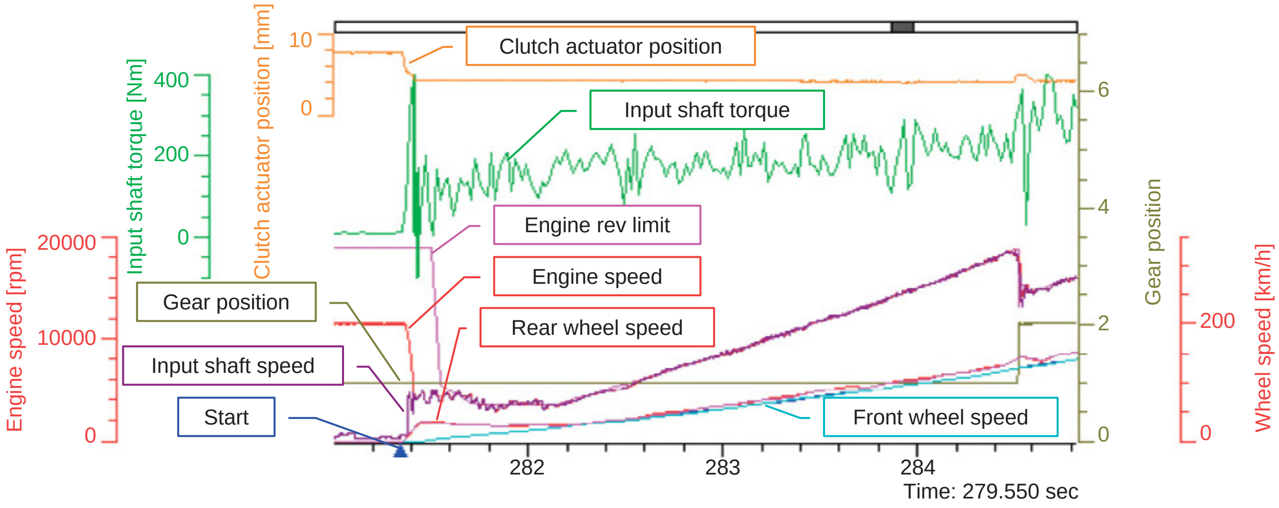


Fig. 9 Clutch dump RS track data (2005)

in order to resolve the weakness of pull clutch position control. This method involved holding the engine in ready state at the rather high speed of 11000 rpm or more, fully engaging the clutch when the clutch paddle was released immediately after racing starts, and thus forcing the wheel spin to be reduced by applying the engine rev limitation for extremely low speed, 5000 rpm or less. Figure 9 shows actual track data for clutch dump RS as it was in 2005.

Although the wheel spin prevents the initial acceleration from rising as high as when a partial clutch engagement start succeeds and there is no wheel spin, this method realized highly repeatable starts because the amount of wheel spin generated was the same every time. The only indeterminate factors in starting were the timing for release of the clutch paddle and the tire temperatures. There were primarily two challenging aspects of control. One was that the engine speed had to be kept at a low speed, even lower than its idling speed, even though for just one second or less, and satisfactory ignition conditions and hydraulic pressure had to be assured during that time. The other aspect was that the regulations specified that the target clutch position (or the hydraulic pressure) determined by the clutch paddle and the actual clutch position (or the hydraulic pressure) are allowed no more than a 50 msec delay. If full engagement occurs instantaneously according to the paddle demand, the shock torque generated in the drivetrain would have easily exceeded the maximum allowable torque for gearboxes of that time. That was why the clutch control was tuned with great care with the order of several milliseconds so that the maximum transfer torque could be reduced as much as possible within the allowed 50 msec time period.

There were no regulations relating to engine rev limit control in 2004, and drivers were permitted to keep their throttles wide open even before the start. In 2005, however, use of the rev limit while in the standby state was prohibited. BAR-Honda proposed an engine torque trimming map exclusively for use in starting and a two-stage method of pressing the throttle pedal, and continued the use of dump RS (Fig. 9) that realized controllability on a par with the real rev limit from the standby state to the standing start while seeking to simplify driver operations.

5.5. Partial Clutch Engagement RS with DPC Pressure Control

In 2006, the 3-L V10 engine used until then was prohibited, and the use of 2.4-L V8 engines was required. This meant that engine torque was diminished by 20% or more, so that the clutch dump system could effectively no longer be used. Figure 10 shows the relationship between torque in a V8 engine and the required torque calculated from the coefficient of friction μ between the road surface and the tires. The μ on a dry road surface ranges from 1.6 to about 1.9, so that in order to satisfy the relationship of Engine Torque > RS Required Torque, an engine speed of from 8000 to 9000 rpm or more is necessary. With the V10 engine, the relationship between required torque and engine torque was such that μ = 1.0, as shown in Fig. 10, allowing the dump system at 5000 rpm or slower. However, the V8 engine was required to provide torque at higher

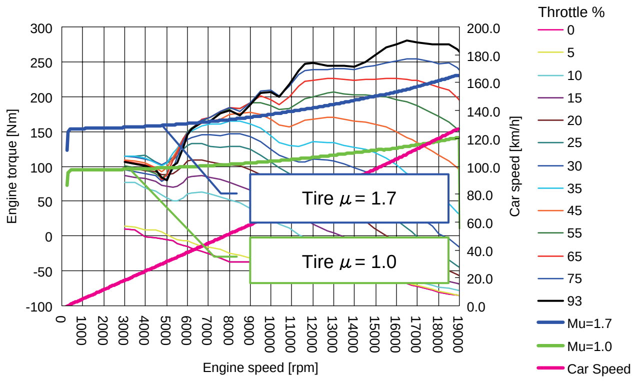


Fig. 10 V8 engine torque and required torque for RS

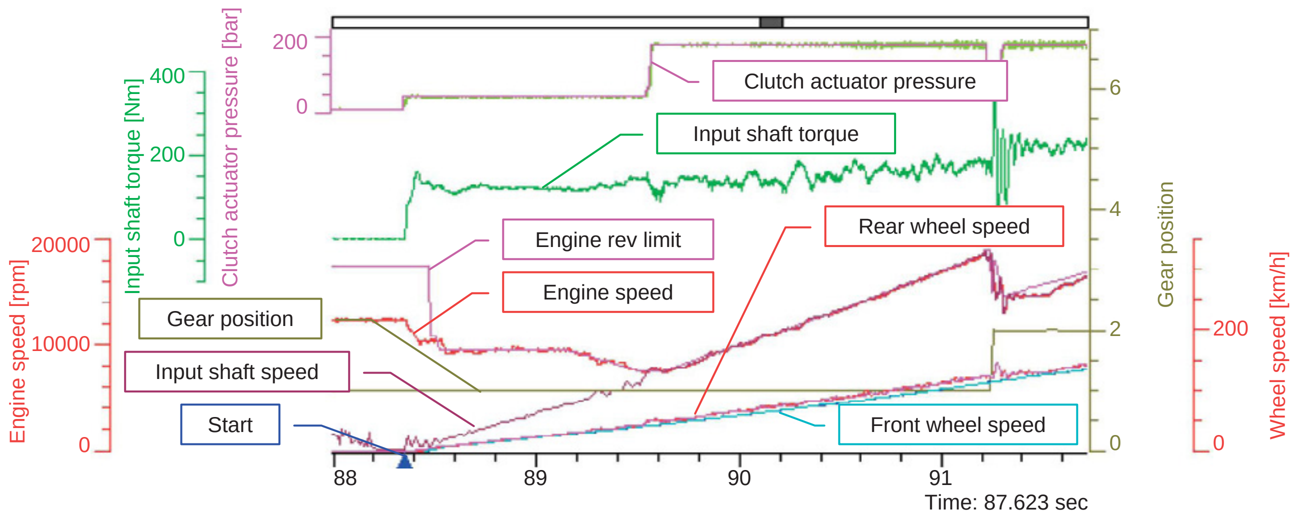


Fig. 11 Clutch partial engagement RS with DPC track data (2006)

engine speeds, also making it necessary to alter the RS strategy on the assumption that the double paddle system with partial clutch engagement would be back in use.

Honda’s unique, newly developed direct push clutch (DPC)[(1)] was adopted as a measure to enhance clutch torque controllability. This clutch resolved the issue of thermal expansion and wear state of the disk causing the clamp load to change, which was a weakness of the pull clutch. DPC used a system with a hydraulic actuator that pushed directly against the disk for the clutch connection. This succeeded in obtaining clamp load with generally linear characteristics with respect to the hydraulic pressure, and the transfer torque was stable. The clamp load was already known, so where race operations was estimating the tire μ as described earlier, now it also estimated the clutch disk μ, further enhancing the precision of paddle map selection in racing starts.

Torque/rev control was also introduced to combine torque control with the conventional rev control method of engine control. In this method, control of the absolute value of torque using the engine formed the primary means for control, and it would switch over to rev control only when the engine speed reached a specified upper or lower limit. The purposes were to stabilize clutch transfer torque by optimizing engine torque output, and to create conditions facilitating recovery from the wheel spin by holding torque at a minimum even when wheel spin occurred.

Figure 11 shows actual track data for partial clutch engagement RS by means of DPC pressure control. The double paddle method keeps DPC pressure (clutch actuator pressure) demand at a certain level, thus holding the transfer torque (input shaft torque) at a constant value as close as possible to an ideal value in the state of partial clutch engagement. This realizes an ideal standing start without causing wheel spin.

The development of this system up to 2007 generated knowhow that was applied wherever possible in the area of tuning in the FIA-common ECU that was made a requirement in 2008.

6. Formula One Control System Development Environment

When third-era Formula One activities got started, the focus was initially on testing in the actual vehicle, and the software development for control systems was also done by hand coding, working from specifications based on flowcharts. However, the project for development of Honda made chassis control systems, begun in 2002 to enhance development efficiency, was the start of a gradual shift to development based on the use of MATLAB Simulink and Stateflow, and from that point on, technology for autocoding was actively incorporated into the development effort. Development of software for control on the chassis side was gradually converted to autocoding with the LC in 2003, and in 2006, development of the chassis-side control unit was switched from Pi-Sigma MCU, which had been used up to that time, to the ATHENA system, which was made by Honda and used for engine control. With this, development on the chassis side was converted 100% to autocoding. Autocoding technology was incorporated on the engine side, as well, but ATHENA had already been employed continuously since 2000, and for that and other reasons, the use of autocoding did not progress beyond partial adoption for use on relatively new systems.

There were also advantages to the use of Simulink and Stateflow. Offline simulation was performed across a wide range on both the chassis and engine sides, from the debugging level to the development of new control algorithms, and the Hardware In the Loop Simulation (HILS) environment, which included a real ECU, was also actively employed. In fact, the LC and RS introduced from 2003, as well as the wheel slip feedback TC and OC introduced in 2006, were developed by simulation using a chassis model that included a drivetrain.

The development of third-era Formula One control systems sought to successfully develop systems with limited opportunities for testing in actual vehicles and to deploy them as quickly as possible in racing machines. Simulation technology can be said to have played a central role in this effort.

7. Conclusion

Throughout the third-era Formula One activities, there was an impetus to move beyond traction control that dealt simply with acceleration, and to incorporate traction control for limit ranges in a variety of different circumstances including deceleration and standing starts. This yielded results and many new insights in connection with (1) the adoption of torque drive system that converts throttle pedal input to engine torque demand, (2) the adoption of a torque interface with chassis control systems, (3) wheel slip control that directly controls wheel slip, (4) coordination with clutch control, and so on.

All this brought a keen awareness that not only the powertrain field, naturally enough, but also the chassis field and coordination with the driver were particularly important for traction control. It also became apparent that alternative testing approaches, to include virtual methods, were of importance in developing control systems for use under limit conditions that are not readily reproducible in the actual machine. It is to be hoped that the experience gained on these two points will also be put to active use in developing mass-production models.

Reference

(1) Kishi, T., Nagatoshi, Y., Nakamura, H., Fukao, Y.: Development of Direct Push Clutch Control during Honda Formula One Third Era, Honda R&D Technical Review 2009, F1 Special (The Third Era Activities), p. 207-210

■ Author ■

Kazuharu KIDERA

Yoichiro FUKAO

Tatsuya ITO

Development of Gearbox Control during Honda Formula One Third Era

Takayuki KISHI* Yoshiki NAGATOSHI*
Hiromasa NAKAMURA* Yoichiro FUKAO*

ABSTRACT

One main development topic for the Honda Formula One third era gearbox was the seamless gearbox (Quick Shift), which reduced the interruption of acceleration to zero.

Gearshift control provided a way to control dog damage and excessive torque, which had been issues affecting Quick Shift feasibility, and this enabled the team to realize first deployment of the Quick Shift in racing, ahead of the other teams. The search for even smoother gear shifting continued after that, and led to the establishment of a mechanism that would reduce to a minimum the torque fluctuation resulting from correlation of the torsional vibration of the driveshaft and the inertial torque generated during gear shifts.

1. Introduction

The structure of the gearbox in the Honda Formula One third era was the seven-gear sequential gearshift of the constant mesh type widely used on motorcycles. This was electronically controlled using a hydraulic pressure actuator, and gearshift operations were carried out using a paddle attached to the steering wheel. Given this structure, shifting gears requires the engine torque to be first reduced to nearly zero. Reducing this gearshift time to the ultimate extent would contribute to increased racing competitiveness, and development of such control was therefore begun. The development effort sought, on the upshift side, reduction of acceleration loss, stabilization of wheel torque, and limitation of torque oscillation when shifting gears. On the downshift side, it sought reduction of minus torque when shifting gears, and reduction of idle running time. In this way, measures were taken to achieve a balance of hardware evolution combined with performance enhancement and establishment of reliability.

The Quick Shift mechanism, development of which started in 2004, was an innovative mechanism that enabled upshifting while still maintaining engine torque at the maximum level. It brought about a major evolution in performance when shifting gears. However, it was necessary to develop new controls to take advantage of this mechanism. Even before the introduction of the Quick Shift, the demand for better gearshift performance had been addressed by enhancing control equipment and changing the controlled subjects. With the seamless gearshift, however, there was a likelihood that significant damage will be made to the dog or the gears depending on the gearshift timing. Control that would eliminate this likelihood and assure a prescribed level of reliability was an absolute condition for the feasibility of this new mechanism. This paper will introduce the main controls that established the reliability of the Quick Shift and the control methods that realized smooth upshifting and downshifting.

* Automobile R&D Center

2. Development Goals

2.1. Establishment of Quick Shift Reliability

The Quick Shift is a mechanism that enables upshifting without lowering the engine torque. However, when one dog collided with another edge to edge, the shock was significant, and sometimes the dog edge would wear so that it could no longer provide driving power. If the next gear provided driving power when there was dog delta speed, an impact torque was also generated that resulted in the gear exceeding its torque limit so that the gear was damaged. The following two points were therefore defined as control objectives:

(1) Zero collision of one dog with another edge to edge
(2) Zero occurrence of gear over-torque

2.2. Realization of Smooth Upshifting and Downshifting

The inertial torque that occurred in gear shifting when shifting up or down could result in excessive wheel slip, rear tire lock-up, deterioration in drivability due to torque oscillation after gearshifts, and other such issues. The following objectives were defined to address these issues:

(1) Upshifting
- Maintain wheel driving torque
- Limit oscillations in engine speed and in torque after gearshifts

(2) Downshifting
- Limit minus torque after gearshifts
- Limit oscillations in engine speed and in torque after gearshifts

3. System Overview

Figure 1 shows the gearbox system configuration. The gear position is determined by the angle of the gearshift barrel, and the gear shifting is executed by input on the gearshift switch attached to the steering wheel. While shifting gears, the system controls the gear position and simultaneously requests the clutch transfer torque, engine speed, and engine torque.

The clutch operates according to analog input from the clutch paddle attached to the steering wheel. In addition to engaging and disengaging, the amount of clutch disengagement is also controlled according to gearshift and anti-stall requests.

The gearshift and clutch actuators are driven by 200-bar hydraulic pressure controlled by a Moog valve. (The control cycle is 1 ms.)

4. Establishment of Quick Shift Reliability

4.1. Avoiding Edge to Edge Collision of One Dog with Another

An overview of the control should include the development of Trigger Shift control. The Trigger Shift control uses rotation sensors for the drive shaft (layshaft), which is connected directly to the engine, and for the driven shaft (main shaft) to constantly calculate the angle of the gear and of the dog ring, memorize the phase difference between the dog ring and the gear when the gear is engaged and driven, and calculate the timing for driving the gearshift barrel so that it can occur in the interval between dogs at the next gearshift. Figure 2 shows a conceptual image of the target position.

The Trigger Shift learns the phase differences in the below order from (1) to (4), and drives the shift barrel accordingly.

(1) The rotation angle of the dog ring is calculated from the main shaft rotation.

(2) The rotation angles of gears 1 to 7 are calculated from the layshaft rotation.

(3) The angle difference between (1) and (2) (phase difference (1)-(2)) for each gear when engaged with the dog ring is calculated and memorized.

(4) The angle up to the point where the target phase difference is reached is predicted from the barrel movement time and the dog delta speed, and the barrel is driven accordingly to shift gears.

The above describes the basic logic involved.

Figure 3 shows a block diagram of the Trigger Shift control.

The following two data settings are necessary to implement this control:

(1) Response time until the barrel reaches the in-gear position

(2) Target position between dogs (upshift and downshift)

These values are determined by the structure, and they do not require fine tuning while the car is actually in operation. The control is efficient also in helping to ensure reliability by reducing the likelihood of operational errors during racing, and the like.

Figure 4 shows example Trigger Shift race results in graph form from the ninth competition in 2006, the Canada Grand Prix. The graph summarizes the in-gear positions between dogs when shifting up from 3rd to 4th gear. The triangle symbols (▲) in the figure represent the positions targeted at by the Trigger Shift system. The open circle symbols (◯) in the figure represent the actual in-gear positions. The figure indicates that the system generally functioned with in-gear positions as targeted

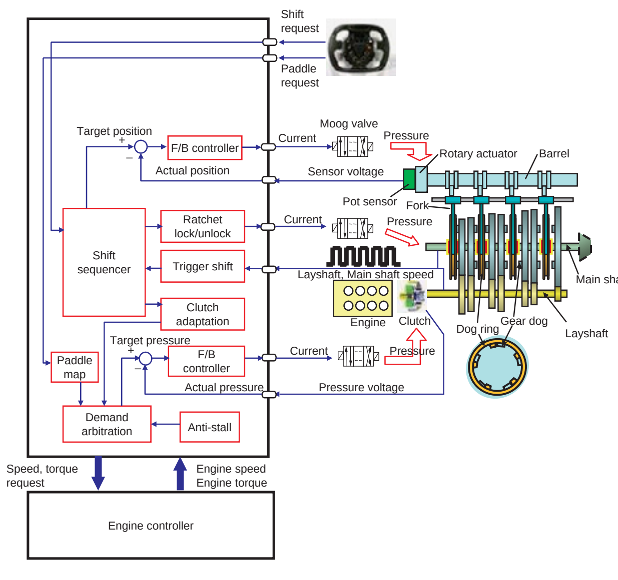


Fig. 1 System configuration

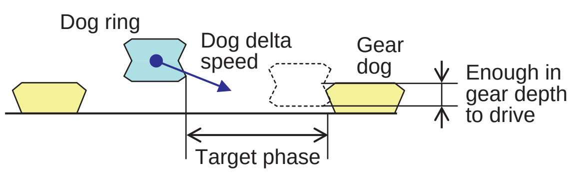


Fig. 2 Image of trigger shift

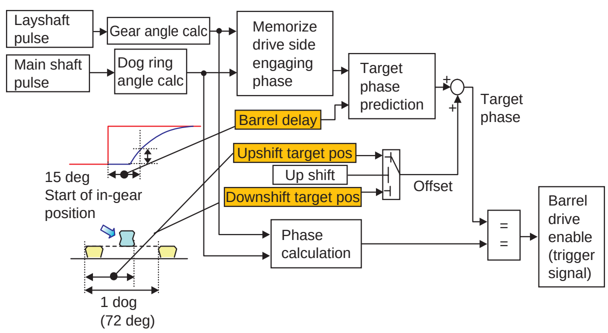


Fig. 3 Trigger shift control system

for, and did so repeatedly with good reproducibility.

The Trigger Shift performance limits will also be described here. If the barrel response is taken to be constant, then the dogs engage with a shallower intermeshing when the dog delta speed is greater, so that the dogs are more likely to collide edge to edge with each other. Furthermore, even when the intermeshing depth is the same, there will be greater damage to the dog edge when the dog delta speed is greater. Figure 5 shows the results of testing on wear limits by intermeshing gear depth and dog delta speed. The maximum value for dog delta speed between dogs is determined from the upshift speeds and the gear ratios in Formula One racing to be approximately 2000 rpm. The intermeshing gear depth necessary to reduce the occurrence of wear in this case, as shown in Fig. 5, is 1.5 mm. By compiling fork speeds in all the 2006 races, it was found that the four sigma point of the distribution was 1.54 mm/ms, and this was taken as the worst case. Given this, the in-gear position capable of avoiding dog-to-dog collision could be expressed in terms of time as 1.5/1.54 ≈ 1 ms, which is the necessary safety margin when the dog delta speed is 2000 rpm.

In actuality, as shown in Fig. 6, it is necessary to take into account variations in the target position due to barrel response times and dog delta speed changes. Taking these factors into consideration, together with the fact that the dogs are placed every 72 degrees, the limit dog delta speed at which the Trigger Shift can assure performance is expressed by formula (1):

(Dog pitch) - (Dog delta speed variation) / (Control cycle error) + (Barrel speed error) + (Safety margin) (1)

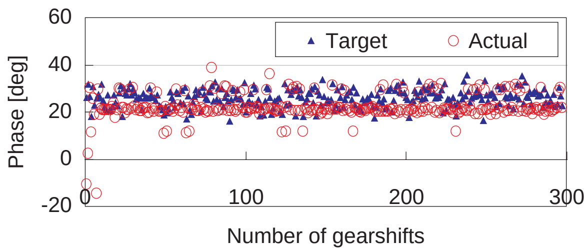


Fig. 4 Trigger shift result (2006 Canada GP)

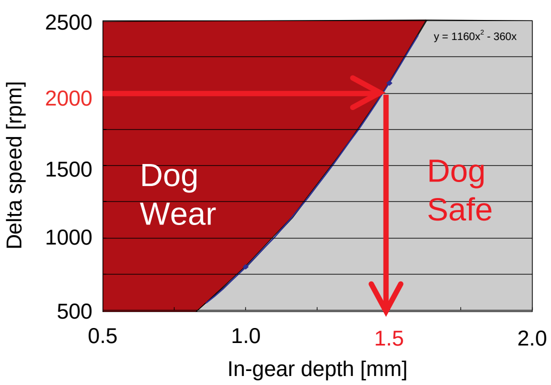


Fig. 5 Relation of dog delta speed and in-gear depth

In this formula, the value will be low for 2000 rpm. The recommended target position will be 32 degrees at the middle of the variation.

4.2. Avoiding Gear Over-Torque

Adjustment of the amount of clutch disengagement when shifting up so that the clutch transfer torque will be between the engine torque and the gear limit torque allows the impact torque to be absorbed by the clutch. However, the torque characteristics of the clutch lacked stability while the car was in operation due to changes in the coefficient of friction μ and the clamp load (the load pushing against the disk) resulting from disk wear. Therefore, stable absorption of the impact torque could not be realized with a fixed amount of clutch disengagement, so an auxiliary control of the clutch disengagement amount called clutch adaptation was applied. This control is focused on the amount of clutch slippage while shifting up, and it applies cumulative adjustments to the amount of clutch disengagement so that it will match the target slip amount. The clutch slippage amount while shifting up is calculated, and if it has not slipped more than the target amount, then the clutch will be adjusted in the direction of disengagement for the next shift up. If it has slipped more than the target amount, then it will be adjusted in the direction of tightening the clutch. Figure 7 shows a block diagram of the clutch adaptation control.

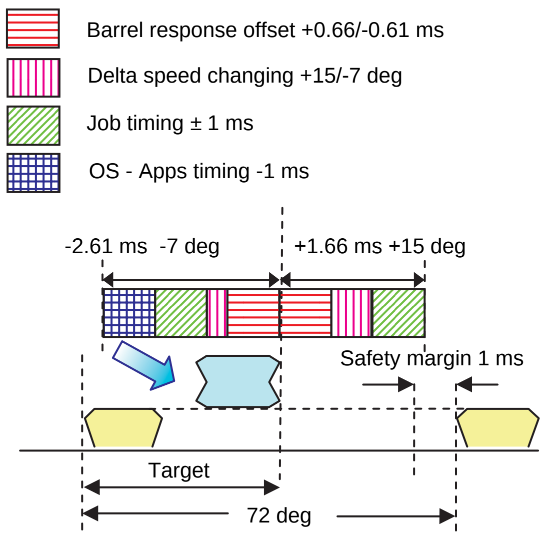


Fig. 6 Control tolerance of trigger shift

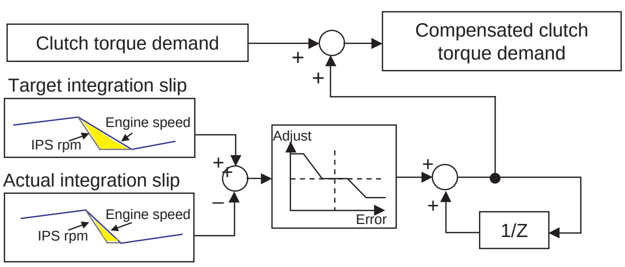


Fig. 7 Configuration of clutch adaptation control

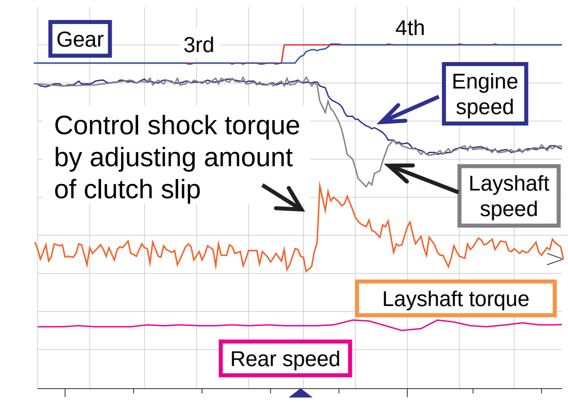


Fig. 8 Example of upshift behavior

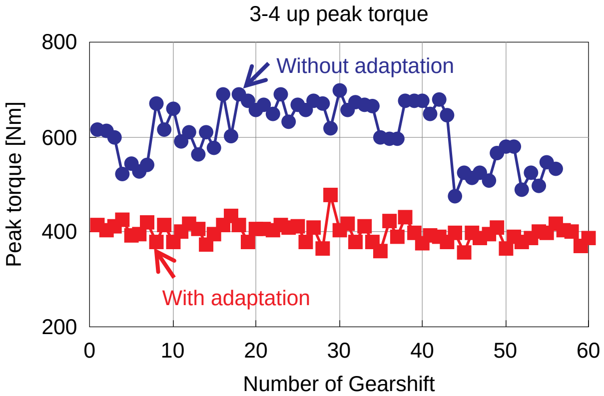


Fig. 9 Clutch adaptation result

Figure 8 shows the waveforms for actual shifts up from 3rd to 4th gear that absorb impact torque by making the clutch slip. Figure 9 shows a comparison of impact torque in the course of driving approximately 10 laps. It is apparent from this figure that when clutch adaptation is applied, the impact torque is held in the vicinity of 400 Nm with stability as intended.

5. Realizing Smooth Up and Down Shifting

5.1. Upshift

When shifting up with Quick Shift, as shown in Fig. 10, the inertial torque when shifting gears results in the wheel driving torque that exceeds the tire limits and induces excess wheel slipping as well as oscillation in the engine speed. The torque oscillation following a gear shift can lead to impaired drivability. Therefore, the following two points take on importance for upshift control:

(1) Limiting engine speed oscillation

(2) Maintaining optimal wheel driving torque

In order to quantitatively evaluate the engine speed oscillation, therefore, the oscillation level was represented by the deviation from a logical value for engine speed in the next gear integrated and depicted as an area, as shown in Fig. 11.

Figure 12 shows the summarized results of the correlation of this oscillation level with the slope when the engine speed is dropping. The horizontal axis of the graph shows the rate at which the engine speed is dropping, while the oscillation level is plotted on the vertical axis. It is apparent that the oscillation increases whether the slope of the engine speed is steep (part A in the figure) or too shallow (part B in the figure), and that it reaches its minimum value at -90 rpm/ms.

Figure 13 shows the summarized results of the correlation of the oscillation level and the delta engine speed with the next gear when the clutch is engaged. The graph has the difference between the engine speed when the clutch slippage ends and the engine speed at the next gear plotted on the horizontal axis, and the oscillation level plotted on the vertical axis. It is apparent from these results that engine speed oscillation when shifting up depends on both the engine speed when the clutch is engaged and the engine speed at the next gear. It is also apparent that reducing the differences between these speeds can limit oscillation, as well.

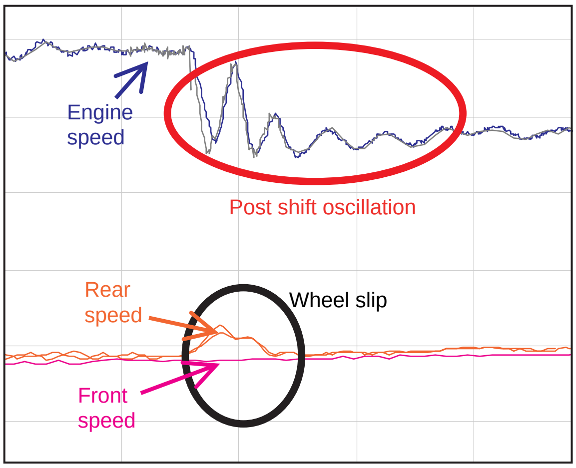


Fig. 10 Upshift oscillation issue

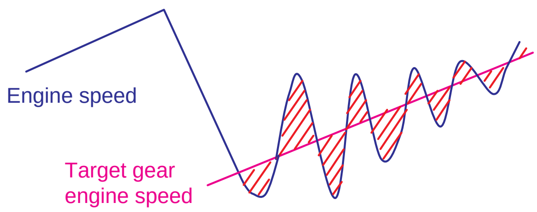


Fig. 11 Definition of oscillation quantity

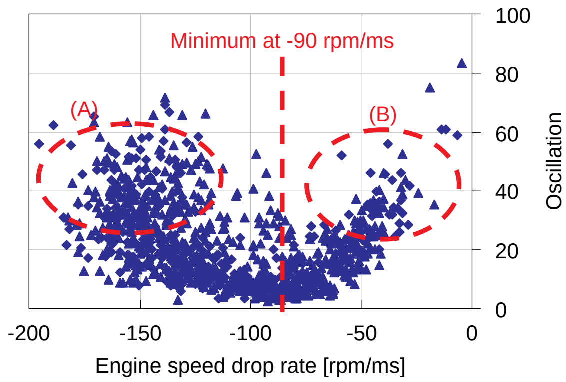


Fig. 12 Oscillation and engine speed drop rate

Analysis based on the above results showed that the oscillation level was smaller in the vicinity of the drop in engine speed at the rate of around -90 rpm/ms, which largely coincided with the torsional oscillation cycle in the drivetrain approximately 20 ms after the engine speed began to drop. Therefore, as shown in Fig. 14, it was found that oscillation could be limited by synchronizing it with the torsional oscillation of the drivetrain so that the clutch slippage is stopped at the point when the speed of the layshaft (which is on the other side of the clutch from the engine) is the same as the speed of the engine at the next gear, thus canceling out the torsional oscillation. It is necessary, therefore, to have the period of time until the engine speed reaches the next gear speed match with the torsional oscillation cycle, and this necessitates appropriate control of the engine torque reduction and the clutch slippage amount.

Control of wheel driving torque will next be described. The torque when shifting gears is subject to the influence of the inertial torque generated when the engine speed is reduced to the speed at the next gear. It is expressed by the following formula.

$$T_{Lay} = T_{ENG} - I_{ENG}\, d\omega / dt \qquad (2)$$

(T_{Lay}: Layshaft torque. T_{ENG}: Engine torque. I_{ENG}: Engine inertia. ω: Engine angular velocity)

In the above formula (2), the term $I_{ENG} d\omega/dt$ normally had a negative value when shifting up, so that it would be added on top of T_{ENG}. This increased the wheel driving torque, which would induce excessive wheel slipping. It was necessary, therefore, to control T_{Lay} so as to keep the wheel torque constant, and the target T_{Target} would then be expressed by formula (3):

$$T_{Target} = T_{ENG} \times \text{Previous Ratio / Next Ratio} \qquad (3)$$

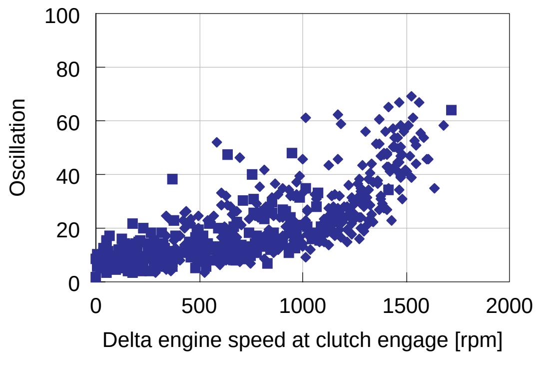


Fig. 13 Oscillation and speed error at clutch engage

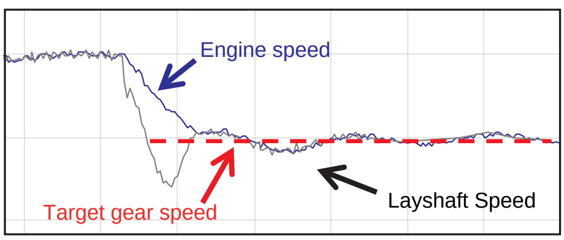


Fig. 14 Upshift example

The logic that realizes these aims can be described as follows. The target time for the engine speed to drop to the next gear speed is replaced by the downward slope of the target engine speed. The inertial torque $I_{ENG}d\omega/dt$ is defined by this slope of the target engine speed. In order to realize T_{Target}, therefore, $T_{Lay} + I_{ENG}d\omega/dt$ is supplied as the engine torque reduction demand during the gear shift.

T_{Target} is the clutch transfer torque T_{Clutch}, and where clutch adaptation had conventionally modified the clutch slippage amount, here instead it modified the slope of the engine speed by applying a correction quantity defined as a coefficient equivalent to the clutch coefficient of friction μ, thereby enhancing the accuracy of T_{Clutch}. Figure 15 shows a conceptual image of these elements of control.

Figure 16 shows the superposed actual upshift waveforms for the conventional upshift and the upshift with the new control applied. A smooth upshift was realized by the optimization of the slope of the engine speed and the engine torque reduction.

5.2. Downshift

Figure 17 shows the waveforms for a conventional downshift. The issues when downshifting are to limit the locking of the rear tires and to stabilize vehicle behavior when braking. The mechanism by which rear tire lock occurs during downshifting is described below.

(1) The dog delta speed immediately before being driven by the dogs is large.

(2) The dogs' engagement with each other leads to rapid convergence of the dog delta speed.

(3) Inertial torque is generated, this applies torsion to the

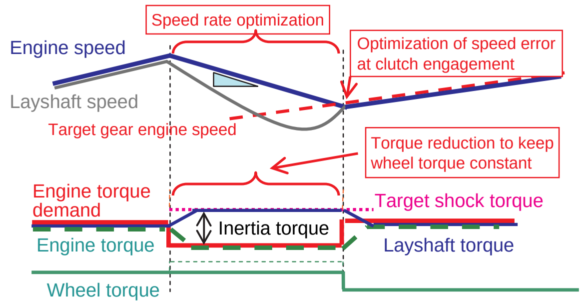


Fig. 15 Image of smooth upshift control

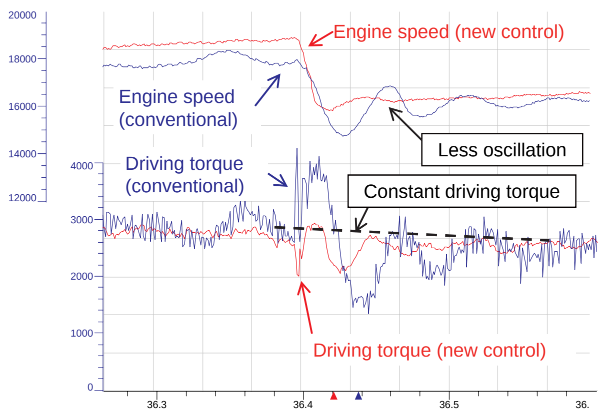


Fig. 16 Upshift comparison

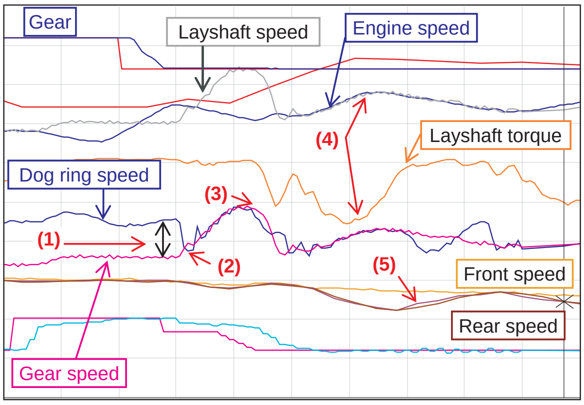


Fig. 17 Example of bad downshift

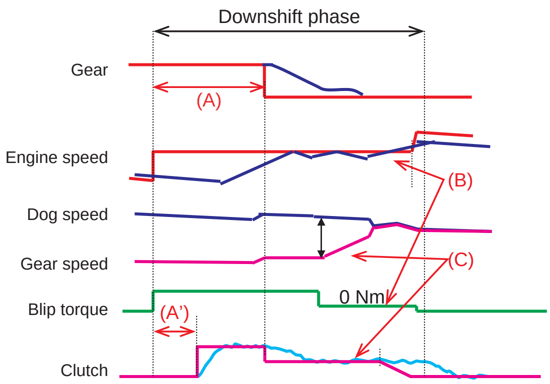
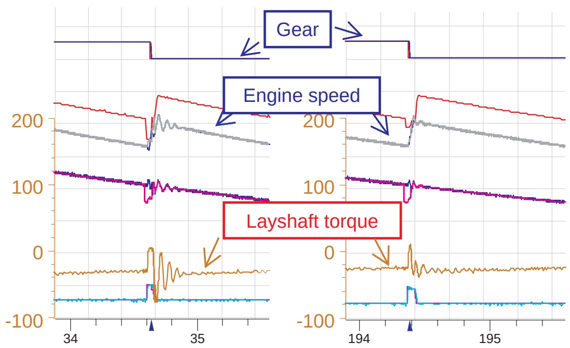


Fig. 18 Downshift control concept



(a) Standard downshift (b) New downshift strategy

Fig. 19 Downshift comparison

driveshaft in the direction of deceleration, and oscillation occurs.

(4) Engine speed is increased and the minus torque exceeds the rear tire drive torque limit.

(5) The rear tires lock.

The above item numbers correspond to the numbering in Fig. 17.

The major reason for deterioration of downshifting is the dog delta speed when being driven by the dogs while in gear. A smoother downshift was realized by the following kind of control of this dog delta speed.

(1) Make the dogs drive while the gear speed is increasing. (Cancel out the inertial torque while being driven by the dogs.)

(2) While being driven by the dogs, make the gear speed lower than the dog ring speed.

In (1), the clutch is disengaged and the engine is blipped to raise its speed. Then, the clutch is engaged, with timing matched to when dog driving occurs. The clutch torque raises the gear speed, and the dog delta speeds converge. In (2), the engine speed demand is set lower than the next gear speed to control the gear speed so that it will be lower than the dog ring speed.

Figure 18 shows a conceptual diagram of this control.

In the diagram, the barrel-driven timing (A) and the clutch disengagement initiation timing (A') are determined by the engine blip response. As noted above, it is important that convergence of the dog delta speed between dogs should occur gently, which is why the engine torque and speed instruction values are defined so as not to decrease the gear speed in (B). Also, as shown in (C), the clutch transfer torque is controlled appropriately so that the gear speed increases gradually, thus relaxing the impact torque when in gear.

Figure 19 shows the results of comparison between these controls and conventional control. Figure 19 presents the waveforms for the conventional downshift (a) and the downshift (b) with the control described above. Comparison of the torque behavior and the oscillation in the engine speed after downshifting confirmed that these are reduced when the new control is applied.

6. Conclusion

The Quick Shift that was applied in racing from 2005 had enhancements made to the Trigger Shift in 2006, and it achieved a record of zero retirement for gearbox reasons in races during the 2007 and 2008 seasons.

The new control was applied from 2007, and the enhancement of gearshifting yielded a zero level of related driver complaints.

■ Author ■

Takayuki KISHI

Yoshiki NAGATOSHI

Hiromasa NAKAMURA

Yoichiro FUKAO

Development of Direct Push Clutch Control during Honda Formula One Third Era

Takayuki KISHI* Yoshiki NAGATOSHI*
Hiromasa NAKAMURA* Yoichiro FUKAO*

ABSTRACT

One main development topic for the Formula One third era clutch was the direct push clutch (DPC), which was developed in order to increase the accuracy of transfer torque.

Conventional pull clutch position control could not readily stabilize the amount of torque transfer during standing starts, and therefore pressure control of the DPC was used in order to stabilize the clamp load. The non-linear characteristics of the hardware posed some issues with the hydraulic pressure response. However, these issues were addressed by control, and as a result response that had no issues in practical terms was able to be assured and the method was successfully deployed in racing.

1. Introduction

In the standing start range, the main focus of development has been to drastically curb wheel spin at the initial stage by enhancing the accuracy of clutch transfer torque estimation and raising the performance of the partial clutch engagement start. For clutches with the conventional structure, both the clamp load and the coefficient of friction μ were indeterminate factors, making it a considerable challenge to estimate torque. The DPC with its control of clutch transfer torque by means of hydraulic pressure was developed in 2005, but the hardware and the control were integrated as one, so that assuring the requisite pressure control performance became an important subject for development as had been experienced in the case of the gear box development. This paper introduces the DPC pressure control system.

2. Development Goals

In the development of DPC, the adoption of a device that uses hydraulic pressure to determine the clamp load increased the accuracy of estimation of transfer torque. However, the relationship between the hydraulic pressure and the length the piston travels in the action from the fully disengaged state to the bite point where pressure against the disk begins is different from that from the bite point to full engagement. As a result, pressure controllability became an issue. Controllability was also diminished by the hydraulic pressure vibration resulting from sticking and slipping of the piston that occur when the hydraulic pressure is raised from the fully disengaged state. In standing starts, speedy and accurate delivery of the amount of torque that results in the tires exercising the maximum gripping force is sought. The clutch response and convergence that will transfer the necessary torque are important, and the following objectives were defined regarding the controllability of the hydraulic pressure:

(1) Hydraulic pressure response of 30 ms or less up to the target value
(2) Hydraulic pressure overshoot of 1 bar or less

3. Overview of the DPC

Figure 1 shows the DPC configuration. The force applied to the piston is conveyed to the pusher in a lever structure with the fulcrum ring as the fulcrum and the pressure plate as the point of action. Thus the clamp load is transferred to the clutch disk.

Consequently, the DPC differs from the conventional pull clutch in that the clamp load can be directly controlled using hydraulic pressure.

* Automobile R&D Center

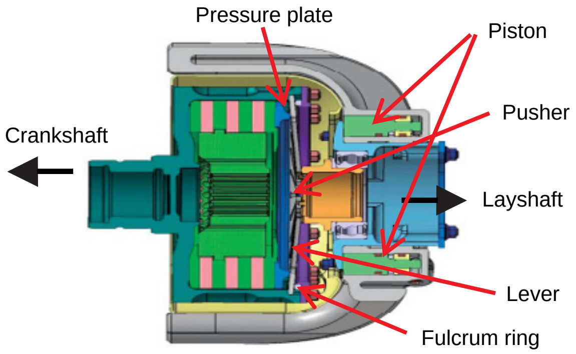


Fig. 1 Configuration of direct push clutch

4. Establishing Pressure Controllability of the DPC

4.1. Issues in Pressure Controllability of the DPC

Figure 2 shows the DPC stroke and hydraulic pressure characteristics as well as the clutch state, which are factors diminishing its pressure controllability.

In Fig. 2, the area (A) extends from the state of full clutch disengagement to the bite point where the clutch begins to engage. The amount of actuator movement is large relative to the amount of change in the hydraulic pressure.

When the wear of the clutch disk progresses, the distance that the pusher moves from the fully disengaged position until it hits the pressure plate grows longer. The area (A) in the figure expands to become (A’).

An area like this became a dead zone when under pressure control, interfering with the response, and not only did it diminish the effectiveness of gearshift, standing start, anti-stall, and other such types of control that require rapid clutch control, but it was unable to satisfy the limit on response time (50 ms) before reaching the target value as defined under FIA regulations. Therefore, this was very likely to result in a violation of the regulations.

In order to address these issues and achieve the target value explained earlier, the following types of measures were taken with regard to DPC pressure control.

4.2. Substance of DPC Pressure Control

4.2.1. Variable P-term gain, variable low-pass filter

In order to achieve the enhanced hydraulic pressure response described earlier, the variable P-term gain and variable low-pass filter (LPF) shown in the conceptual diagram in Fig. 3 were employed. This gain and LPF cutoff frequency are hydraulic pressure variables, and the aim was to raise the P-term and LPF cutoff frequency in transient states of the low-response area in the dead zone in order to achieve better hydraulic pressure response.

4.2.2. Simple state predictive P-term control

With variable P-term gain and LPF, the switching hydraulic pressure for the gain and cutoff frequency

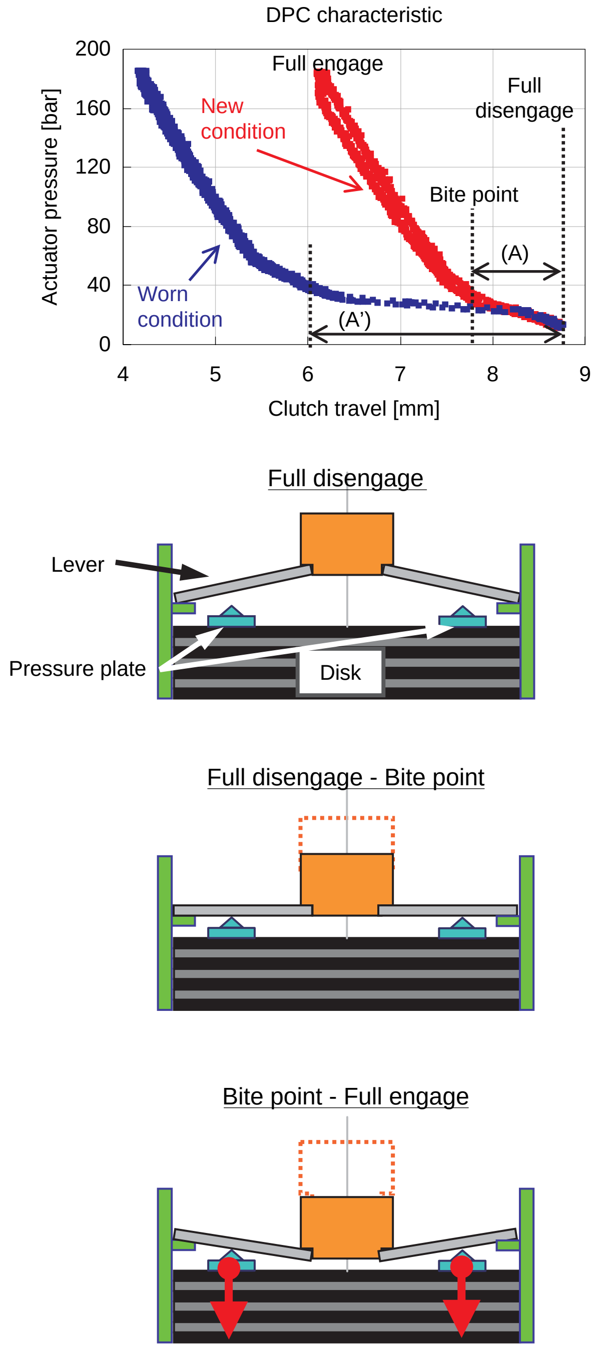


Fig. 2 DPC pressure-stroke characteristic

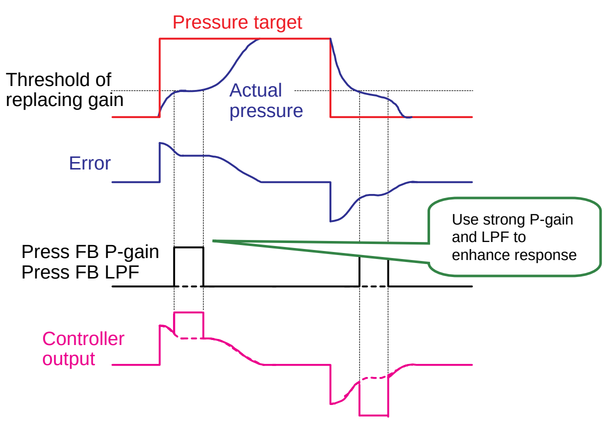


Fig. 3 Image of variable P control

depends on the hydraulic pressure in the dead zone. As explained earlier, the hydraulic pressure in the dead zone changes according to the wear of the clutch, and robustness could not be assured by variable control alone.

Therefore, the hydraulic pressure that follows after the time of non-response by hydraulic pressure (waste time) was estimated from the slope. Predictive control was implemented as shown below in order to use this predicted value as feedback to raise the P-term gain while reducing overshooting. Figure 4 shows the conceptual diagram.

(1) The time of non-response by hydraulic pressure is defined as the specified time. The hydraulic pressure following the specified time is estimated from the slope of the current hydraulic pressure.
(2) The amount of operation is calculated according to the difference between the predicted value and the target value.
(3) If the predicted value is greater than the target and the current value is smaller than the target, then the difference is considered to be zero and the amount of operation zero.
(4) If the predicted value is greater than the target and the current value is also greater than the target, then the amount of operation is calculated according to whichever has the least difference relative to the target.

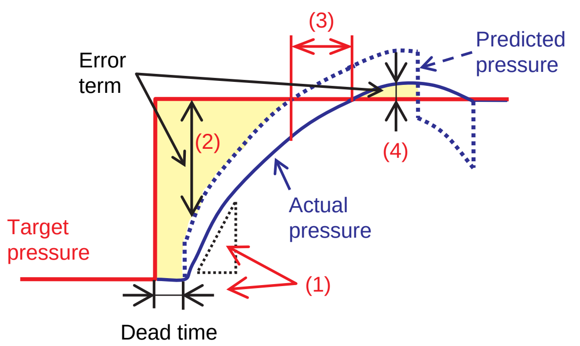


Fig. 4 Predictive P-term control

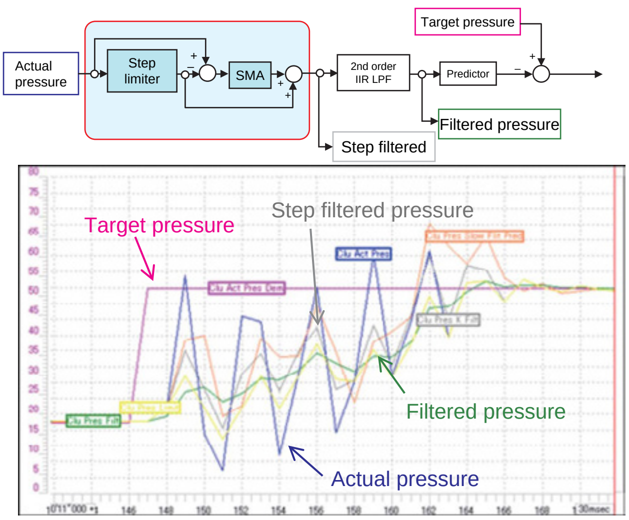


Fig. 5 Step limit moving average

4.2.3. Step limit moving average

Limits were imposed on the size of the steps in actual clutch pressure, taking the effective response frequency of the actual hardware into account. This limited the delay and resulted in the occurrence of significant damping to occur. This damped the vibration generated by sticking and slipping of the piston. Figure 5 shows the conceptual diagram.

4.2.4. Additive I-term control

Overshooting and undershooting were determined by the deviation from the target value and the slope of the hydraulic pressure, and the I-term was converted into a step form to increase the convergence. Figure 6 shows the conceptual diagram.

The filter value discussed earlier is fed back to the integral control. When this filter value deviates from the target value by more than a certain figure and the slope remains small for a set period of time, it is determined to be overshooting. The I-term is then reduced regardless of the feedback so that convergence with the target value will occur more quickly.

4.3. Results

Figure 7 shows the pressure controller that was

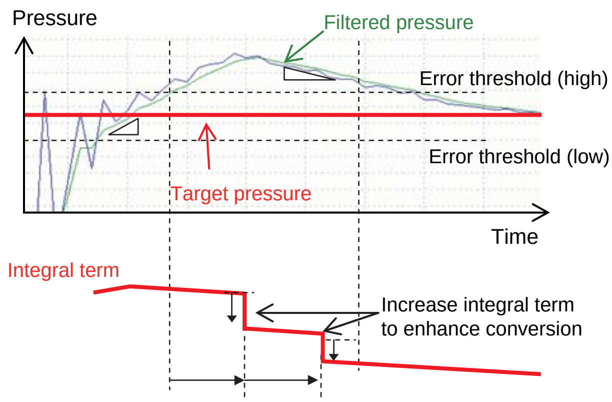


Fig. 6 Additive I-Term control

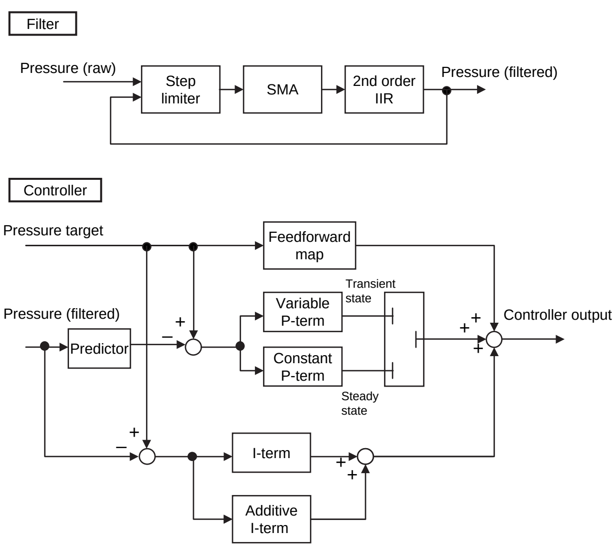
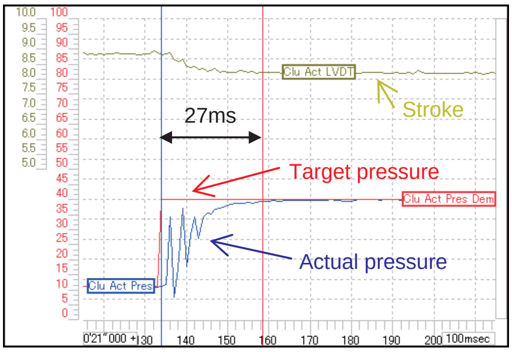


Fig. 7 Configuration of DPC pressure controller

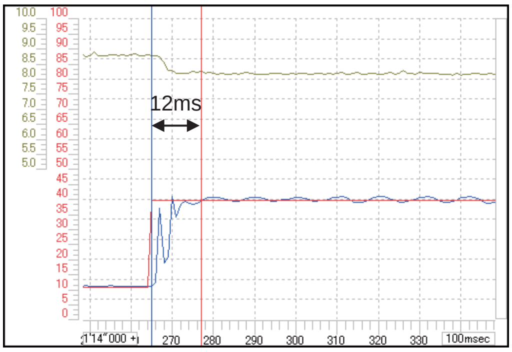
ultimately realized.

Figure 8 shows a comparison between standard PID control and the control described above. The comparison was carried out during step response assuming a standing start, when accuracy and speed of hydraulic pressure response are required. Comparison was also carried out with 3.0 mm of clutch wear, which corresponds to the amount of wear from a single race, since response can be assumed to diminish when clutch wear results in an expanded dead zone.

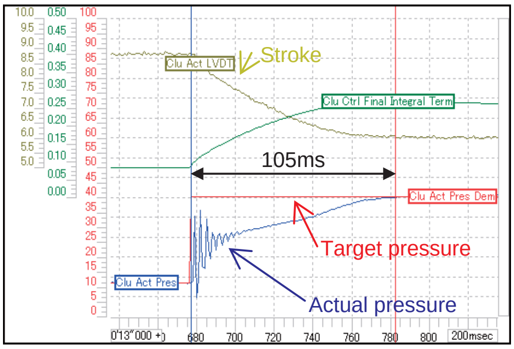
The hydraulic pressure response time was successfully reduced, without overshooting, by 44% when there was no wear and by 71% when there was wear. Even when wear had occurred, the diminishment of standing start performance was kept to a minimum.



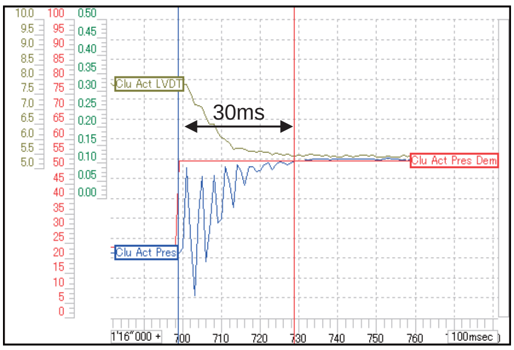
(a) Standard PID (no wear)



(b) New control (no wear)



(c) Standard PID (3.0 mm wear)



(d) New control (3.0 mm wear)

Fig. 8 Comparison of pressure control

5. Conclusion

The establishment of pressure control of the DPC enabled an increase in the accuracy of clamp load (transfer torque) control.

The DPC was employed in racing from 2006, and the implementation of a partial clutch engagement start throughout the entirety of the standing start enabled realization of slip starts without any torque feedback control. This yielded the advantage of a maximum 0.46-second reduction in 0-100 km/h acceleration (an 11-m advantage in terms of distance).

■ Author ■

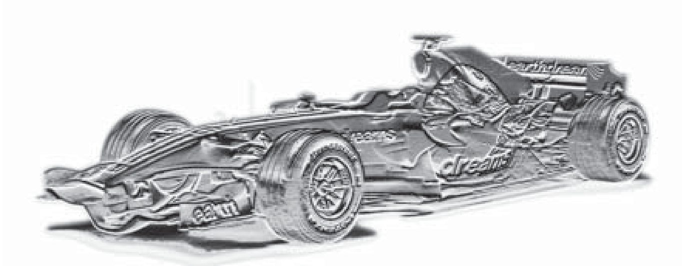
Takayuki KISHI

Yoshiki NAGATOSHI

Hiromasa NAKAMURA

Yoichiro FUKAO

Descriptions of Electronic Equipment Technologies



Development of Electronic Control System for Formula One

Kenichiro ISHII* Toshiyuki NISHIDA* Kohei TOSHIMA*
Masaki NEGORO* Masataka YOSHIDA* Yutaka MARUYAMA*

ABSTRACT

The application of electronic control systems has been rapidly increasing in Formula One cars as well as in other vehicles. System performance is a crucial element in conducting precision control and measurement of cars.

Starting with the 2006 season, the Honda works has been working to apply original Honda systems not only in the engine control system, as before, but in all vehicle electronic control systems.

In order to pursue higher performance at the same time as enhanced in-vehicle mountability in Formula One electronic control systems, it is necessary to carry out optimization of these systems together with thoroughgoing miniaturization. On-board systems have an electronic control unit (ECU) with integrated functionality linked by a high-speed network with units located in every part of the vehicle. High-speed telemetry is used to coordinate these with the garage system in order to optimize systems.

The enhancement of unit performance by means of higher speed and greater precision contributes to heightened controllability, and particularly to enhanced precision of driving force control and gearbox control. High-speed communication also contributes to greater measurement performance in the pit.

1. Introduction

Formula One cars in recent years have incorporated more than just the engine control and gearbox control found previously. From mid-2001 to the 2007 seasons, traction control and engine brake control have come to be allowed in the regulations. Clutch control was not prohibited up until 2007.

Traction control and gearbox control, in particular, require precisely coordinated control of the engine and chassis in real time. This has required advanced computational capability and measurement performance. Furthermore, Formula One cars demand aerodynamic performance, so that there are limited on-board installation locations for electronic control units in Formula One cars. This means that systems require greater compactness and lighter weight.

The functions of electronic control systems for Formula One are generally for two purposes, for driving and for analysis. In order to achieve compactness and light weight, the latter functions are assigned as much as possible to pit systems. In this way the former functions can be optimized, and this was the basic conceptual approach to the construction of these systems.

This article will provide an overview of Honda Formula One electronic control systems, introduce the issues involved in development of the systems, and describe the contents of the development.

2. System History

Figure 1 shows the history of electronic control systems for the third era of Honda Formula One. Development of third-era Formula One systems began in 1998 with a view to applying integrated engine and chassis systems to racing.

Under the BAR-Honda system from 2000 to 2005, engine control systems were supplied to the team. From 2006, when Honda works joined in the competition, the Honda full system was provided as a chassis integrated

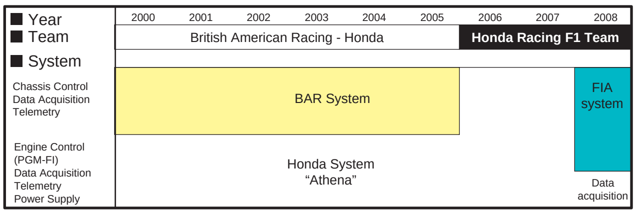


Fig. 1 History of Honda Formula One system

* Automobile R&D Center

management system. The system was named Athena after the Greek goddess of wisdom and victory.

In 2008, it was made mandatory to apply International Automobile Federation (FIA) standard systems. The purpose was to standardize engine and chassis control and to reduce costs. As a result, Honda's in-house developed systems that were used in racing were limited to the devices for engine use, sensors, and data logging systems.

3. Configuration of Electronic Control Systems (Athena)

3.1. Optimization of System Configuration (Functional Integration and Distribution)

A car's aerodynamic performance, which determines its body design, makes a very important contribution to lap times. In order not to place constraints on the freedom of the design, it is necessary to have system configurations that are compact and have outstanding mountability.

The main locations for mounting electronic unit were, as shown in Fig. 2, beneath the radiator ducts on either side of the chassis. The only other locations were near the throttle pedal toward the front of the car and inside the cockpit. The systems were therefore mounted in a distributed configuration and were linked by a high-speed network in order to achieve a balance of mountability and functionality.

Figure 3 shows the system configuration. The network in the chassis is centered on the ECU and ties together the various units, sharing sensor data and fail data to operate the systems. The ECU has the functionality for integrated control of the engine, chassis, and measurement. It acquires all information from the chassis and conducts the operation of every device as well as the processing of measurements. Each unit that has been distributed can manage the measurement and control functions for the different applications, and is connected to the network as necessary. Of these, the Front Data Acquisition (FDA) unit acquires data from numerous sensors at the front of the chassis. Since it processes signals used for real time control computations, it is connected with the ECU by high-speed communications at 10 Mbps.

In the pit, the client PC and the ECU are connected

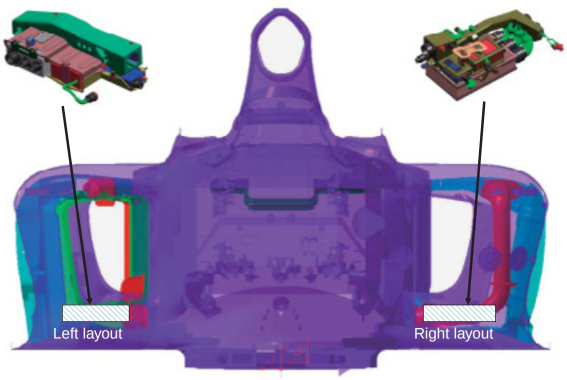


Fig. 2 Electronic unit layout (front view)

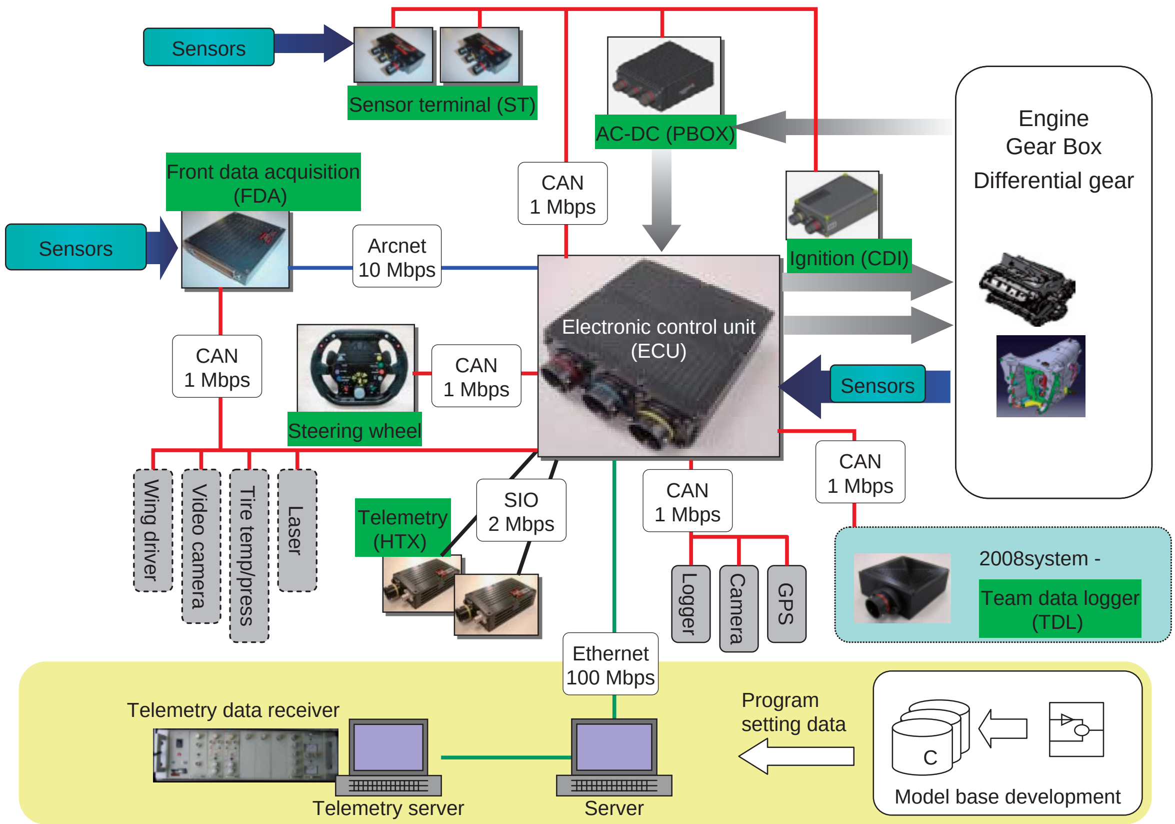


Fig. 3 Athena system

by Ethernet communications (100 Mbps) so that logging data and other high-volume data can be received in a short time. All vehicle checking applications that handle high volumes of data are also placed on the client PC side in order to simplify on-board systems.

3.2. Explanation of Each Component

(1) ECU

The ECU has three roles, in engine control, chassis control, and measurement control. In engine, it controls the ignition timing, volume of fuel injected, and throttle opening, and realizes the requested torque. In chassis control, it uses vehicle behavior information to perform wheel drive torque transfer control and seamless shift control computation for gear shifting. In measurement control, it performs measurement computation of two kinds, in telemetry and data logging.

(2) Capacitive Discharge Ignition (CDI)

The CDI is the unit that supplies the energy for ignition to the engine ignition coil. The CDI method provides greater real time effectiveness than the full transistor method and greater precision in drive torque control (traction control and engine brake control) and engine speed control. This unit also simultaneously has misfiring detection functionality by means of ion current detection.

(3) Power Box (PBOX)

The PBOX is the unit that operates as the voltage regulator and power distributor (sequencer function). The voltage regulator part uses a highly efficient converter method and supplies two power outputs (14 V and 7 V) so that the power components where this power is supplied can be made more compact.

(4) Batteries

The system uses NiMH batteries for compactness and lighter weight. Battery temperature monitoring and charge control are handled by the PBOX.

(5) Telemetry transmitter and receiver (HTX, HRX)

The HTX is the data transmitter unit by which data processed by the ECU is sent by radio to the pit. The HRX is the data receiver unit that receives radio data.

(6) Front Data Acquisition (FDA)

The FDA is the unit that acquires data from sensors at the front of the chassis. It is equipped with input/output (I/O) capable of multi-channel analog input. This functions to process acquired analog signals by filtering computations and then transmits them to the ECU by high speed communications (10 Mbps).

(7) Sensor Terminal (ST)

The ST is a compact sensor measurement unit used for measurement of the environment in the chassis and in the engine. Up to eight ST units can be connected to the ECU and CAN communication lines.

(8) FIA/FOM units

The FIA/FOM units include a data logger, camera, and GPS units. Installation of these units is required by the Fédération Internationale de l'Automobile (FIA), the umbrella organization for automotive matters, and by the Formula One Management (FOM), the organization that conducts racing. Car data from the ECU is sent to the units by CAN communications, and the racing circuit data for each team is managed and used by the FIA and FOM.

(9) Team Data Logger (TDL)

The TDL unit has a logging function and reads data from all types of sensors. The use of FIA standard systems has been a requirement since 2008. This unit is therefore developed so that assistance tool used up to 2007 can continue to be used. The TDL is connected with FIA standard systems by CAN communications.

(10) Garage system

The garage system is system infrastructure composed of the server installed in the pit and the client PCs used by each engineer. It was designed exclusively for use in Formula One racing. As for the communication between the car and PC, Ethernet was used.

3.3. High-Speed, High-Precision Control (Engine, Chassis, and Measurement Control)

The ECU was required to have high computational capability in order to provide satisfactory control performance and measurement performance. In the third era of Formula One, the main items for control development have been drive torque control and seamless shift control. The efficiency of this control development was enhanced by the introduction of model base development using floating point operations. Figure 4 shows the controls that were applied, together with the changes in computational capability. The regulations made driver assistance control (slip control, typified by traction control) permissible in mid-2001. From then up to 2007, when it was outlawed, this demanded the greatest computational capability. This also involved an increase in the volume of data for control analysis, and the CPU and data transfer speeds were increased in order to be able to support the demand for high-speed sampling and logging on multiple channels for measurement function.

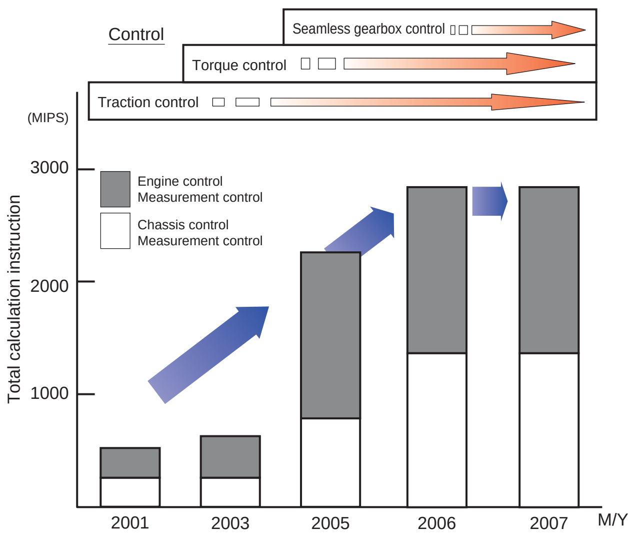


Fig. 4 Processing speed increase

3.4. On-Board Layout

The main electronic control units were placed beneath the radiator duct. This location was at the center of the chassis as a whole, as well as at a low position, and it was the largest area where units could be mounted without affecting the vehicle dynamics. As shown in Fig. 5, the ECU and CDI are placed on the left side of the chassis between the radiator duct and the under floor. They are affixed to carbon fiber brackets on anti-vibration mounts. As shown in Fig. 6, the PBOX, battery, and HTX are placed on the right side between the radiator duct and the under floor, like the units on the left side. In order to cool the units, a duct shape capable of introducing the necessary cooling airstream was simulated (Fig. 7), and on that basis a duct shape that would have minimum effect on the aerodynamics was created. This meant that the heat radiation structure of the electronic control units was crucial.

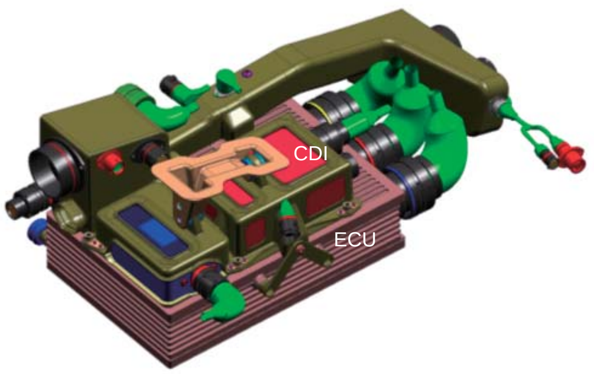


Fig. 5 Left side electronic units

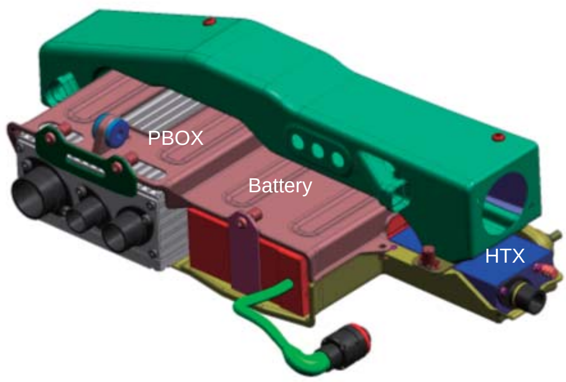


Fig. 6 Right side electronic units

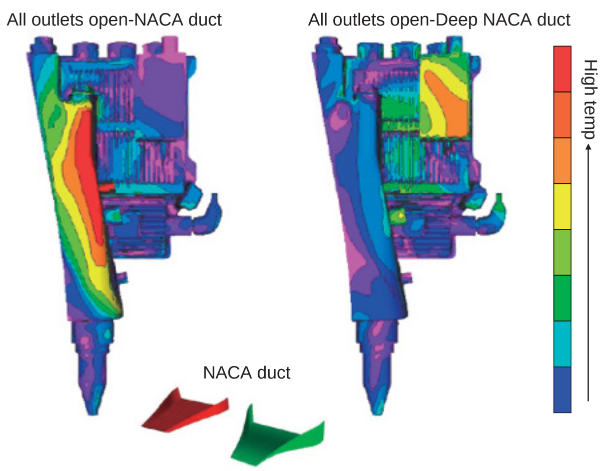


Fig. 7 Cooling duct simulation

4. ECU Architecture and Functions

4.1. ECU Hardware Architecture

In order to realize the performance requirements for the electronic control systems described at the start of this article, the hardware performance of the ECU in particular was important. An architecture with multiple CPUs optimally suited to the various functions carried out by the ECU was created. The memory architecture that coordinates those CPUs and their communications links are described below.

The basic architecture of the ECU, as shown in Fig. 8, includes three CPUs: the Application CPU (ACPU), the Device CPU (DCPU), and the Gateway CPU (GCPU). The ECU was configured in two blocks, one for the engine and the other for the chassis, and their I/O were configured for their respective input and output devices.

A balance between real time operation of the devices and high-speed computational processing of applications was sought by distributing functions so that processing would not be concentrated in a single CPU. In the part for device operation, operations of the injector and the ignition device take place at 150 μs intervals when the engine is running at high speed. Consequently, these were made into functions handling only device I/O operations, and were separated from the processing of applications that require high throughput computation. The selection of a dedicated automotive CPU (DCPU) with better I/O functionality therefore satisfied the functionality requirement.

A high-speed processing CPU (ACPU) was selected to handle the control applications used with the engine and the chassis. What had until then been controlled using three automotive microcomputers was combined into one. This reduced the need for data access between CPUs for applications and reduced unnecessary processing.

The CPU used for measurement control carries out high-volume data communications with PCs and communications with telemetry units. A CPU (GCPU) that is high-speed and is equipped with many varied communication devices was therefore selected, allowing data operations that make use of high-speed serial and Ethernet communications.

The multiple CPUs with distributed functions have to look up each other's data. For example, data that is necessary for telemetry and logging is sent to the GCPU and the device instruction values that are output by control computation are sent to the DCPU. The calibration data and commands that are sent from client PCs in the garage system are shared by every CPU. In order to implement these actions, DPRAM was placed between all the CPUs and high-speed synchronized access was enabled by means of trigger signals when they make connections.

Local memory is assigned to each CPU according to the functions that are required. Memory functions include flash memory for program saving, SDRAM for program execution, NVRAM for data backup, and Compact Flash (CF) memory for use in high-volume logging. CF memory is installed with two cards in parallel in order to expand the data bus width from 16 bits to 32 bits. The purpose was to achieve higher-speed data processing and access processing during logging.

The engine and chassis blocks are connected by ARCNET communications at 10 Mbps. This is to transfer the parameters required for coordinated control above 1 kHz process cycle rate.

Up to 2008, the GCPU performed logging data retrieval processing and data download processing in alternation. How to conduct these processes simultaneously, therefore, became the technical issue when a balance between increased logging volume and reduced download time was sought.

The high-speed memory controller shown in Fig. 9 was therefore developed. The logging data retrieval process and download process could be conducted in parallel by using the PCI bus and the CPU local bus. The increased speed and greater volume in reading and writing logging data was realized by means of a circuit architecture with multiple SD memories connected in parallel. This memory controller was installed in the TDL developed for the 2009 model, and it achieved a 30% increase in data download speed together with an increase in capacity (to a maximum of 8 GB).

4.2. ECU Software Architecture

The ECU software can be generally divided into these main parts: the engine control part that handles control of engine ignition and throttle, the part that handles control of the gearbox and vehicle behavior, and the data measurement part that handles the recording of circuit data and its transmission to the garage system.

A real time operating system was implemented on each CPU. This met the demand for the advanced real time performance needed to realize high-speed control computations, sensing, and communications. As shown in Fig. 10, the software is divided into three distinct and independent layers, namely, the OS layer, the middle layer that performs communications and I/O control, and the application layer that performs control computations.

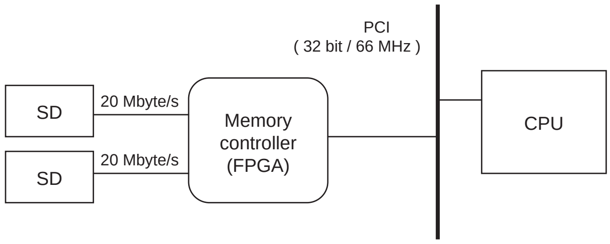


Fig. 9 SD memory controller

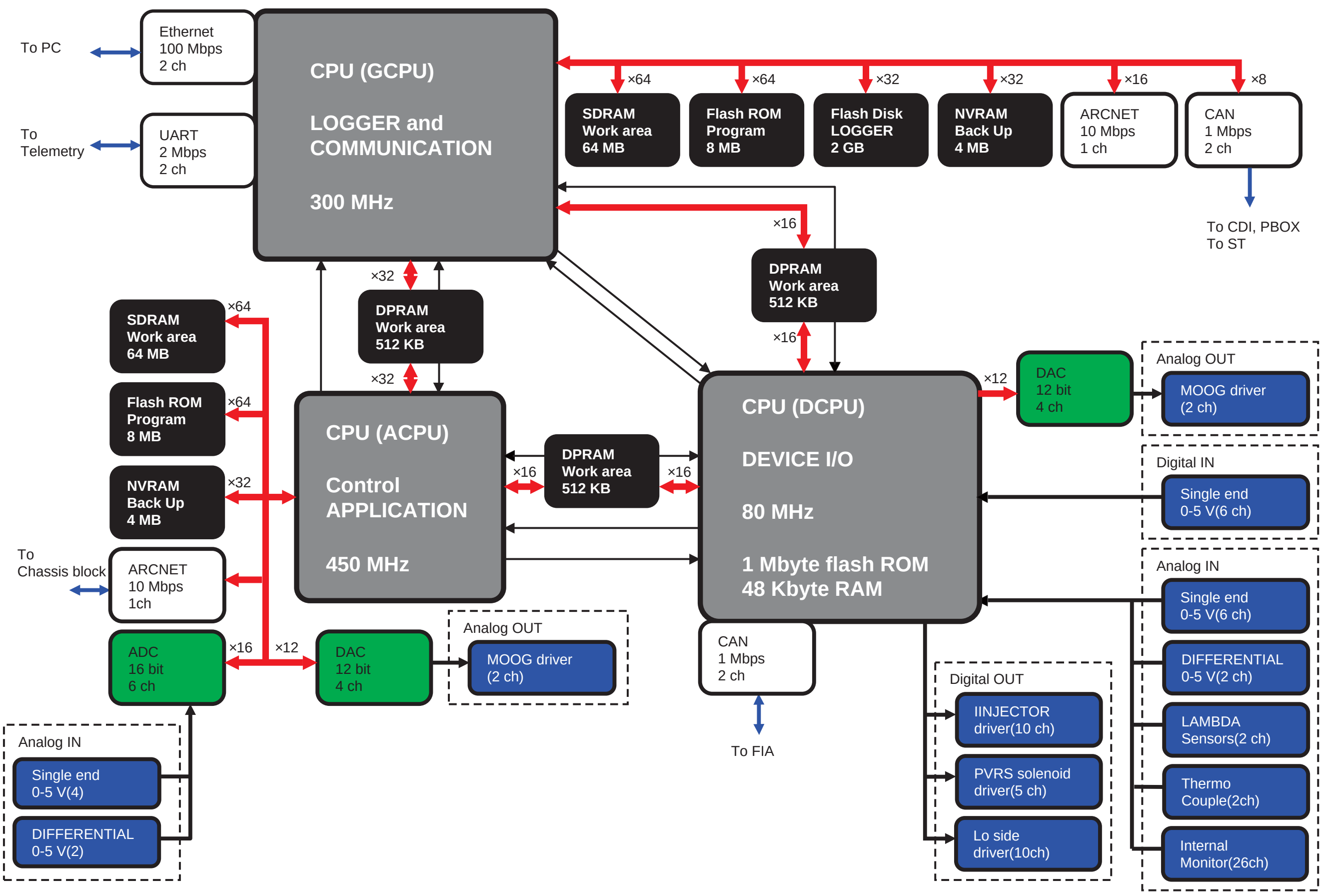


Fig. 8 Hardware architecture

In order to enable independent control development by application users in model base development, a development environment capable of automatically generating execute files was set up and control development efficiency was increased (Fig. 10).

4.3. Circuit Data Measurement Function (Logging)

The logging function is used to make settings in the chassis and power train, to monitor conditions, and to conduct simulations. It therefore needs to provide accurate data and handle large volumes of data. This required specifications that include 2000 channels and sampling rates from 10 kHz to 1 Hz.

For the efficiency of circuit tests, it is important for the engineers to be able to quickly determine conditions in the vehicle after it has been driven, and make it ready for the next driving session. It was necessary, therefore, to shorter the time from data download to data display, and to provide techniques for easy analysis.

Logging data is accumulated each time the car is driven. This is used not only on the circuit, but also for analysis and simulation at the various development centers. It was necessary, therefore, to associate the circuit data and the test items.

4.3.1. Measurement data format

In order to facilitate data management in every driving situation, data measurement using the logging system was set up to split off the data from each driving session and write it to the logging memory (Fig. 11). A single driving data is composed of the Run Header and the Run Data.

The Run Header area records the Lap Data, which is updated on every lap; the Straight End Data, which gives the highest speed point at the ends of straightaways; and the Fail/Warning Data, which gives information on malfunctions. These were arranged to enable engineers to readily identify characteristic points in each lap. Index data for logging-related data is also recorded so that circuit data, calibration data, and the like can be tied together and accurate data management can be conducted after driving sessions.

The Run Data area contains records for every channel subject to logging.

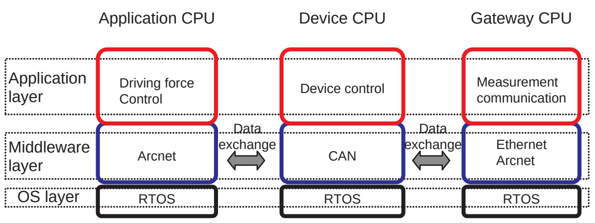


Fig. 10　Software architecture

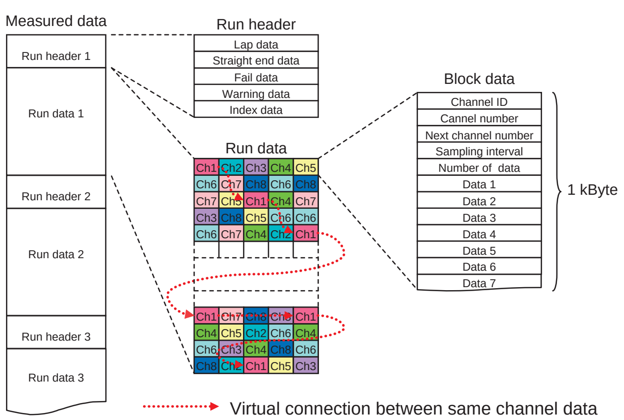


Fig. 11　Logging data format

4.3.2. Data recording techniques

As explained earlier, it is necessary for data logging to record data from measurements conducted at a high sampling rate. Data logging with a high sampling rate involves increased amounts of data, which leads to increased download times, and this ultimately diminishes the efficiency of circuit tests. For this reason, a trigger logging function was developed. Under specific circumstances, such as when a gear change is performed, when a malfunction occurs, or when some other condition necessitating acquisition of detailed data at a certain rate occurs, past data is recorded at a high sampling rate for a maximum of one second back from a trigger point. At other times, data is recorded at a low rate so that the volume of logging data can be reduced.

The trigger conditions are defined on a client PC in the C language and written into the logging configuration file using reverse Polish notation. After the ECU has decoded the trigger conditions, it implements trigger processing when those conditions are met. The system was set up to allow the configuration to be changed according to test circumstances.

When initially developed, sampled data was simply put in time series and the data on each channel was recorded in logging memory using a discontinuous method. Since the aim of this was to reduce the processing burden on the ECU. However, when CPU performance was enhanced, the volume of logging data increased and the issue of the time required to display a graph occurred.

The use of a high-speed CPU (GCPU) enabled the data recording format to be changed so that the data in each channel are in continuous sequence and this upgraded the drawing speed. The data recording format was changed so that the measurement data for each sampling from the same channel is placed in a single block with a fixed length of 1 kB, and measurement data is recorded as an aggregation of blocks. (Figure 11 shows data from the same channel in blocks of the same color in the middle column.)

With this change in format, the block of the same channel is read by unit and taken united, so an increase in graph display speed was achieved as a result.

This logged data can not only be displayed in analysis tools, it can also be input to Hardware In the Loop Simulation (HILS) and used to reproduce driving conditions on a PC. It has also been used to feedback measures to solve circuit-running issues.

4.4. Garage System Functions

The purpose of the garage system was to manage circuit data, provide race strategy support, and assist in conducting vehicle checks. The functions that are important in running the car were built into the ECU. Other additional functions for support purposes were to be taken on by the garage system. This was the conceptual approach in development, and it was tied-in with miniaturization of the ECU (Fig. 12).

In the pit, the car is connected to the garage network where such actions as logging data capture by client PCs and transfer of calibration data take place. The logging data is stored on the garage system server so that the data can be looked up by multiple client PCs. It is also given numerous functions for support of racing strategy and test runs, and the system was set up to allow operational commands to be issued to an ECU that is connected to the network. Typical support functions and their association with the ECU are described below.

(1) Data setting support function

At the actual race location, engineers change the engine and chassis settings on the basis of circuit data. Changes to settings are made frequently, and such changes need to be made up to the point immediately before a race begins. Therefore, the system was set up so that the data transmitted from client PCs to the ECU would be the only data changed by engineers on client PCs, enabling instant setting changes.

(2) Auto Warm-up

Performance enhancements in the engine and gearbox have brought increased complexity in engine starting and warm-up as well as the engine checking mode. Auto warm-up is a support function that automates these activities. This check mode includes numerous check modes for determining the engine status, and the engine is controlled using control instruction values sent from a PC. The ECU is not prepared beforehand with a profile of these check modes. Instead, the system specification calls for the profile to be stored in the form of files on client PCs in the garage system. This allows the creation of a variety of check modes adapted to engine specifications and environment of use without changing the ECU software.

(3) Configuration of Steering Wheel Functions

In addition to driving operation, the steering wheel has the capability for the driver to check or make changes to settings and car information while driving, in accordance with racing strategy and changes in the circuit environment. Figure 13 shows how the steering wheel is equipped with multiple switches, buttons, and an LCD display (Fig. 14). The functions given to all of these controls and the content shown on the display can be freely assigned and set up from applications on a PC as requested by the individual driver. By sending information to the ECU through the garage system, the steering and ECU functions can be linked together. Frequently used and important functions are assigned to push switches and rotary switches so that those operations are carried out with one action. Other operations are placed in command hierarchies and accessed using a Mode Function switch and the "+/-" switch. This realized a balance between operability and multifunctional switchability.

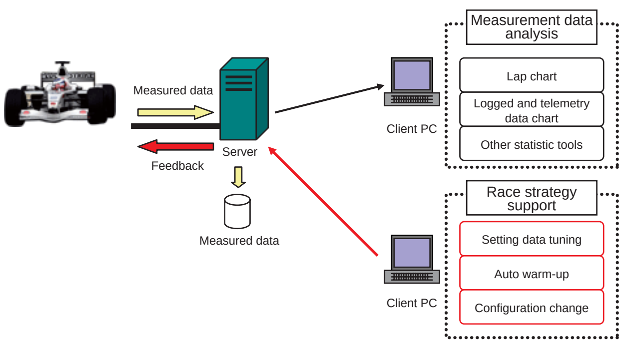


Fig. 12 Garage system

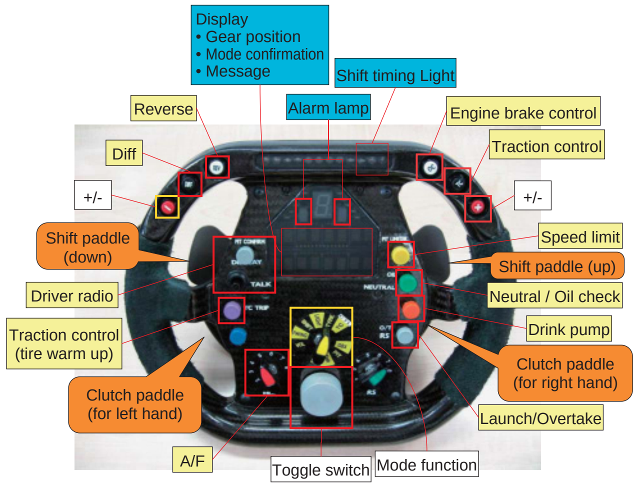


Fig. 13 Steering wheel functions

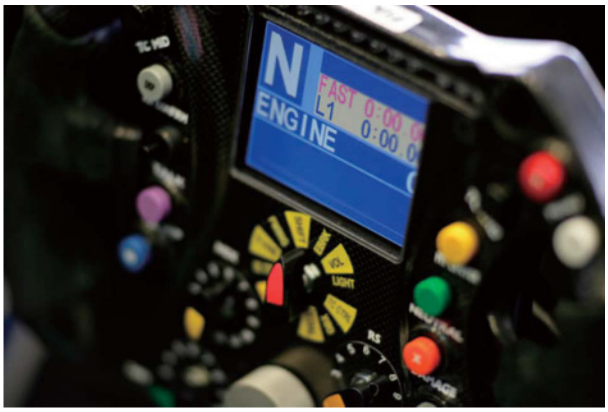


Fig. 14 Steering wheel display

5. Hardware Development

The electronic control units were given a three-dimensional structure using multiple circuit boards in order to increase the package density. Circuit boards densely populated with electronic parts have large numbers of wiring connections, and it was necessary to consider techniques for making connections between boards. Higher-performance CPUs also operate at higher speeds and generate considerable heat. Techniques for heat dissipation in miniaturized packages were an issue.

5.1. High-Density Packages

Electronic component packaging techniques that had no track record of use in automotive electronic components at that time were employed in 2003 to realize high-density packages. These were the Fine-pitch Ball Grid Array (FBGA) and the Build-up printed circuit board (PCB), and their use realized miniaturization. Figure 15 shows a circuit board populated with CPUs using the Build-up PCB technology. For the 2009 model TDL, a prototype unit was fabricated through the application of Device Embedded PCB technology. This was confirmed to achieve 20% greater packaging density than Build-up PCB products.

CPU operation at higher speeds was accompanied by faster bus clock rates and CPU electrical power supply at a lower voltage (1.8 V). This made the units more susceptible to being affected by disturbance noise and cross talk. Consequently, there was an even greater demand than before for stabilization of pattern shields and ground electric potential during pattern design, and circuit board wiring design techniques based on high-speed transmission path simulation were adopted.

5.2. Inter-Board Connection Methods

In the case of unit structure from multiple circuit boards, the method used to connect boards with signal wires becomes an issue.

Larger units involve a great number of inter-board signal wire connections. Therefore, not only the connectors can cause dead space, but it is apparent that even more space is required when wire routing is taken into consideration.

In order to resolve this issue, it was decided that the 2008 model TDL would reduce the inter-board connectors used up to that time by instead using Rigid-Flexible circuit boards to optimize inter-board wiring. At the same time, pattern shields could be built in at signal wire connections for communications and small signals, and the connections could be made electrically stable as well (Fig. 16).

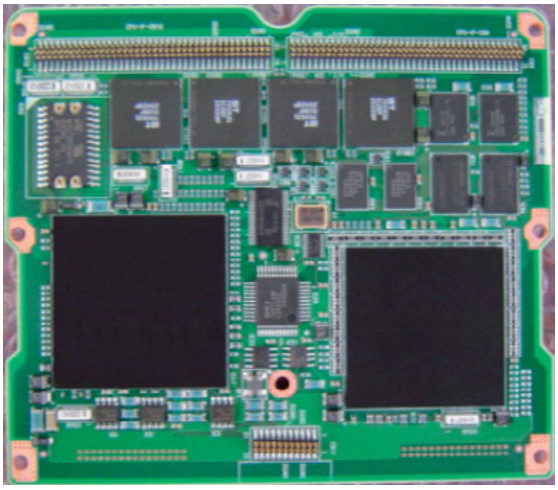


Fig. 15　High density PCB surface

5.3. Heat Dissipation Structure

Generally speaking, the automotive CPUs used in an ECU often do not require heat dissipation measures. The high-speed CPUs used in Formula One ECUs, however, have power consumption that rises as high as about 4 W, and they exceed the guaranteed operating temperature of 105°C even at room temperature (25°C).

In order to achieve a balance of high processing performance and miniaturization in the Formula One ECU, the greatest issue was how to deal with heat. This was a question of whether the unit could be made suitable for use in the actual car environment.

Heat dissipation for high-speed CPUs and other such heat generating devices is generally accomplished by forced air-cooling using heat sinks. In Formula One ECUs, for which miniaturization is sought, maximum use was made of the case and the circuit boards to enable guaranteed operation even in the actual car environment where the ambient temperature rises above 60°C.

The heat dissipation measures will be described here, using the ECU as an example. Figure 17 shows the main electronic parts that generate heat, the heat dissipation structure, and the heat dissipation pathways. It is important to dissipate the heat that is generated and to increase the heat capacity.

Regarding the former, a pattern was formed with a circuit board affixed to an aluminum plate made from the same material as the ECU case in order to enhance heat conduction from the device and the circuit board, thus using the case structure for heat dissipation. Regarding the latter, metal core PCBs were adopted for the circuit boards that have CPUs mounted. These have high heat conductivity in the planar direction, and the heat capacity of the circuit boards was increased as a result.

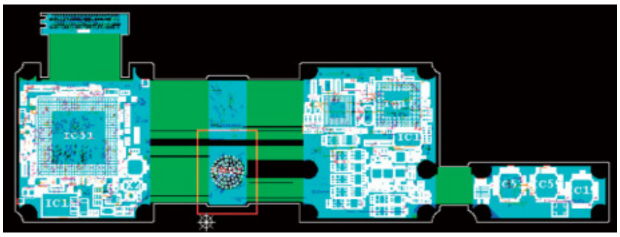


Fig. 16　Rigid-Flexible PCB of TDL

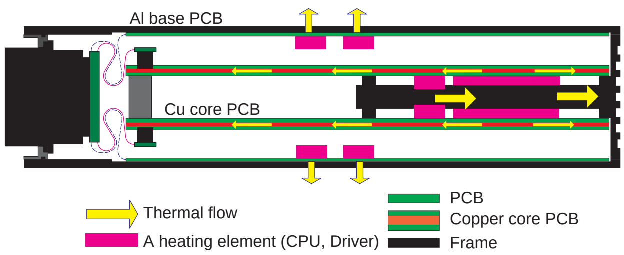


Fig. 17　High heat conditioning structure of ECU

By means of these various techniques, the unit could be made suitable for the actual mounted environment in the car.

5.4. History of Miniaturization

Figure 18 shows the historical changes in the ECU's package size and the technologies employed. There has been demand for higher processing performance since the 2004 specifications. This was achieved employing the technologies described above, so that higher performance was realized while maintaining the size of the 2004 model.

Growing demand for higher performance and greater miniaturization of ECUs in mass production vehicles, as well, will no doubt lead to importance being placed not only on measures for heat dissipation in electronic components, but also on technology to inhibit the generation of heat in devices.

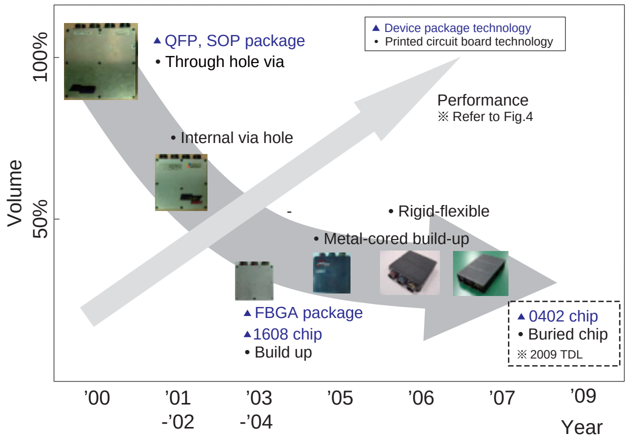


Fig. 18 ECU package

6. Telemetry

6.1. History of Telemetry Development

Telemetry is the system that takes data from the various kinds of sensor data and the like while the vehicle is running and uses radio to transmit it to the pit. Telemetry was first developed for use in races by Honda in the mid-1980s, during the second era of participation in Formula One competition. Telemetry was used during the second era by acquiring important data on engine speed, engine oil and water pressure, engine oil and water temperature, fuel consumption, and the like while the car was passing in front of the pit, and using that data to manage conditions in the car. In the third era, by contrast, that system evolved to acquire and analyze all data concerning the engine and the chassis in real time. Figure 19 shows the history of change in communication speed and communication protocols used in telemetry.

In 2000 and 2001, the telemetry system developed during the second era was carried over. Its carrier wave frequency was in the 400 MHz band, the modulation method was Frequency Shift Keying (FSK), and the communication speed was 19.2 kbps.

A change in the regulations in 2002 enabled the transmission of data from the pit to the car, initiating the development of interactive telemetry systems. The communication speed in one direction was itself increased to 38.4 kbps. In 2003, however, the regulations changed again and interactive telemetry was outlawed.

In 2004, the carrier wave frequency was moved to the 1.7 GHz band, and the communication speed was raised as high as 460 kbps.

The communication speed was further raised significantly again in 2006. Of all the data collected while running, the items required for real time analysis amount to approximately 1000 channels. To transmit these by telemetry would require a transmission speed of approximately 1 Mbps. The demand for high-precision measurement described in section 4.3. required a maximum data rate of 1 kHz. In order to meet these demands, the chosen modulation method was Phase Shift Keying (PSK), which has high-power efficiency and frequency-use efficiency as well as low error bit rates at low receiving levels. In addition, the communication speed was increased to 2 Mbps, and a trigger function that increased the rate of acquisition for certain data at specified times was also implemented.

In 2006, as a result of the evolution in technology noted above, practically all the data from running cars could be acquired by the pit in real time. This capability became an indispensable part of race operations, as the data from the start of the formation lap could be used, for example, to change the clutch control mode as the race was starting.

6.2. Issues Associated with Increased Speed

Telemetry systems were checked for data acquisition performance in a simulated desktop environment. After that, circuit tests would be implemented; however, there were many points where testing on the actual circuit and desktop simulation indicated different characteristics. This was particularly the case when Quadrature Phase Shift Keying (QPSK) was adopted as the modulation method to increase the speed. In desktop simulation, the packet acquisition rate (the percentage of data packets

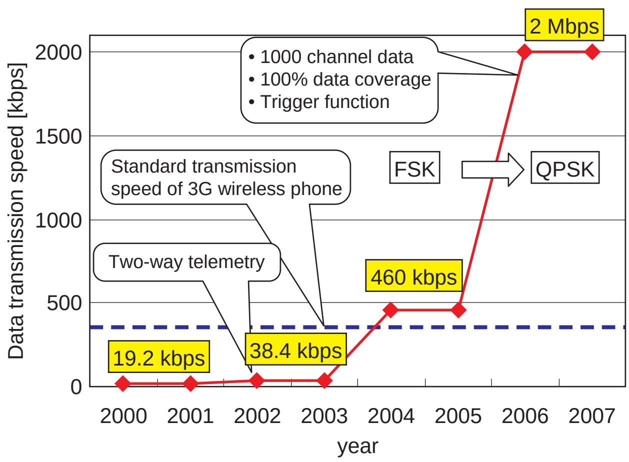


Fig. 19 Progression in telemetry transmission speed

received normally) was 90% or higher, but in circuit tests, the rate declined significantly. The data was being retransmitted, however, so the coverage (the percentage of necessary data acquired) maintained a level of about 100% (Fig. 20).

Figure 20 shows a conspicuous decline in the packet acquisition rate on the circuit, even in sectors that are relatively close to the pit. When data acquisition performance declines even at short distances, in other words, when reception levels are adequate, the cause is conceivably the influence of fading, defined as mutual interference on the receiving side from multi-pass waves, which are reflections from the circuit road surface, the stands, and the like. The extent of influence from fading

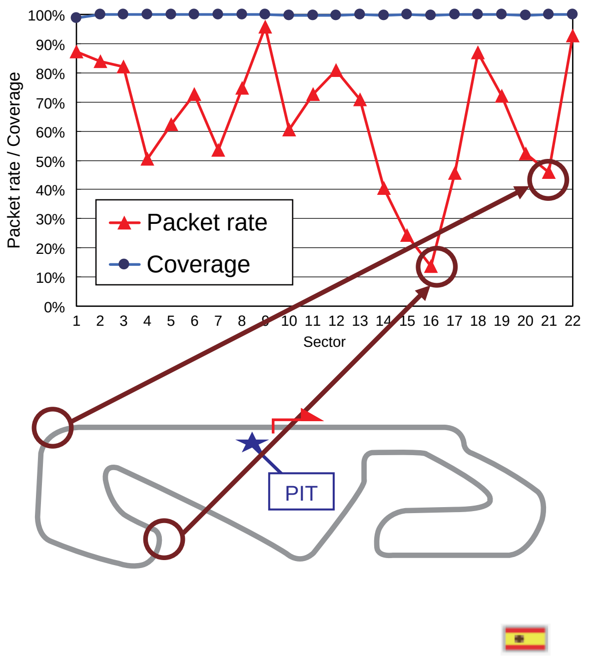


Fig. 20 Packet rate and coverage for circuit one round (2007 Catalonia)

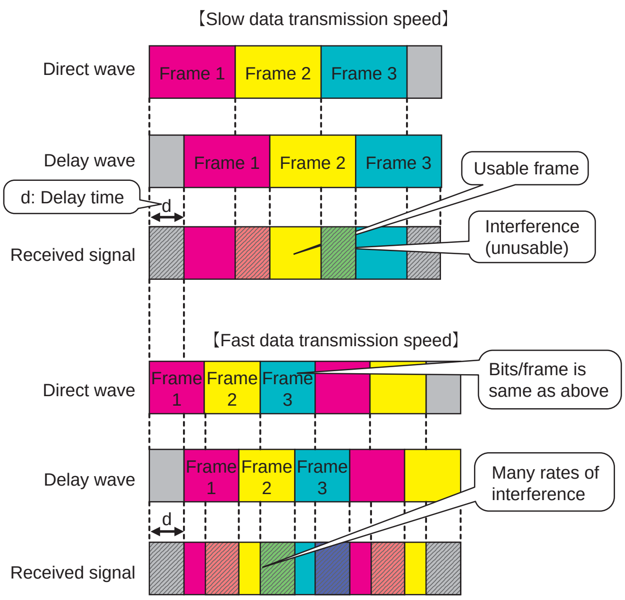


Fig. 21 Difference in influence of fading by data transmission speed

depends on communication speed. Figure 21 shows how the length of a single frame is reduced when the communication speed increases, so that the amount of interference is greater proportionate to the frame length. As a result, the influence of fading becomes significant and communication quality declines.

A variety of measures were tried to reduce the above type of decline in the packet acquisition rate, including:

(1) transmission power was increased to mitigate the effects of noise;

(2) transmission speed was lowered to mitigate the effects of multi-pass waves;

(3) a directional antenna was used on the receiving side in order to avoid multi-pass waves; and

(4) forward error correction (FEC) was adopted to lower the error bit rate.

Ultimately, however, these measures did not achieve any major reversal in the decline.

6.3. Consideration of a Radio Wave Propagation Model for the Circuit

As the development of telemetry depends on desktop simulation, an accurate grasp of the radio wave propagation environment on the circuit is of major importance. Consequently, telemetry development from 2007 on included efforts to raise packet acquisition rates on actual circuits by conducting investigations into radio wave propagation environments that resembled the actual environment.

The environment used for Formula One racing differs substantially from that of ordinary mobile communications (mobile phones and the like). Movement speeds can exceed 300 km/h at the maximum, there is no clear line of sight from the pit to the farthest point, and adaptive transmission cannot be used because the communication is one-way from the car to the pit.

Figure 22 shows the radio wave model (fading model) that was used in simulations up to 2007. In this model, there are two routes (paths) followed by radio waves, and the maximum delay time between paths was 0.2 μs. There is generally said to be a reflection delay time of 0.01 μs on land where the line of sight is clear, and a reflection delay time of 0.1 μs (a maximum of 0.2 μs) where there are buildings. This model is thought to diverge from the actual situation on the following two points:

(1) The path delay time is short.

The delay time of 0.2 μs represents no more than 60 m when converted into an optical path length. Considering that in the actual environment there would be reflections from distant stands, surrounding

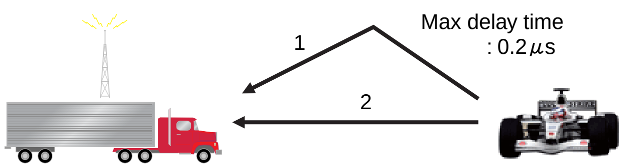


Fig. 22 Fading model used for development to 2007

mountains, and the like, it can be inferred that the path delay time would be longer.
(2) There are too few paths.

Since the telemetry transmission antenna is non-directional and transmits radio waves in every direction from the car, it can be expected that waves reflected from every structure on the course will be arriving at the receiving antenna at the pit, and the number of paths can therefore be conjectured to be greater than two.

Measurements made on site are thought to be the optimal way of ascertaining the radio wave propagation environment. As explained earlier, however, measurement on the circuit presents issues. Therefore, a variety of values were assigned as fading model parameters and the actual measured packet acquisition rate and the error pattern in data received with errors on circuit tests were compared with simulated tests. The following results were obtained:

- the maximum number of paths is six; and
- the delay times of multi-pass waves of dominant strength were about zero to 1 μs, and weaker multi-pass waves had a maximum delay of 5 μs.

It was found that the above fading model is similar to the actual environment (Fig. 23).

The above fading model is known as the Rayleigh fading model. Since the influence of overlapping multi-pass waves in large numbers can cause consecutive errors to appear in the data, little effect is generally felt from increases in transmission power or forward error correction. Furthermore, the existence of a large number of paths means that even the use of directional antennas will not fully avert the effects of multi-pass waves, and since the delay times are so long, decreasing the transmission speed of 2 Mbps to just one-half or one-quarter cannot be expected to resolve the issue of communication quality. When the model in Fig. 23 is viewed in this light, it becomes quite understandable that the measures described in section 6.2. to address the reduced packet acquisition rate did not yield any major effects to better the situation.

In development from 2008 on, the model in Fig. 23 was employed to conduct evaluations of new communication protocols to realize higher-speed communications while still maintaining an adequate packet acquisition rate even under this model.

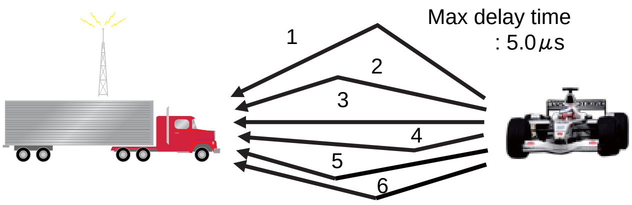


Fig. 23 Fading model used for development from 2008

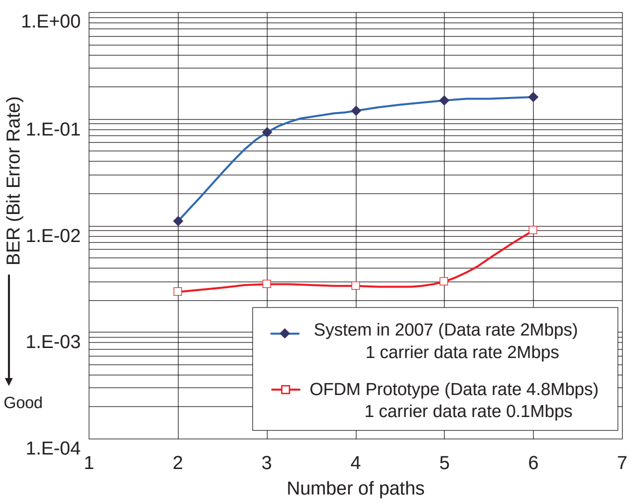


Fig. 24 Comparison of conventional system and OFDM

6.4. Technology for Next-Generation Telemetry

The aim from 2009 had been to do away with downloading by developing a system that uses real time telemetry to transmit all the data that has been logged by the ECU and the logger up to that time. Investigation was in progress with the goal of achieving further increases in speed. In order to seek still greater speed under present circumstances, it is important to adopt a multi-carrier system capable of reducing the transmission rate separately by carrier as a measure to increase fading resistance. Conceivable specific techniques to realize the aim of increased speed include the use of Orthogonal Frequency Division Multiplexing (OFDM) for enhancement of frequency-use efficiency and fading resistance.

A prototype for desktop use of OFDM was created and an evaluation of its characteristics was carried out using the fading model shown in Fig. 23. Figure 24 shows a comparison of the characteristics of the telemetry transmission equipment from 2007 and the OFDM prototype. The vertical axis in Fig. 24 represents the Bit Error Rate (BER: the number of bit errors divided by the number of bits transmitted) while the horizontal axis represents the number of fading paths. It can be confirmed that the adoption of OFDM has realized higher speed and higher reliability.

7. Component Development

As the Formula One engine and chassis have 100 or more sensors and actuators mounted on them, miniaturization of these devices was required with a view to mountability. Particularly, sensors mounted on the engine (for measuring pressure, temperature, timing, and throttle position), along with ignition coils and the alternator, were thoroughly miniaturized and made highly efficient. In addition, the greatest issue was the guarantee of their sustained functionality under engine vibration.

This article will provide details on the development of the following:
(1) throttle position sensor; and
(2) engine wire harness

7.1. Throttle Position Sensor

In the engine of an ordinary automobile, the intake flow is measured by the intake manifold pressure or by using an air flow meter, and that measurement is used to control the fuel injection volume. In the highly responsive engines used for racing and the like, however, the throttle opening is taken as the basis for setting the fuel injection volume. The throttle position sensor, therefore, is one of the key sensors in a Formula One engine, and requires high accuracy (within 1% at Full Scale) and reliability.

When Honda returned to Formula One racing in 2000, the brush type of contact sensor (potentiometer) was initially being used. There were various issues with it, however, including wear of the brush contact surface from vibration and foreign matter (such as oil or dust). As vibration from the engine reaches approximately 500 G, there was an urgent necessity to isolate the sensor from contact for the purposes of accuracy and reliability. From late 2003, therefore, this sensor was changed to a non-contact type using a magneto-resistance (MR) element to detect the magnetic vector. As this is the basic sensor for fuel injection control, it is necessary to provide high accuracy across the entire temperature range. An IC using two MR elements was adopted, and accuracy was assured by a technique that cancels out their respective temperature characteristics.

A split type sensor was adopted that has the sensing element mounted on the throttle body, and a magnet (polarized to generate the desired magnetic field) attached to the shaft of the throttle butterfly, which is a rotating body (Fig. 25).

This sensor was mounted toward the back end of the engine, and heat damage occurred frequently.

The main cause of heat damage was exhaust heat from the exhaust manifold. Due to the aerodynamic requirements of the chassis, the engine cowl was being squeezed to a smaller size every year. This had an impact on the flow of air inside the engine cowl, in addition to which the layout of the exhaust manifold was also changed accordingly so that it was closer to the engine where the sensor was mounted. The result was an increasingly harsh thermal environment. Even when the MR element and other IC parts are guaranteed at high temperatures, such guarantees ordinarily cover up to 130 °C. In order to guarantee up to 150 °C for racing, high-temperature solder and other such materials were used, while the range of guaranteed accuracy was restricted and adjustment made by means of software. Through these and other such measures, sensor accuracy was able to satisfy the condition of staying within 1% FS.

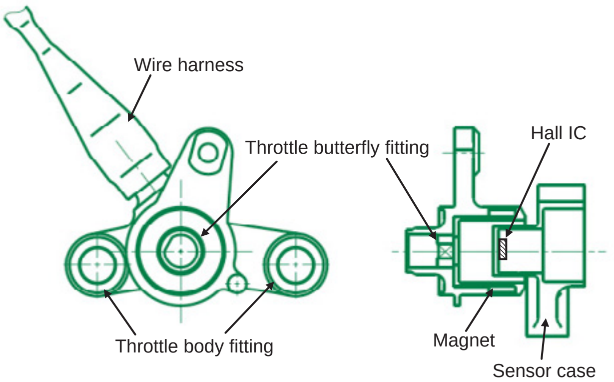


Fig. 25 Throttle sensor

7.2. Wire Harness

The Formula One wire harness was designed with an emphasis on reduced weight, durability, and on-site maintainability. Up until engine homologation was prescribed in 2007, there were no engine weight regulations. Consequently, measures were specialized in weight reduction, and the wire harness was fixed in place using simple stays that were mainly placed around the airbox and cylinder head. When engine weight regulations were instituted, there was a shift in orientation to maintainability and reliability rather than weight reduction. A junction box was placed above the cylinder head and the circuits for all the sensors were brought together in that box (Fig. 26).

In terms of reliability, it was a struggle against breaks in the wires. The top speed of a Formula One engine is double or more that of a mass production vehicle, and normal engine speed extends across the entire range, so that areas around the engine are subject to constant high-frequency vibration. In the case of a mass production vehicle, the wire harness is bunched together with tape or other such material and then encased in outer sheathing material (plastic tube) to counter wire breakage and wear at points of contact due to vibration. In Formula One cars, since the use of finer wires and reduction in weight are also important objectives, heat-shrink tubing made of woven polyester fiber is used to protect against external contact as well as to realize the use of finer wiring and lighter weight. Wire breakage resulting from tension load on the harness generated by vibration and acceleration has further been addressed by the use of silver-plated high-strength copper wire for racing use. The harness wires are then twisted together and bundled inside shrink tubes, enhancing harness

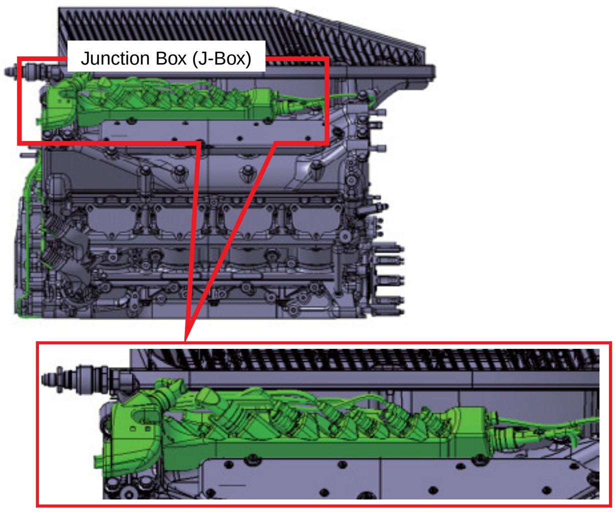


Fig. 26 Junction box on engine

strength against breaks (Fig. 27). This twisted structure not only enhances strength against wire breakage, it also heightens the flexibility of the harness itself and enhances engine mountability.

Even when these techniques are used, however, there are still places on the engine where vibrations at the level of several hundred Gs are generated. There have been cases, therefore, where just the copper wire has broken inside its cladding or inside the shrink tubing even when no external damage is visible. This is thought to be generated by sympathetic vibration of the copper wire or of the harness itself due to the high-frequency vibration, and the vibration is conjectured to result in displacement and tensile loads that cannot be absorbed. The method was therefore adopted of fixing the entire harness in place between the attachment points, including the oscillating parts, by encasing it in rigid retaining parts made of carbon material. This has succeeded in preventing wire breakage.

Vibration has been an unavoidable obstacle in the course of Formula One development. In the initial phases of development, it was important to grasp the levels and patterns of vibration in advance and to skillfully combine the measures taken to counter them. Another point is that the short Formula One development period allows limited opportunities to conduct the durability checks under engine operating conditions that are necessary to guarantee reliability. Consequently, it is necessary to ascertain the symptoms of faults and the effectiveness of countermeasures early on, and methods of analysis using X-ray and other such non-destructive inspection devices in durability testing have been introduced to the harness development process.

It has been necessary to manage the design effort so as not to interfere with weight reduction measures by pursuing vibration-resistant design that goes beyond what is required in actual vehicles.

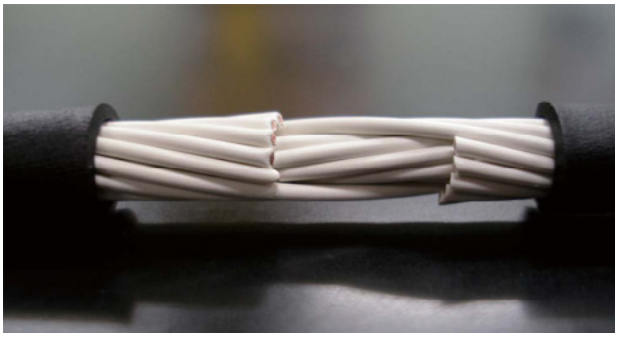


Fig. 27 Harness

8. Conclusion

In retrospect, Honda’s Formula One system development proceeded without break from the time of the test runs at Pembrey Circuit in the United Kingdom in 1999, one year after their development started, up to the Brazil Grand Prix in 2007, and a wealth of findings regarding new technology has been obtained. The electronic control systems developed for Formula One are for unique vehicles and are further specialized for racing functionality. However, the conceptual approach to system construction and the basic electrical system technologies that were employed are also likely to be held in common with the advancing automotive technology of the future.

It has been decided that the development of electrical and electronic technology for Formula One racing underway since the adoption of FIA standard systems will be superceded by the development of kinetic energy recovery systems (KERS), which is employed from 2009. It is to be hoped that Formula One will continue to stand as the highest achievement in automobile racing, as well as the highest achievement in automotive technology in times to come.

■ Author ■

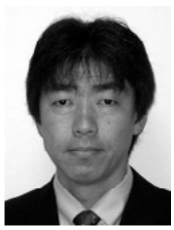
Kenichiro ISHII

Toshiyuki NISHIDA

Kohei TOSHIMA

Masaki NEGORO

Masataka YOSHIDA



Yutaka MARUYAMA

Development of Hybrid System for Formula One

Masataka YOSHIDA*　Masato KITA*　Hirofumi ATARASHI*

ABSTRACT

In 2009, the regulations for the Formula One World Championship were amended to allow the use of kinetic energy recovery systems (KERS). The new regulations stipulated that the KERS drive shaft be limited to the rear wheels, that output should be no more than 60 kW, and that the amount of energy used per lap be no more than 400 kJ. Honda had been conducting R&D in this area since the summer of 2007, and had developed a high speed, high output, direct oil-cooled motor, a water-cooled power control unit (PCU) which integrated a motor drive inverter unit and voltage control system, as well as a high power density lithium ion battery, all based on being small and lightweight enough for Formula One characteristics.

This system was first used to drive on straight roads in April 2008, and in May of that year Honda beat out other teams to conduct the world's first driving tests in an actual vehicle at the Silverstone Circuit, where the technology's superiority and high level of safety were proven.

1. Introduction

Honda has been developing electric vehicles, fuel cell electric vehicles and hybrid vehicles to find alternatives to fossil fuels, reduce emissions and mitigate the impact of automobiles on global warming.

As for the Formula One World Championship, the regulation amendments of 2009 allowed usage of KERS, which recovers and utilizes braking energy as drive power assist. To work with the new rules, Honda chose for its energy recovery method an electrical hybrid system and proceeded with development for use in Formula One based on the electric drive technology it had been developing.

Because the maximum output and the amount of assist energy that can be used per lap are stipulated by regulations, this development focused on making equipment as small and lightweight as possible with high output and high torque technology without changing the high level of dynamic performance unique to racing cars, and therefore Honda developed a motor, PCU and lithium ion battery capable of installation on a racing car. Honda additionally achieved high responsiveness to meet the requirement for output characteristics during racing.

Development began in earnest in the summer of 2007, and in just nine months, actual driving tests were conducted using the prototype vehicle RA1082 (a vehicle built to check functionality), and subsequently KERS was run at full power on a racing course for the first time in the Formula One environment. Based on the basic functions that had been confirmed with the RA1082, the technology then went into the RA1089 (race prototype) and then the RA109K (racing vehicle) by the end of 2008 (Table 1).

This paper recounts the development of the Formula One hybrid system.

2. Development Concept

Under the 2009 regulations, KERS could only be connected to the rear wheel, with maximum energy use per lap of 400 kJ and maximum output of 60 kW. The necessary performance targets set as development themes for this project were as listed below, not only to make the equipment compact and lightweight in order to be installed on a Formula One vehicle, but also to create a vehicle capable of winning races.

(1) System weight: no more than 30 kg
(i.e., no more than 60% of the vehicle ballast weight of the 2006 vehicle)

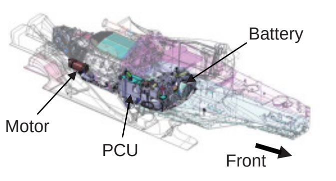
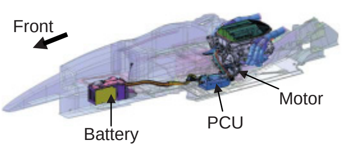
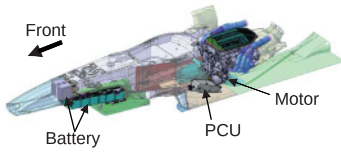
(2) Assist performance: at least 5 continuous seconds at output of 60 kW
(i.e., the output and assist time enabling the vehicle to overtake others)

When setting targets for performance, weight, center of gravity and the like for the various functional components, these parameters were investigated from many angles, including race strategy and the use of vehicle dynamics simulation, but some major concept-related issues were encountered when doing these investigations.

One was whether the race strategy should emphasize

* Automobile R&D Center

Table 1 Specifications of KERS

Vehicle code		RA1082	RA1089	RA109K
Shake-down		2008-APR	2008-NOV	2009-JAN
Motor	KERS power	60 kW	60 kW	60 kW
	Torque	45 Nm	45 Nm	45 Nm
	KERS energy	800 kJ	400 kJ	400 kJ
	Location	Transmission	Front of engine	Front of engine
	Dimension	ϕ100 x 202	ϕ100 x 190	ϕ100 x 190
	Max rpm	21000 rpm	21000 rpm	21000 rpm
	Weight	7.7 kg	6.9 kg	6.9 kg
	Cooling	Transmission oil	Engine oil	Engine oil
PCU	Power module	Si-IGBT/SiC-diode	Si-IGBT/SiC-diode	Si-IGBT/SiC-diode
	VCU type	Boost copper	Switched capacitor	Switched capacitor
	Operating voltage	680 V	560 V	580 V
	Weight	11.2 kg	8.0 kg	7.4 kg
	Cooling	Water	Water	Water
Battery	Type	Li-ion	Li-ion	Li-ion
	Cell number	114	108	106
	Weight	22.4 kg	21.2 kg	20.8 kg
	Cooling	Air	Air	Air
Layout				

overtaking or lap time. We compared single assist, which uses the 400 kJ all at once, or multi assist, which splits the maximum 400 kJ to use it on multiple instances; simulations showed that compared to single assist, a multi-assist system that settled the energy budget at each corner could reduce lap times by about 0.1 seconds. Figure 1 shows the time gain achieved on each circuit owing to the difference in type of assist. A multi-assist system allows smaller energy storage, so installation of a super capacitor with low energy capacity but great power density was considered. This would have the synergistic effect of having a lesser impact on the chassis. However, one cannot win a race unless one gets out ahead of other cars. For example, if assist is begun at 180 km/h, where the tire grip exceeds the drive torque (i.e., the tire is not skidding), assuming that the output of 60 kW will be used for 6.666 seconds (an energy equivalent of 400 kJ), vehicle speed can go 15 km/h faster than without assist, which in distance terms is a difference of 20 m. Even supposing that it were not possible to use assist for 6.666 seconds because of the course layout or other reasons, this would be a distance gain of 5 m (one car length) in 2.78 seconds and 10 m (two car lengths) in 4.22 seconds over another vehicle, which allows overtaking, so single assist, in this case, is more effective in terms of race strategy (Fig. 2). The opinions were thus divided on how to use KERS in actual races, so there was even some wavering on target requirements, but after several discussions with the Honda Racing Formula One Team (HRF1), the concept of emphasizing overtaking was ultimately decided upon.

The second issue was to maximize KERS's recovery of energy from braking while also maintaining drivability. Brake cooperative control, and the like is prohibited under the regulations, and therefore the amount of energy recovered varies not only according to course layout but also according to driving style, including such factors as how long and at what pressure

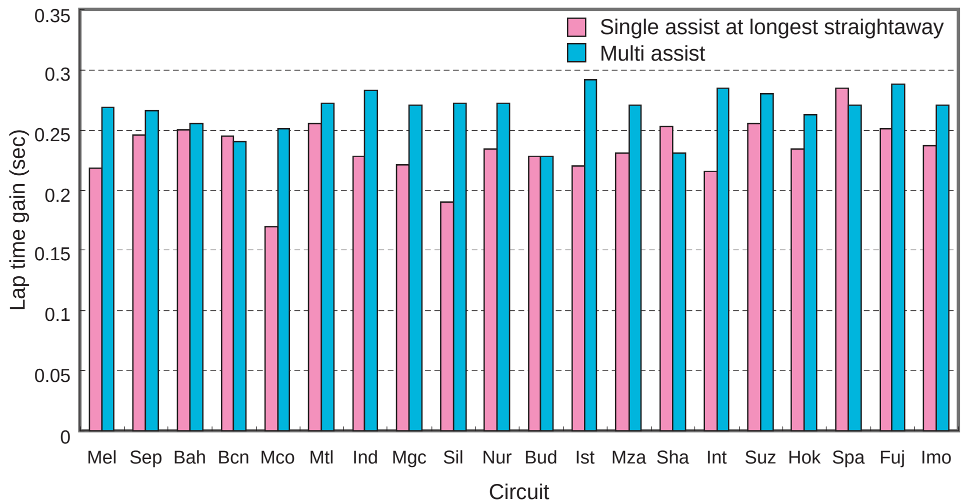


Fig. 1 Lap time reduction

the brake is operated. Additionally, simulations indicated that, depending on the circuit, it may not be possible to recover the target amount of energy. To address these issues and minimize the differences in driving style, we minimized loss from zero torque control that occurs during shift change and when offsetting motor friction, and furthermore achieved optimal allotment of engine braking and regenerative braking and made settings to let each functional component work with maximum efficiency in the range of usage. In order to get more regenerative energy, Honda also decided to propose to the Federation Internationale de l'Automobile (FIA) that we revoke a number of regulations relating to regenerative conditions, for example, disallowing regeneration if brake pressure is not below a set value.

3. Development of KERS

3.1. The Story of On-board Package Development

3.1.1. Development of RA1082 (functionality check vehicle)

The RA1082 was built to demonstrate the advantages of KERS in track tests and find any issues with the conformity of functional components to Formula One conditions. Since the PCU and battery are heavy components, using the RA106 (the 2006 race vehicle) as the base, we altered the area behind the driver's seat inside the monocoque (a CFRP body forming a cockpit) and placed these components there so that we would have little impact on the vehicle's center of gravity. Although placing high voltage components like the PCU and battery in this area has advantages in terms of the vehicle's center of gravity and electrical safety, at the same time it lessens the capacity of the gas bag (fuel tank) located in this area; the gas bag could now hold 83 liters, approximately 40% less than the earlier 140 liters. This was considered a racing strategy issue. As for the motor, originally it should have been directly installed with the engine close to the center of gravity, but because of the engine homologation (development freeze) in effect since 2006, no engine alterations were allowed and connections could not be made to the drive shaft, so the motor was placed inside the transmission case. Giving priority to minimizing the bulge of the case to avoid interfering with the aerodynamics to the rear, the motor gear was connected to the five speed gears of the lay shaft (the driven side shaft). The motor diameter of 100 mm was decided upon based on the installation fastening of the transmission and engine.

3.1.2. Development of RA1089 (race prototype)

We considered the issues of the RA1082 and began designing the RA1089 with the aim of minimizing the impact of KERS being on-board while maintaining a high level of vehicle dynamic performance. This vehicle also had an important position as the winter test vehicle (i.e., pre-season test vehicle) in anticipation of the coming 2009 season. The unprecedented installation layout, offering both vehicle performance and KERS performance, ran into troubles, and even once into chassis production, there were numerous design changes that put a burden on the production site. Moreover, engine changes to install KERS that were originally unapproved were later allowed after a request from Honda to FIA, so the installation of functional components was reviewed again. About that time, the

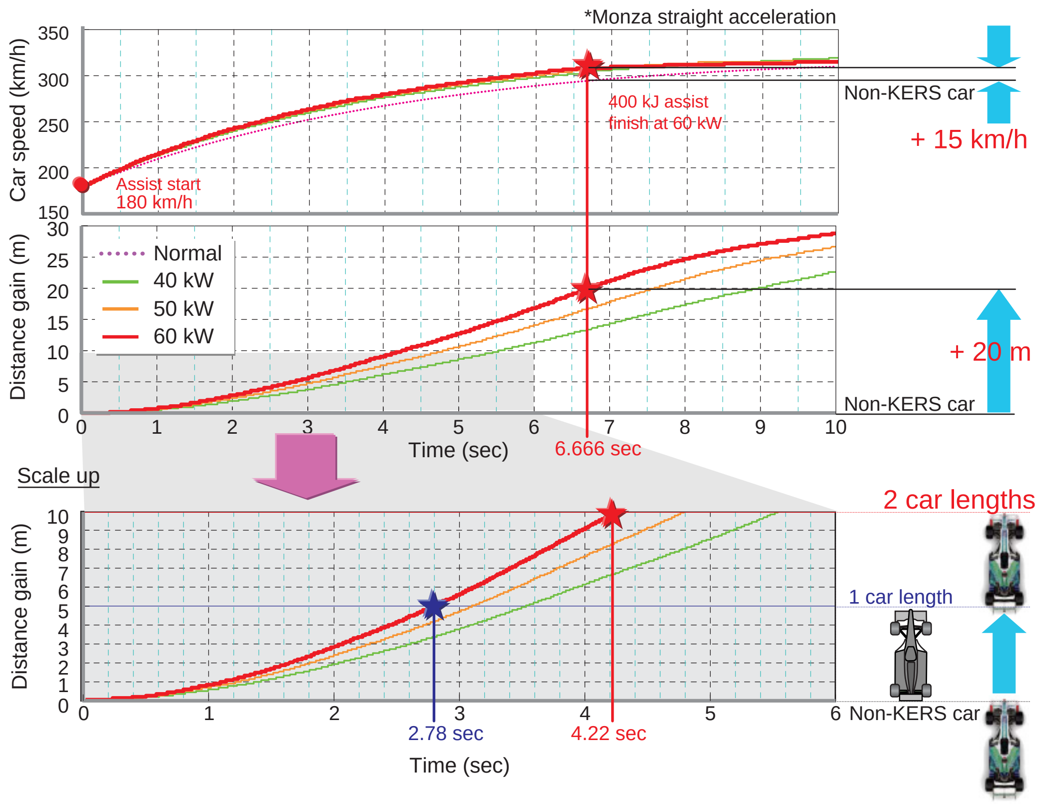


Fig. 2　Assist time / car speed and distance gain

time to decide the chassis layout for RA1089 was already coming soon, but Honda felt that vehicle dynamic performance was the number one priority and so, in a short period of time, worked with the HRF1 engineers to modify the engine and reinvestigate the cooling specifications of functional components and the installation position as determined by all items, including safety in the event of collision. As a result, from the many ideas proposed, it was decided to install the motor on the engine front side, the PCU in the left side pot, and the battery in the monocoque front keel. Table 2 shows a comparative investigation based on motor installation position. This decision not only made it newly necessary to have a motor gear train that connected the engine and motor drive shaft, but there was also an urgent need to ensure toughness against collision for the PCU and battery located outside the monocoque and provide for electrical safety with unprecedented techniques. It was decided to give these the highest priority and reinvestigate the specifications of functional components from the beginning. This put much pressure on the period of stand-alone testing for the PCU and battery, and meant that system launch would have to be taken place in a short period of time.

The next issue was the relative position between the gas bag and motor. If the engine's motor drive is transmitted directly to the crank shaft, it is necessary to put a recess around the gas bag, which is located in front of the crank shaft, of a volume equivalent to the motor. The motor was offset to the left side where it would not impact aerodynamics and the gear train was placed between the back end of the monocoque (the bulkhead) and the engine so the loss of gas bag volume was minimized as much as possible.

Because the functional components underwent completely new development, the challenging work, performed both in Japan and in the UK, dragged on for a long time, so the completion of the RA1089 was delayed. As such, the system check was not in time for the original target of the season's first winter test in November, and the target was changed to the second winter test.

3.1.3. Development of the RA109K (racing specs vehicle)

The RA109K was designed as the racing vehicle. KERS components were located in about the same position as in the RA1089, but in the RA109K, the periphery of the engine cowl, one of the aerodynamic components, was made smaller than in the RA1089, and it was necessary to review from the beginning the forms of the PCU and battery located inside the cowl. Aerodynamic performance is affected by wind tunnel testing time, and therefore aerodynamic components were first produced with a target of being ready for the opening race in March with subsequent updates following in turn. However, the specifications of KERS components cannot easily be changed because we take so long to produce and check for reliability. To ensure development of specifications meeting both these needs, the Japanese and UK bases proceeded with development 24 hours a day until the end of November. In particular, the battery module occupies the greatest volume and weight of all functional components and affects the vehicle's weight distribution and aerodynamics package. The decision on the placement of the battery disrupted the original timeframe for deciding on specifications, but ultimately it was decided to use a dispersed placement, with the battery module located in the keel and also in the nose, a position which until then had not been approved. However, the FIA collision regulations are very strict as far as the nose is concerned, so HRF1 conducted simulations and bench collision tests with the collision regulations to prove the layout's safety and finally earned the FIA's authorization (Fig. 3).

Table 2 Comparison of motor position

Motor position	Mass	Front-weight distribution	C.O.G
Engine front - LHS	8.69 kg	-0.60%	+1.0 mm
Gearbox	10.30 kg	-1.13%	+1.4 mm

3.2. Overview of KERS

This section discusses the RA109K vehicle intended for races.

Figure 4 shows an overview of the Formula One hybrid system. The motor drive shaft is connected to the engine's cam gear train through the five-gear train for the motor. The motor and PCU were on an anti-vibration mount on the left side of the monocoque, with the motor cooled by engine oil and the PCU by special cooling water. The battery was fixed to the front of the monocoque, and the temperature controlled by the draft air from the front. Besides these modifications, the PCU was connected to the FIA standard ECU (S-ECU) by

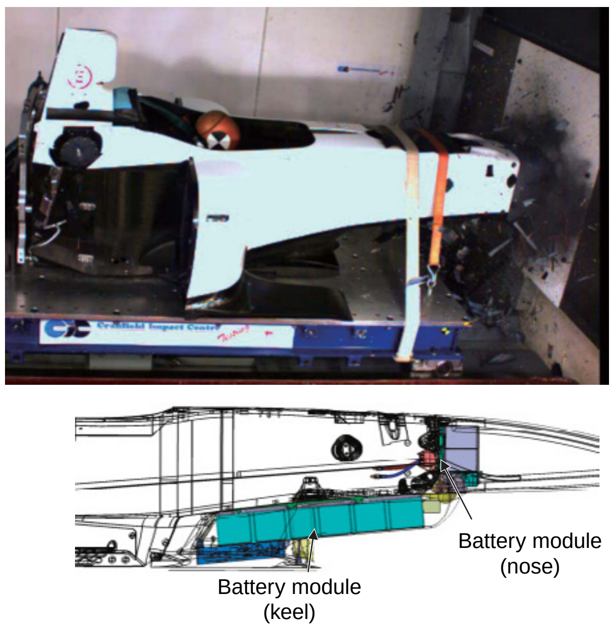


Fig. 3 Front collision test for FIA regulations and battery layout for RA109K

CAN communication for mutual commands and monitoring. Assist and regeneration were controlled in linkage with the driver's operation of the vehicle.

In Formula One vehicles, the vibration environment is more severe than in mass-produced vehicles, so the PCU and battery were built so that even at a vibration of 20 G, there would be no issues in terms of functionality. The motor is required to be able to withstand even greater vibration, because it is exposed to engine vibration, but since this surpasses the capacity of the vibration generator, no stand-alone evaluation was performed; instead, reliability was judged by checking on an engine bench and in track tests. Moreover, in light of collision safety, a structure was chosen that met the FIA's side impact requirements.

3.3. Motor Gear Train

A five-gear train was used as a means of connecting the crank shaft with the motor, which had been moved to the left side of the monocoque. The gears engaged with the engine's cam gear train to connect to the crank shaft (Fig. 5). These KERS-specific parts were made lighter by the use of magnesium covers, titanium bolts and ceramic ball bearings, the use of which is prohibited in the engine itself under the regulations.

There was already only a few mm of clearance between the engine and monocoque, and to put the gear train here required creating some space by putting a recess in the bulkhead. Such a recess diminishes the rigidity of the monocoque and the capacity of the gas bag, so the layout characteristics were enhanced by integrating the gear train housing with the engine front cover. The gear train was in a parallel configuration of five gears with a height of less than 30 mm, but this was the our first experience with a gear train of this form and a technique transmitting torque in both directions, so during development there were frequent issues, such as broken shafts because of resonance in the gears. After subsequent changes of specifications in fine areas and repeated durability testing, we were able to ensure durability in time for track testing.

3.4. Motor

For a motor to be adopted on a Formula One vehicle, it should of course be compact and lightweight and offer high output, and in addition high efficiency is a crucial factor for ensuring enough energy for overtaking; it is no exaggeration to say that this factor determines victory or defeat. A brushless DC motor was therefore used to achieve both of these factors at a high level. From the point of view of installing the motor in the vehicle, as stated before, the diameter was designed to be within 100 mm and the full length within 200 mm, but calculated from required output, the power density is at least 8 kW/kg, which is a far more severe requirement

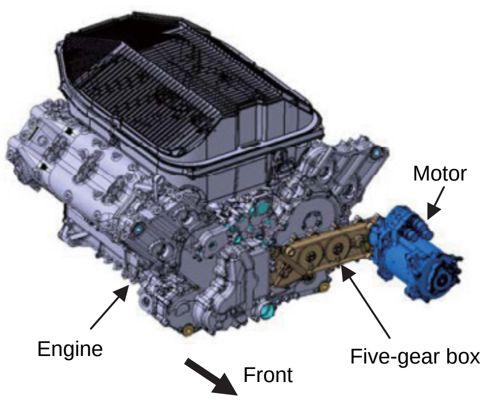


Fig. 5 Five-gear box for motor

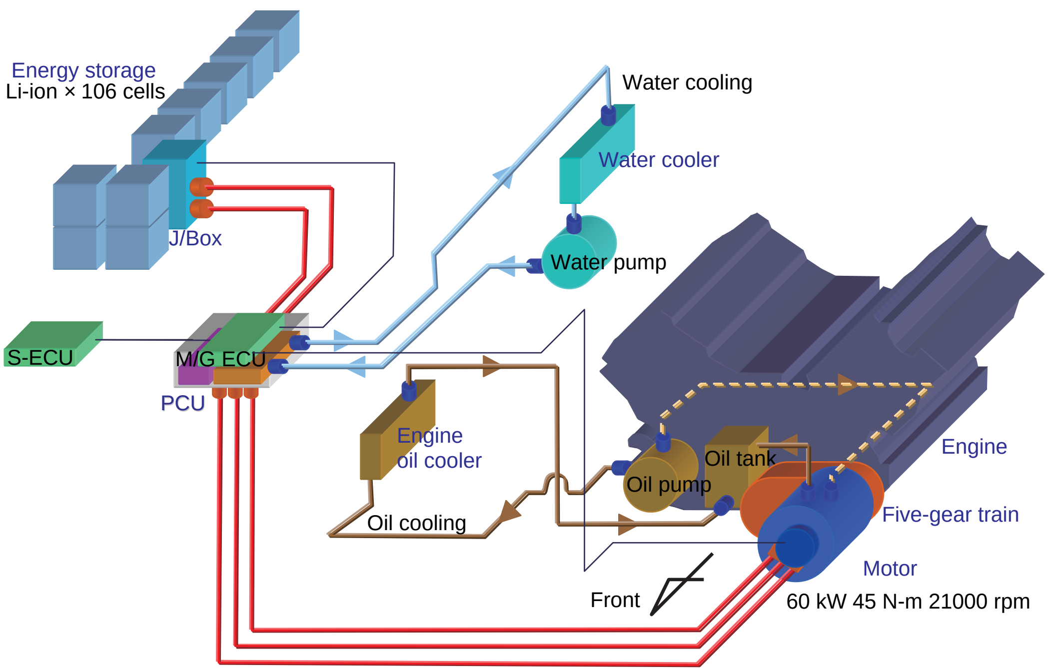


Fig. 4 Hybrid system

than with mass produced motors. Furthermore, we ran KERS simulations with the driving data from past race, and set the requirements that the motor could assist and regenerate up to 60 kW at 13000 rpm and above so that the system could ensure the minimum amount of regeneration on any circuit. To do this, we reviewed the entire motor design, including development of new electromagnetic steel plates and magnetic materials, review of winding techniques and using new cooling techniques and rotor structures, with the result being the achievement of a high power density of 8.7 kW/kg while maintaining high efficiency (Fig. 6).

The rotor used a newly developed high coercive force magnet (iHc = 1.1 MA/m at 160°C) and at the same time, in order to achieve high revolution speed of more than double that of earlier motors, it used filament winding with organic high strength fiber to prevent magnet scattering, thus protecting the magnet circumference. We additionally enhanced torque by 8% by setting a magnetic field angle of θ in, θ out to concentrate the magnetic field orientation of the magnets to the polar center, and divided and stuck the magnets along the length of the shaft to mitigate temperature increases resulting from eddy current loss. For the electromagnetic steel plate making up the stator, on a base of iron and cobalt (49 Fe-49 Co-2 V), we succeeded in reducing eddy current loss by making a 100 µm thin panel, in reducing hysteresis loss with a past-rolling heating treatment, and in enhancing the volume fraction with an oxidized insulation membrane. As a result, saturation magnetic flux density was enhanced by 30% and torque by 15% while iron loss was reduced by 60% as compared to a conventional 200 µm silicon steel panel (Fe-Si). Lap winding was used for the stator winding for high torque and low loss; connecting parts (turnaround parts) at both ends of the stator were press-molded, using injury-resistant copper wire, achieving an unprecedented low connecting (turnaround) height (Fig. 8). Furthermore, because smaller motors made it impossible to keep the coil within the tolerable temperature during driving of the motor in conventional water-cooled jackets with stator housing, a stator structure was used such that the coil ends were directly cooled by engine oil (Fig. 9), so no special radiator was needed. Additionally, the agitation of cooling oil with the rotor increases friction, so a cylindrical cover was used to prevent cooling oil from sticking to the rotor, and the oil chamber of the stator side was completely sealed. As a result, the motor’s stand-alone efficiency averaged 95% at an average motor speed of 20000 rpm during assist and 93% at an average of 16500 rpm during regeneration, thereby achieving both high output and high efficiency (Fig. 10).

Fig. 6 Formula One KERS motor

Fig. 8 Core assembly

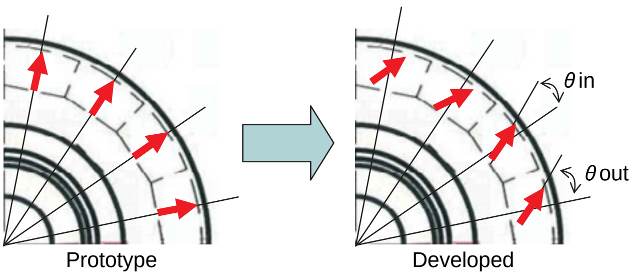


Fig. 7 Direction of magnetization

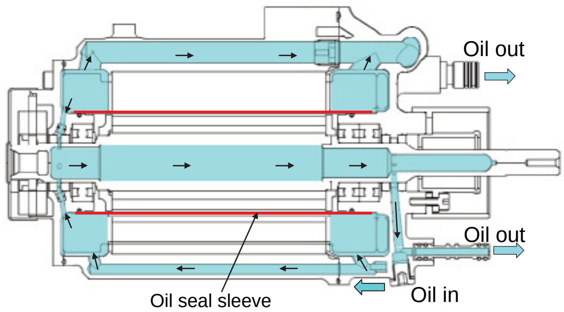


Fig. 9 Section of motor with cooling oil flow

3.5. PCU

The PCU consists of an inverter unit (PDU) to drive the motor and a voltage control unit (VCU) that allows the voltage to be raised or lowered freely. The PCU is used to supply the optimal amount of voltage and current from the battery to the motor and to recharge the battery, and it makes a great contribution to increasing motor efficiency while making it more compact. An intelligent power module (IPM), installed in the PDU, used a special design to enhance compactness and reduce electrical loss, while an SiC diode was used on-board for the first time to reduce flywheel diode noise.

Development of the VCU started out with a boost chopper form, but the PCU also faced severe on-board installation requirements and had to be made smaller and lighter, so system operation frequency was increased and the form was changed to switched capacitor, which offers the potential for a smaller, lighter reactor (voltage step-up coil) (Fig. 11). An effort was made to make the reactor smaller and lighter, not only by increasing the system operation frequency as previously stated, but by forming a 3D core by using dust core materials. However, there is no history of using a switched capacitor system on-board, and the technology was still at the level of proving the principles behind it, so it was difficult to establish the technology, and we were pressed to take measures until just before moving to the circuit for winter testing. The result of their effort was that the PCU was 3.8 kg lighter than that of the RA1082.

The PCU of the RA1082 was located inside the monocoque, but starting with the RA1089 it was hurriedly put outside the monocoque, so that it faced new collision requirements. To meet both the need for lighter weight and collision requirements, we considered changing the aluminum case used up to that point to a CFRP case. In addition, the form of the PCU case was reviewed every time an aerodynamic cowl component was changed because the inside of the cowl, which prioritized aerodynamic performance, had very little installation space.

Under the initial specifications for the RA109K PCU, the unit was cooled by a special cooling water circuit, but to ensure high heat tolerance and low loss, we developed a power module using an Si-C MOSFET (metal oxide semiconductor field effect transistor), and a new heat spreader (silver and diamond compound), thereby creating a smaller, lighter PCU that shared the engine's cooling water; the plan was to release this at the British Grand Prix, midway through the season.

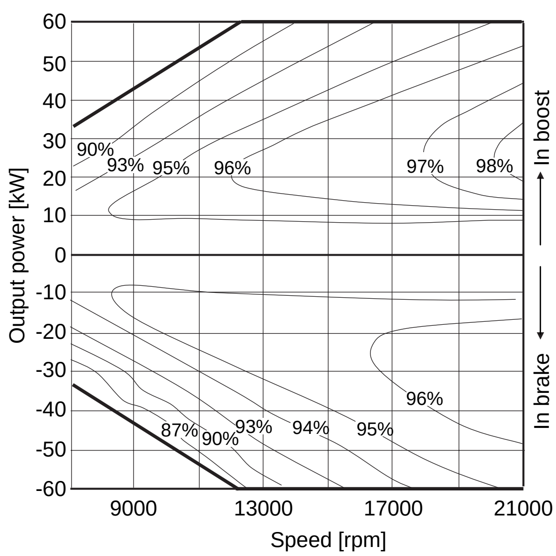


Fig. 10 Motor efficiency

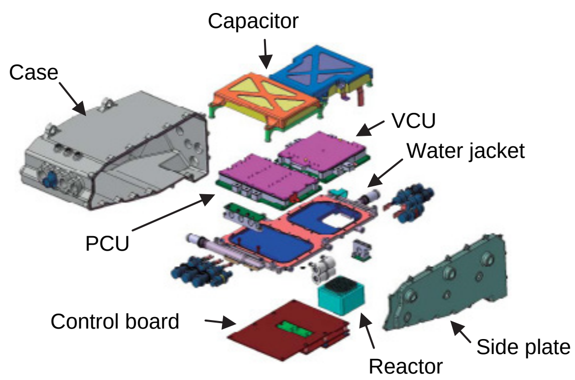


Fig. 12 View of PCU for race specification

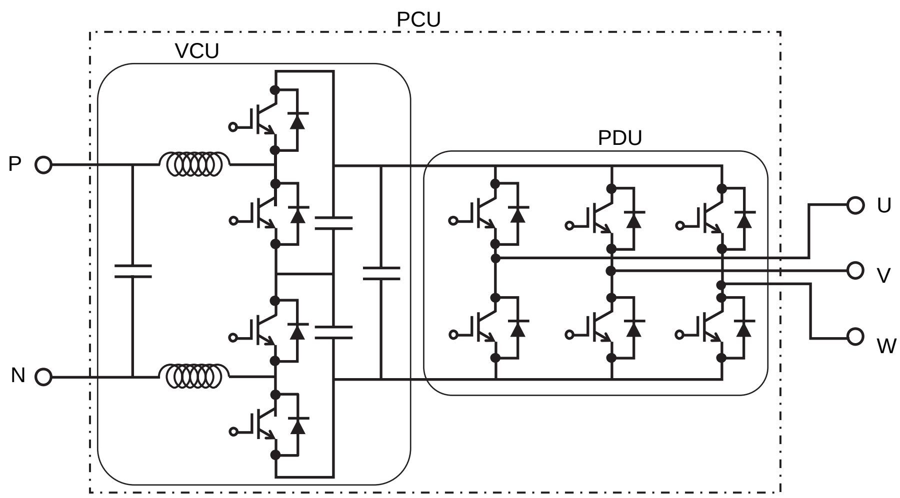


Fig. 11 Race specification PCU block diagram

3.6. Energy Storage

In order to meet the targets for the Formula One KERS, targets for energy storage (ES) were set as below, taking system efficiency into account.

(1) The system should have at least 70 kW at the ES output end and at least 500 kJ of actual charge/discharge energy.
(2) To ensure collision safety, boxes should have at least as much strength as required in the FIA side impact test, with the internal construction having enough strength to withstand at least 100 G-force.

The battery accounts for a high percentage of the weight among KERS functional components, and to get an output of 60 kW, a conventional lithium ion battery weighs at least 30 kg, which reduces the competitiveness of any race vehicle in which it is installed. For that reason, a battery emphasizing output was newly developed based on a lithium ion battery undergoing R&D at that time for ordinary market vehicles. The enhancement of power density continued until just before the system for providing vehicles for the 2009 race season was established. Ultimately, the power density advanced from approximately 5000 W/kg under the RA1082 specifications to approximately 6000 W/kg for the RA109K. This is about five times the power density of a typical Ni-MH battery and about twice that of a typical lithium ion battery (Fig. 13).

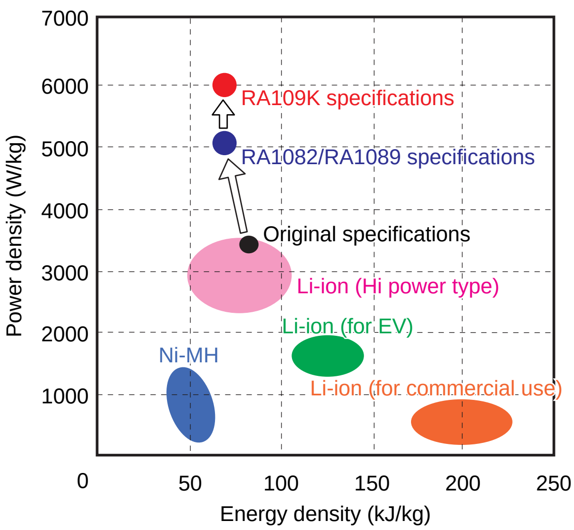


Fig. 13 Battery specifications

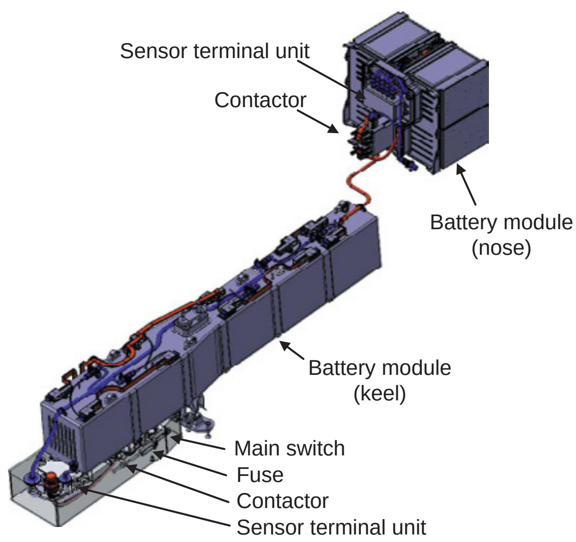


Fig. 14 Battery pack with control unit

On the other hand, the battery of the RA109K was split for installation into two areas, the nose and the keel, to meet the collision requirements. To minimize the impact on vehicle weight, the cooling system depended on draft air from the front of the vehicle; for waterproofing, cell connections and high voltage terminals were molded when building the single module. Furthermore, sensor terminal units, which constantly monitor cell temperature and voltage, were placed in each battery box (Fig. 14).

3.7. High Voltage Safety

Honda's basic stance on high voltage safety is that high voltage components should be placed in an area that is unlikely to be crushed even in the event of collision. In races, however, for reasons of strategy and vehicle packaging, KERS high voltage components were placed outside the monocoque and could therefore potentially be crushed. To ensure high voltage safety under these conditions, we decided that it would be necessary to actively announce our proposal for safety measures as based on mass production experience when implementing KERS, and Honda put together its own safety stance prior to the FIA's move to implement unified safety standards. The high voltage power source in KERS consisted of the following three units.

- High voltage battery
- PCU capacitor (when recharging)
- Motor (counter electromotive force is generated when the motor is turned by external force)

To ensure high voltage safety, it is important that KERS unit and cables be securely insulated. Furthermore, if it is possible that the insulation can be destroyed by accident or contact with another vehicle, it is necessary that the supply of high voltage from the above three units definitely be cut off. Consideration also needed to be given to removal of the insulating structure (the covers, case, and the like) when performing maintenance. It is also important to provide a monitoring function (ground fault detection). Based on the above perspectives, safety was ensured with the specific structure below.

3.7.1. Safety during collision

(a) Destruction of the insulating structure (case and covers) by loads anticipated during collision was prevented. Specifically, enough strength was ensured that the insulating structure would not be destroyed under conditions equivalent to those of the FIA's side impact test.
(b) In the following cases, the electric charge in the PCU capacitor was discharged at the same time the main contactor was cut off.
  - When the FIA driving data inspection record unit that

is connected to the S-ECU has detected acceleration beyond a set value
- When the acceleration sensor on the PCU control board has detected acceleration beyond a set value

Specifically, the electric charge was shifted between the VCU's two capacitors by VCU switching, and electric energy was changed to heat and released by switching loss and conduction loss.

(c) A circuit that discharges electric charge from within the PCU capacitor if the control power source supply is cut off was located on the control board.

(d) A structure was used such that the high voltage connector keep plate would be destroyed if more load than anticipated were applied. The load required to destroy the plate was set to be smaller than the load to break cables, so that in the event of collision, connectors would come off before high voltage wiring breaks (Fig. 15).

3.7.2. Safety during maintenance

(a) The electric charge in the PCU capacitor was discharged at the same time the main contactor was cut off, when the ignition was off.

(b) A structure was used such that an interlock mechanism was used for the DC cable connector and high voltage would not be applied to terminals when high voltage connectors were removed. To remove or put on the high voltage connector cover, one had to remove the control harness, and if the control power source were cut off, the PCU would discharge and not apply high voltage to connectors. A similar structure was also used on the battery side to ensure safety.

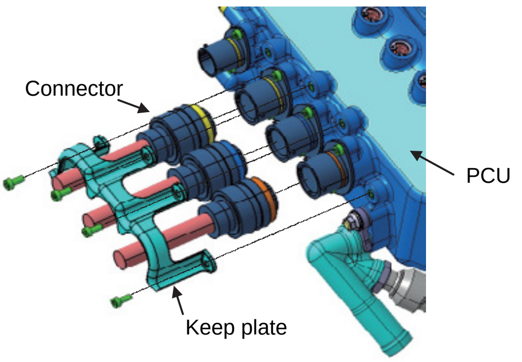


Fig. 15 Structure of high voltage connectors

3.7.3. Ground fault detection system

A detection circuit was mounted on the PCU control board, and during KERS operation (i.e., when the main connector is connected), the high voltage area ground fault detection system was constantly put into operation. Safety was ensured by making a warning lamp turn on and simultaneously cutting off the main connector when a ground fault is detected.

3.8. Control

KERS control system consisted of an S-ECU control and PCU control (Fig. 16).

The S-ECU control calculated electric power commands based on driver operations and monitored motor output and per-lap assist energy. Assist occurred when the driver pushed an assist button with the accelerator pedal fully depressed, while energy regeneration only occurred when braking. Motor output was measured using values from torque sensors, and other values, and if it was determined that a violation had occurred, an output restriction penalty lasting a few seconds was imposed. Assist energy was calculated from the total motor output, and assist was stopped when 400 kJ was reached.

To use KERS to its full capacity, one has to use the 400 kJ efficiently and recover enough energy to make that possible. Trial calculations indicated that to get 400 kJ of assist, one needs about 500 kJ of regenerative energy, taking into account such factors as loss depending on system efficiency and the energy consumed by zero torque control. Because carbon disk brakes have good brake force and can slow down vehicles quickly, it is conceivable that if regenerative torque is added, brake force will be unstable and stability will decline, and we predicted that depending on the circuit, road surface conditions and other factors, it may not be possible to perform regeneration sufficiently and thus there would not be enough energy. We therefore

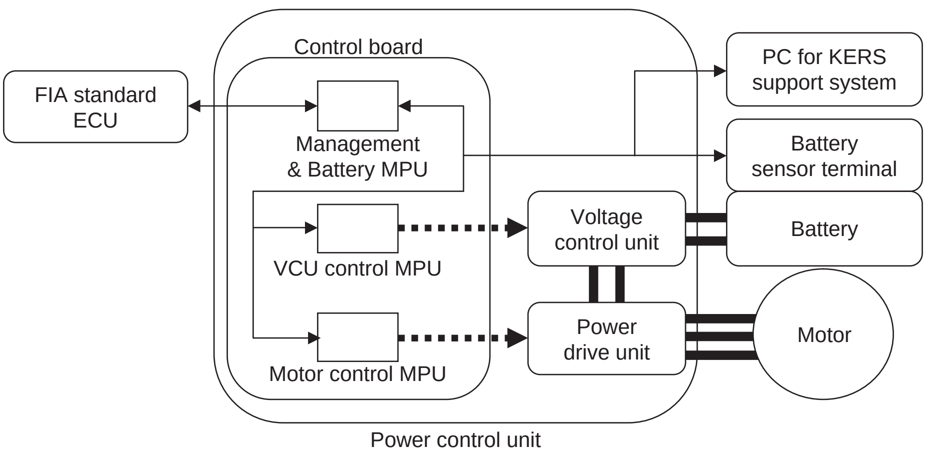


Fig. 16 Control system configuration

optimized each control component, including cooperative control, and also optimized control of each functional component by incorporating the various simulation results into specifications. Through track tests, we also endeavored to optimize the front-to-back brake balance and distribution of regenerative braking and engine braking. These tasks all require a high level of balance, and track test settings were predicted to be difficult because we are subject to S-ECU function limitations and annual mileage limitations, but drivers were not demanding any enhancements and potential issues were considered unlikely for initial specifications.

It was decided that the operation of starting assist, which is entrusted to the driver, would take effect at the lowest vehicle speed, at which excessive driving force would not result in wheel spin. This was to simultaneously prevent both an unnecessary increase in the driving force in low gear due to motor torque and a decline in assist efficiency resulting from an increase in driving resistance. Acceleration by assist is also affected by gear ratios and aerodynamics settings, so the amount of time and energy needed to overtake on each circuit was calculated and the effectiveness clarified, and results were used effectively in determining ES capacity and designing how each functional component would be cooled.

In the PCU control, each functional component was controlled based on electric power commands from the S-ECU control. This consisted of PCU management control, motor control, VCU control and battery control, and the necessary functions were created for racing use based on existing specifications.

The system operations such as system stop and output stop or power save were determined using PCU management control, based on mediation between electric power commands and system status.

Motor control and VCU control controlled motor output. Because of the motor's connection to the gearbox in the RA1082, it was necessary to have some control so that torque would be lost instantly (torque loss control) when downshifting. When downshifting, on the other hand, energy regeneration was performed while decelerating, so torque loss control, during which energy could not be regenerated, should be applied in as short time as possible. If responsiveness were to be within 50 msec as stipulated in the regulations, the time it would take to recover from torque loss control would be wasted, since energy could not be regenerated then, so we targeted responsiveness of within 15 msec to reduce wasted time. It was decided that electric power commands would change step-wise, at each change of gear, from -60 to 0 kW or from 0 to -60 kW, and we had to ensure controllability during great fluctuations of revolution speed and load. For motor control, current feedback control was performed using vector operations, but the behavior of field component current in response to step-wise electric power commands could not be fully controlled and excessive current sometimes occurred. For VCU control, output voltage feedback control was performed, but this arrangement could not keep up with sharp load fluctuations and excessive voltage frequently occurred. The countermeasures against these issues continued until just before track tests, and we were able to achieve both stability and responsiveness to the target of being within 15 msec by enhancing their techniques for calculating target current values with motor control and by applying voltage with feed-forward items in both control parts.

In addition, we unstintingly added the technological elements deemed necessary to the pursuit of speed, such as performing zero torque control during ordinary driving (without assist) with motor control to prevent interference with engine acceleration. We furthermore developed one-pulse control, which set the former pulse width modulation (PWM) duty to 100% and reduced the number of times switching occurred in order to minimize PDU loss, and also developed VCU variable-voltage control to maximize motor efficiency, with the aim of releasing these developments during the 2009 winter tests. These systems were also subjected to cooperative control and systems were optimized simultaneously.

For battery control, we made it possible to use lithium ion batteries safely and efficiently by monitoring battery voltage and temperature as found at the sensor terminal and accurately determining charge status as based on calculation of the amount of electric power charged and discharged.

4. Bench Tests and Track Tests

4.1. Bench Tests

Two Formula One engine test benches were used, to check the system and its durability. A battery simulator (BTS) and inverter stand (Fig. 17) were implemented along with these benches to make them KERS-compatible. A completely new BTS was implemented to deal with a level of responsiveness unprecedented in mass produced vehicles, making it possible to test the overall system with all parts assembled (the engine, motor, PCU, battery, and other parts) in coordination with development of the RA1089 without experiencing troubles under transient loads equivalent to actual driving.

In this way, we substituted the bench for all testing from confirming operation of each functional component of KERS system to confirming the functionality of the

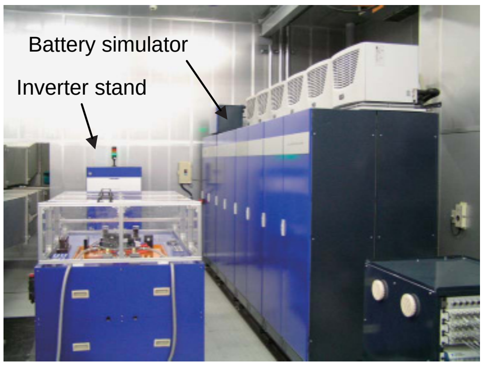


Fig. 17 Battery simulator and inverter stand

overall system on an actual vehicle, including functional assurance testing, and sought to bring KERS as a whole to early fruition without the track tests that are so severely restricted under the regulations.

Figure 18 shows the basic form of the test bench. KERS was mounted with the engine as part of the jigs. Using bench testing equipment, miscellaneous evaluation tests were performed as shown in the events of Table 3, under steady state or transient load conditions. For the transient load tests on the bench, the system underwent mode operation, reproducing actual driving conditions based on circuit data from actual driving experience, and transient performance evaluation, particularly energy management, measurement of the temperature of each part, and the like, was performed, making it possible to acquire useful data and obtain feedback in order to speed up development. Figure 19 gives a typical example of mode operation.

The data in the figure was acquired with the following combination. This experience demonstrated that it is possible to conduct function and durability bench tests before actual circuit driving.

- Driver: Jenson Button
- Circuit: Monza
- Car: RA1082 + KERS

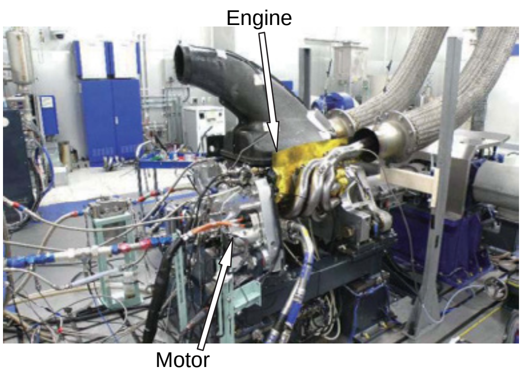


Fig. 18 Combination examination of engine and motor

Table 3 Testing items on engine dyno bench

Function test	-System operation / setting -System output -System efficiency -Fail safe action -Temperature measurement -Vibration measurement -Gear behavior -Oil pressure measurement -Tool operation check
Durability test	-Engine and KERS parts durability test -Energy management

With the RV (real vehicle) bench [(1)], furthermore, it was possible to mount an actual Formula One monocoque and cooling system on the bench and perform a variety of tests in an actual vehicle environment in addition to the above power train test bench environment (Fig. 20). The RA1082 was actually placed on the RV bench so that countermeasures for electrical noise that was unique to KERS were completed before driving on a circuit. Admittedly, because of the

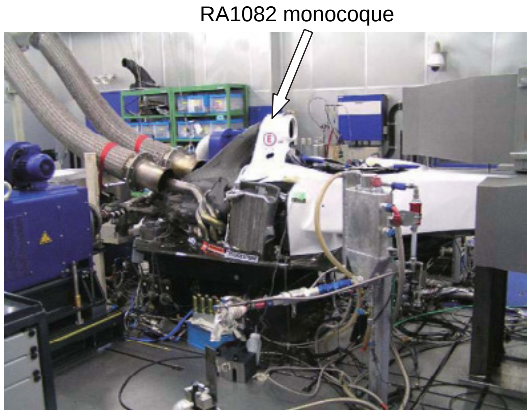


Fig. 20 Bench examination using RV bench with real Formula One KERS car

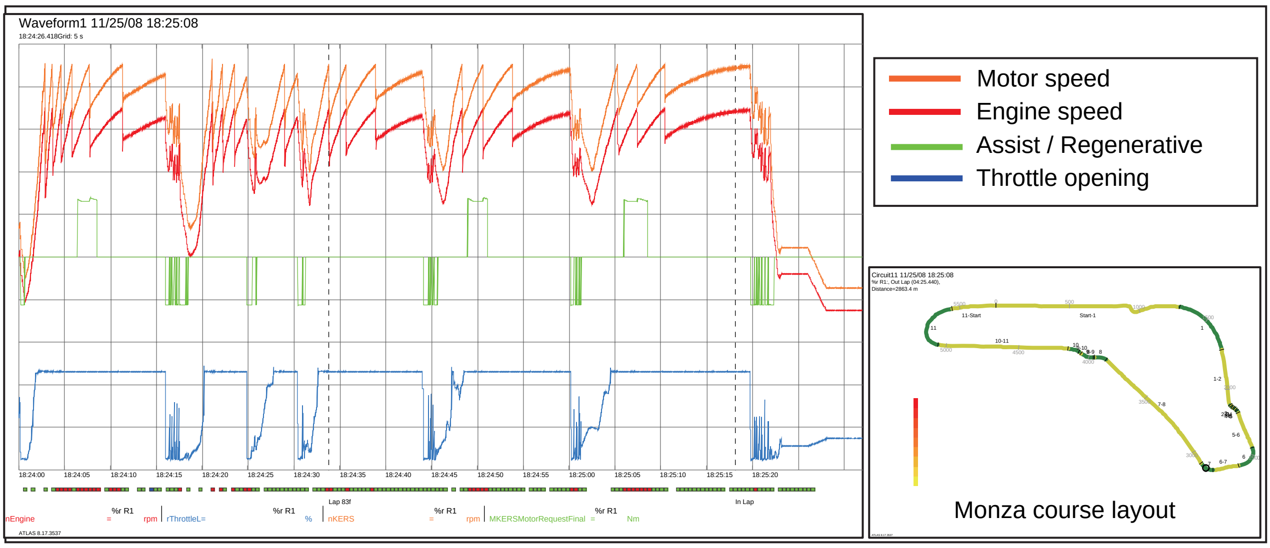


Fig. 19 Circuit mode test on dyno

Table 4　Durability test results of dyno

Event	Purpose	Mileage
KE-01	Check environment and first issue parts	238.4 km
KE-01.5	Check track-1 specification parts	465.4 km
KE-02	Track test simulation (for Jerez circuit)	611.0 km
KE-03	Check track-2 specification parts	748.8 km
KE-04	Check race specification parts	1351.9 km
KE-05	Check engine gear durability with extreme circuit mode (test stopped due to schedule)	566.1 km

production schedule, the RV bench was not used with subsequent vehicles, but power train durability was tested on the bench prior to circuit driving and mileage equivalent to more than two race events (1350 km) was assured with all functional components (Table 4).

4.2. Track Tests

In late April 2008, HRF1 conducted the shake down of the RA1082, the first machine with KERS, and subsequently conducted actual driving test four times with the RA1082 and twice with the RA1089.

RA1082, the first actual vehicle, underwent modifications and the vehicle was finally complete four days prior to shake down, which cut deeply into the originally planned two weeks of system checks, and with vehicle settings, additional modifications, and the like, there was in effect a preparation period of only about two days. The track test members had to deal with the system operations check and initial troubles in a short period of time and were not able to get to complete operation by the prescribed date. For that reason, in the initial shake down we decided to maintain the electric current status, and to limit themselves to checks of system safety and the function of functional components. At HRF1, we were not able to confirm system operation by chassis dyno as with the RV bench, so all we could do was to check system operation by firing-up the engine in a factory. However, to fire-up the engine required a large number of engineers and mechanics and a launch sequence starting several hours in advance. For the sake of engine durability, moreover, it is not possible to let it run for long periods with no load, so we could only get the desired data in a very short period of time, and it was difficult to do system check of KERS. Ordinarily fire-up only takes place once or twice before shake down, but fire-up was conducted dozens of times with the RA1082. The first shake down was not done under the maximum load allowed by regulations, but the run was memorable for being the first KERS actual driving test among all 10 Formula One teams. At the factory, we subsequently responded to nonconformities found during shake down, and a week later conducted output and regeneration tests on the Silverstone Circuit, confirming that KERS functioned effectively under Formula One conditions. The team subsequently made more enhancements, conducted a private test in July and took part in a joint test at the Jerez circuit in Spain in September, thus beginning the first regular run on a circuit (Fig. 21). The benefits of KERS were verified at the maximum load under regulations at the Jerez circuit; the following benefits were confirmed.

(1) As Fig. 22 indicates, lap times were reduced by approximately 0.4 seconds when assist with energy of about 400 kJ per lap was applied.

(2) As shown in Fig. 23, car speed increased by 7 km/h with distance gain of 7.8 m (1.6 car lengths) with continuous assist of 324 kJ on one straightaway.

Because the amount of tire skidding and aerodynamic specifications, including speed during assist and road surface conditions, differed from initial simulation results, there was some discrepancy in values, but the effectiveness of KERS was sufficiently demonstrated.

A driver who experienced the full power assist of KERS commented, “I was impressed by how amazingly

Fig. 21　KERS car and engineers

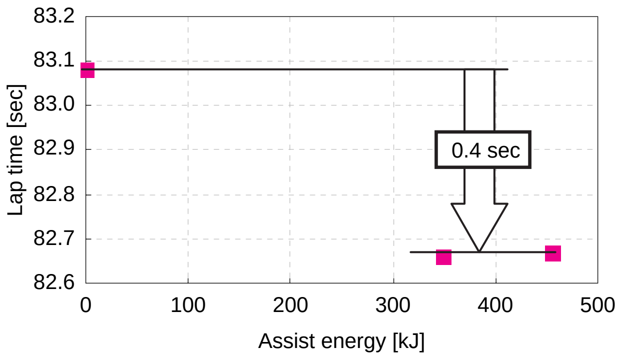


Fig. 22　Lap time against the assist energy

fast I approached the hairpin when the assist kicked in. It makes acceleration from the engine alone feel as if carrying a heavy weight.”

Then the RA1089 was completed in November 2008 ahead of winter testing. However, the system of the RA1089, mounted with a new VCU as described earlier, is very different from that of the RA1082, and so noise frequently resulted in system failures, and together with bench analysis, day after day was spent in noise analysis on actual vehicles and discussions on countermeasures. For the RA1089, the HRF1 engineers, mechanics and other track test members worked together to conduct more than 100 fire-ups, more than we did with the RA1082, with the result that we overcame all the troubles. This was the result of all the members, from the UK and Japan, coming together to solve some very difficult issues.

Then on November 28th, the RA1089 underwent shake down on the straight course of Santa Pod in the UK (Fig. 24), confirming that all systems were operating normally, and since the judgment was made that the system could be used on a circuit, the RA1089 and its equipment were loaded onto a trailer on the evening of December 4th as the test vehicle for joint testing on December 9th at Jerez. On December 5th, however, it was announced that Honda was pulling out of Formula One racing, and all activities ceased, which meant that the shake down of November 28th was the last track test of Honda’s third-era Formula One activities. The record of KERS circuit track tests is given in Table 5.

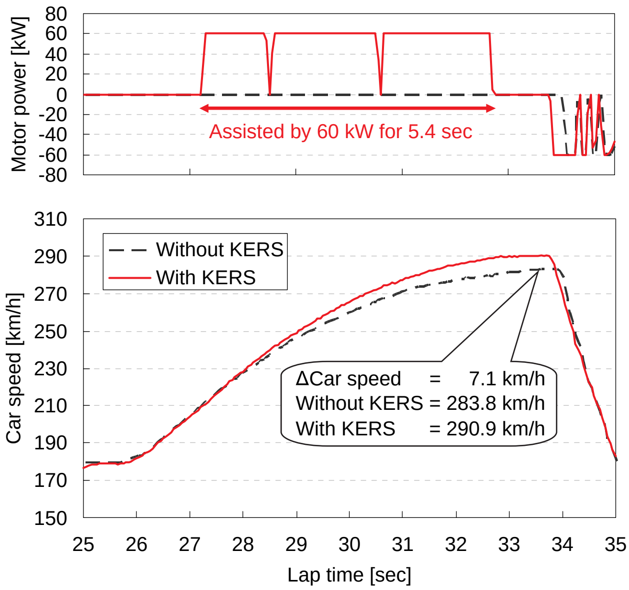


Fig. 23 Car speed comparison

Fig. 24 Launch of the RA1089 with KERS system

Table 5 Track test results of KERS car

Date	Circuit	Driver (chassis)	Mileage	Notes
29-Apr	Santa Pod	A. Wurz (RA1082)	2.4 km	Shake down
7-May	Silverstone (short course)	J. Rossiter (RA1082)	25.0 km	Shake down 25 kW assist 35 kW recharge
29-Jul	Silverstone (Stow School)	M. Conway (RA1082)	48.6 km	Shake down 60 kW assist 60 kW recharge
16-Sep	Jerez*	M. Conway (RA1082)	88.6 km	60 kW assist 60 kW recharge
19-Sep	Jerez*	A. Wurz (RA1082)	225.8 km	60 kW assist 60 kW recharge
13-Nov	Kemble	A. Davidson (RA1089)	10.5 km	Shake down
28-Nov	Santa Pod	A. Davidson (RA1089)	33.5 km	Shake down 25 kW assist 25 kW recharge

*: Full race track

5. Conclusion

The effort to develop an electrical hybrid system that intended to introduce KERS in 2009 produced the following results.

(1) A compact lightweight hybrid system was developed and a car was produced with performance to realize maximum output of 60 kW and 400 kJ of assist on each lap of a circuit.
(2) Honda became the first in the world, beating out other companies, to conduct actual driving tests of KERS on a circuit, and demonstrated faster lap times with KERS (about 0.4 sec faster with 400 kJ of assist) and the effect of an overtaking boost (about 7 km/h with assist of 324 kJ).
(3) The team demonstrated the safety of KERS in all processes in the development of functional components and the vehicle, as well as in circuit driving.

Afterword

The development effort was confusing at first, because several things were happening simultaneously: R & D on completely new KERS functional components, all staff members concerned with KERS – the technical development staff, race engineers, the mechanics, other site managers, vehicle production staff, and vehicle electrical equipment staff in the UK and Japan – receiving training and mastering high voltage systems, cooperation with the FIA on safety, and working to revise regulations. But the fact that the development took place in such an unusually short period of time (nine months from the start of development until first driving, and 15 months until shake down of the racing specs vehicle), is a tribute to the joint development system with the HRF1 members, overcoming language barriers and national borders and showing mutual respect, and to the joining of minds of everyone concerned, including cooperating manufacturers, those involved in distribution, business travel and translation, and local staff. This is something all members are proud of.

Reference

(1) Nakamura, S., Motohashi, Y., Hayakawa, S.: Development of Wind Simulator Equipment for Analysis of Intake Phenomena in Formula One Engines, Honda R&D Technical Review, 2009, F1 Special (The Third Era Activities), P. 95-100

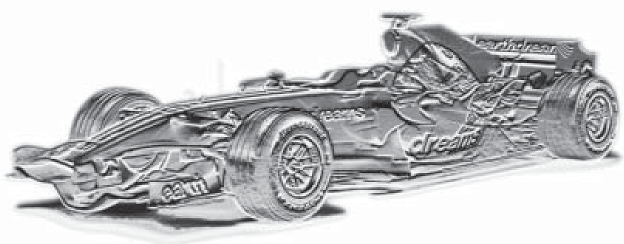
■ Author ■


Masataka YOSHIDA


Masato KITA


Hirofumi ATARASHI

Descriptions of Material Technologies



High-performance Shell Bearing from New Material

Kiyoshi ITO*　Makoto ASAMI*　Hiroshi KOINUMA*

ABSTRACT

A material that combines high thermal conductivity, seizure toughness, and high strength was developed by enhancing hardening methods and fine homogenization in the deposit phase of a Corson alloy. This material was used as the back metal for shell bearings, it was used in parts as a “high strength high thermal conductivity shell bearing,” and it achieved consecutive use in events under one-engine, two-race rules.

1. Introduction

Shell bearings for use in Formula One racing engines require sliding durability to a high PV value (surface pressure P x peripheral velocity V: 2000 MPa m/s or above) in order to sustain the inertial force and explosion force experienced under high speed conditions. When concerns about insufficient durability existed, there were sometimes cases when engine power performance would be suppressed in order to assure durability.

Figure 1 shows the cross-sectional structure of shell bearings. Conventional shell bearings have a three-layered structure. On the other hand, shell bearings for use in Formula One racing require sliding durability that is enhanced to an ultimate degree, and therefore Cu alloy back metal with a two-layered structure that emphasizes heat drawing performance was adopted. With this structure, the material properties of back metal Cu alloy have a substantial influence on shell bearing durability, and it was therefore decided to aim for development of a new Cu alloy.

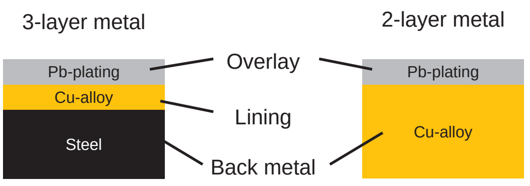


Fig. 1　Cross-sections of each metal

* Automobile R&D Center

Table 1　Main properties of several Cu-alloys

Material	Electric conductivity IACS (%)	UTS MPa	0.2%YS MPa
BeCu50	45-50	690 - 800	–
BeCu25	25	>1000	–
CF-2 (Cr-Cu alloy)	65	490	–
NC50 (Corson alloy)	40-45	690	590

2. Development of New Cu Alloy for Use in Shell Bearings

A Cu alloy for use in shell bearing back metal should have the following three properties in combination:

(1) Thermal conductivity: Efficient cooling of heat generated by sliding

(2) Sliding performance: Inhibit seizing against nitriding crank shaft

(3) High strength: Assure contact pressure with connecting rod big end

2.1. Selection of Alloy

Table 1[1] shows the main properties of typical Cu alloys. Cu alloys with high thermal conductivity as represented by electric conductivity (IACS%) are Be-Cu alloys (hereafter BeCu), Cr-Cu alloy (hereafter CF-2), and Corson alloy (hereafter NC50)[2].

These Cu alloys were subjected to a test of toughness against seizure with nitriding steel using the sliding rig test of Daido Metal Co., Ltd. The results are shown in Fig. 2. NC50 has greater seizure toughness than BeCu alloy, which makes an excellent sliding material for use in a bush and is commonly found in that application. NC50 is a Cu alloy of the deposit strengthening type that has Ni, Si, and Cr as main additive constituents. The deposit phases (Ni_2Si, Cr_3Si) are thought to suppress its seizure against steel. Material development was carried out using NC50, a material that achieves a balance of high thermal conductivity and seizure toughness, as the base.

2.2. Balance of Thermal Conductivity and Strength

The use of NC50 material as shell bearing back metal to assure contact pressure against the connecting rod big end under high PV conditions necessitates strength.

Greater strength can be achieved by increasing the deposit phase, but increasing the quantity of additive elements results in a reduction in thermal conductivity. Instead of changing the quantity of additive elements, therefore, it was decided to seek higher strength by adding hardening methods (extrusion method, multi drawing method) to the process. In addition, the heat treatment was optimized (by multi heat treatment) in order to achieve a finer, homogenized deposit organization, and thermal conductivity was increased. Relative to NC50 (the base material), the developed material NC50ES achieved a 30% increase in strength and a 12% increase in electric conductivity (Fig. 3).

3. Confirming the Effectiveness of Shell Bearing Parts

3.1. Shell Bearing Rig Test

Prototype shell bearings were fabricated with NC50ES material as the back metal. These were subjected to performance evaluation using a shell bearing tester[3]. Figure 4 shows the results.

The horizontal axis shows the flow rate of the oil supply from the shaft to the shell bearing, while the vertical axis shows the shell bearing temperature (measured by a thermocouple embedded inside the shell bearing). It was confirmed that NC50ES has a shell bearing temperature 3-5°C lower than that of conventional materials. This is conjectured to be the effect of increased thermal conductivity in the back metal material.

3.2. Engine Tests

Figure 5 shows the state of damage to the shell bearing sliding surface after conducting engine endurance testing. The condition of the remaining overlay on the shell bearing surface (No. 1 cylinder on the connecting rod beam side) is shown three-dimensionally. There is more remaining overlay (area) on the NC50ES shell bearing. It is speculated that when the temperature of the shell bearing as a whole is reduced, this also causes the temperature of the overlay to decrease, resulting in an increase in wearing toughness. This material also made it through durability testing in modes matching the most

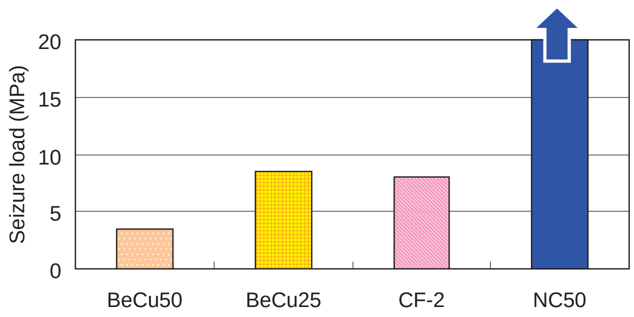


Fig. 2 Seizure toughness of Cu alloys

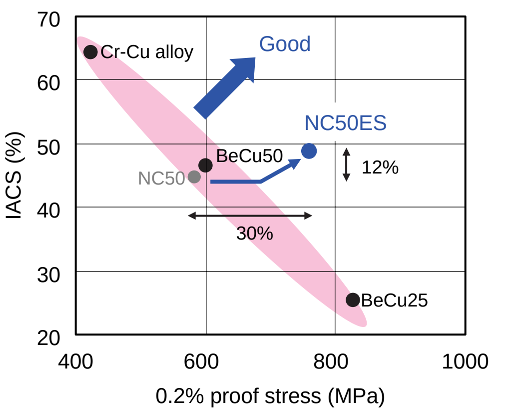


Fig. 3 Position of NC50ES material

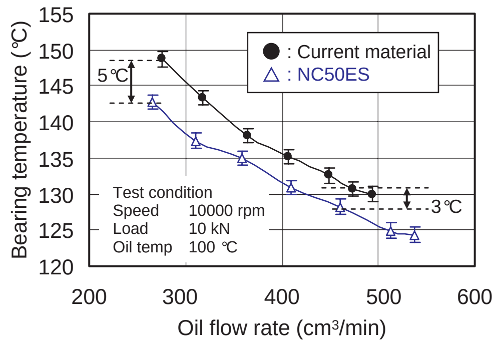


Fig. 4 Result of test on bearing tester

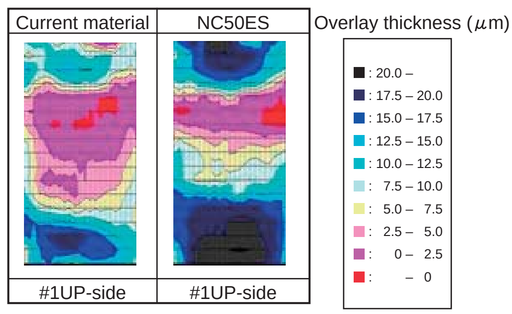


Fig. 5 Overlay thickness after engine endurance test

demanding high-load conditions on the circuit, and produced satisfactory results without adhering, seizing, or cracking.

4. Conclusion

The new Cu alloy NC50ES was developed for use as shell bearing back metal. Shell bearing parts were fabricated from the material.

The reliability of the shell bearings was enhanced with respect to the high-speed, long-distance assurance of Formula One engine regulations. The shell bearings achieved continuous use in events under one-engine, two-race rules as well as straight-end 19000 rpm performance enhancement.

References

(1) http://www.yamatogokin.co.jp, 2009/03/27
(2) JP, 3563315, B (2004)
(3) Machida, K., Takahashi, S., Ueshima, H., Fujiki, K., Proposal of New Bearing Simulation Rig Tester for High Efficiency Engine Bearing Development, Honda R&D Technical Review, Vol. 18, No. 2, p. 131-139

Acknowledgement

This occasion is taken to express the warmest gratitude to Miyoshi Gokin Kogyo Co., Ltd., and Yamato Gokin Co., Ltd., which cooperated generously in the development of this material.

■ Author ■

Kiyoshi ITO

Makoto ASAMI

Hiroshi KOINUMA

Development of Hollow Crankshaft

Kiyonobu MIZOUE* Yasushi KAWAHITO* Ken MIZOGAWA*

ABSTRACT

A hollow crankshaft of light weight and high quality was realized through the use of friction welding. The oil passage in the journal was made a hollow structure, and this had the effect of increasing average oil pressure by 500 kPa (reducing the pressure drop), reducing the pulsation amplitude by 400 kPa, and raising the minimum oil pressure by 600 kPa. The connecting rod bearing clearance was also adjusted and confirmed to yield 1 kW of performance enhancement and over 1200 km of durability as well as curbing bearing friction. Although this new crankshaft was never used in a race, this developed technology has provided the prospect of the first hollow crankshaft of 10 kg or less for practical use in the Honda Formula One.

1. Introduction

The durable reliability of the crankshaft is essential in the high speed and high power development of Formula One engines. Accordingly, this part is also considered to require lower friction and lighter weight in the pin and journal rotating portions as well as higher stiffness with respect to torsion resonance. The development of a lightweight, high stiffness hollow crankshaft has been underway since the end of 2003 as a horizontal deployment of the hollow structure of the connecting rod in other parts. Meanwhile, the International Automobile Federation (FIA) included a provision in its regulations that no welding would be permitted between the front and rear main bearing journals from 2006. This would make it impossible to use a welded hollow crankshaft in cars entered in FIA races. Honda therefore established the structure manufacturing technology for a hollow crankshaft in 2005, and set out to install it in a racing vehicle for use that year.

The creation of a hollow crankshaft required joining technology capable of providing stable strength and precision in the junctions that are of greatest importance. It also required manufacturing technology capable of assuring quality by enhancing the accuracy of machining on junctions as well as by removing burrs and the like.

2. Establishment of Hollow Structure Joining Technology and Quality Assurance

In order to create this hollow structure, the joining locations and strength feasibility were verified and joining methods were selected. The manufacture of a high-quality, high-precision V10 hollow crankshaft involved enhancing the accuracy of friction welding (hereafter FW) on five journals and implementing quality assurance regarding residual burrs at joins, and deformation from machining and nitriding. Figure 1 shows the hollow structure and an external view of it.

2.1. Hollow Structure Specifications

The location and optimum shape of joins in the hollow were subjected to CAE verification, from which the joins at the center of the journal, as shown in Fig. 2, were selected.

The hollow pin with a diameter of 34 mm (journal wall thickness: 6 mm) opened the way to satisfying the strength requirements.

The bending stress, caused by explosion force, is

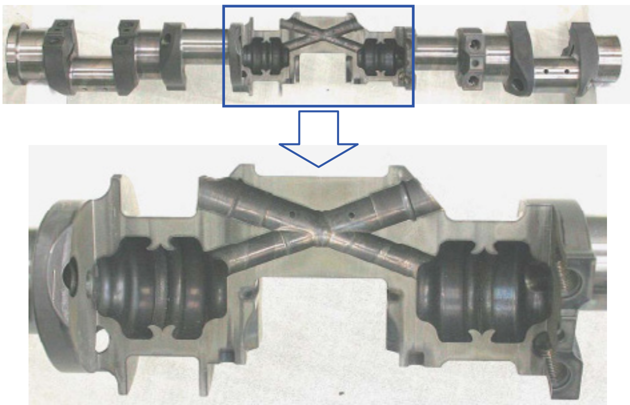


Fig. 1 Section of hollow crankshaft

* Automobile R&D Center

mitigated by the hollow structure, so that the stress was lower than in a solid. Furthermore, the inertial force was little influenced by the hollow structure, and the hollow pin with 34 mm diameter had lower stress than the solid pin with 33 mm diameter.

2.2. Joining Technology and Strength Verification

The three manufacturing techniques of electron beam welding (EBW), diffusion bonding, and FW were subjected to comparative evaluation by rig testing of fatigue strength. FW, which yielded join strength equal to the strength of the base material, was decided on for the specifications.

2.3. FW of Five Journals

The basic conditions were set at a speed of 1900 rpm, pre-force of 10 tons, a pre-heat time of 9.5 sec, an upset force of 30 tons, and upset length of 9 mm. The surface roughness of the join surfaces was enhanced and acetone degreasing was done during the cleaning prior to joining (Fig. 3).

Key points in determining the FW conditions included shaping the flash form to rise up as much as possible and ensuring that the media is kept from being worked in.

The chucks for gripping the workpieces for the series of five joins were bore–machined together to align their center lines, enhancing the accuracy of butting by the rotating side as well as the clamping stiffness on the fixed side. The equipment peculiarity was addressed using adjusting shims to correct for eccentricity of axis. With the join conditions established, the series of five FW joins (joins made while undergoing 180 degree phase reversal) was made, achieving an offset of 0.35 mm at the largest with respect to the target offset of 0.5 mm (Fig. 4).

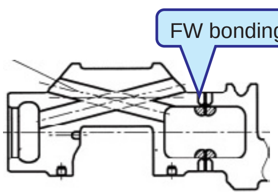


Fig. 2 Journal joint

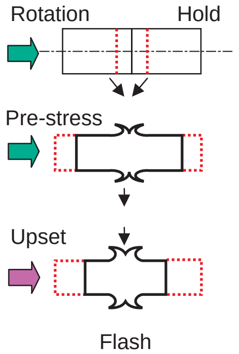


Fig. 3 FW bonding

2.4. Quality Assurance of Hollow Portion

The hollow portions joined by FW also serve as oil passages, making it necessary to conduct thoroughgoing quality assurance regarding the removal of burrs. First of all, the FW conditions were set so as to stabilize the flashes and control their shape. As shown in Fig. 5, scale, machining burrs, and spatter are removed after nitriding by barrel finishing with vibration using steel spheres and cleaning by cavitation. All were then checked by microscope and guaranteed. Figure 6 shows the nitriding quality of the hollow interior. Although the central portion of the flashes were not nitrided, the joins and the hollow portion were confirmed to be uniformly nitrided. Rig testing of the 2.5 cylinder joined items confirmed strength equal to the base material.

2.5. Heat Treatment and Manufacturing Completion Accuracy Assurance

Deformation from heat treatment was addressed by using crankshaft pin milling items as dedicated jigs and setting conditions by corrective hardening. When the offset exceeded 1.0 mm, the bend was trued up. For working accuracy, a base adjustment eccentricity center was used and uneven thicknesses in the finish of the hollow portion were minimized. The target for wall thickness of 6 ±0.75 mm in the journal hollow portion was achieved with 6 ±0.3 mm and a maximum uneven thickness of 0.55 mm.

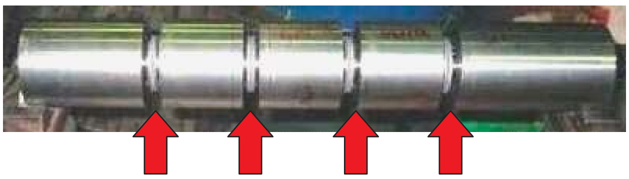


Fig. 4 Friction bonding

Fig. 5 After cleaning

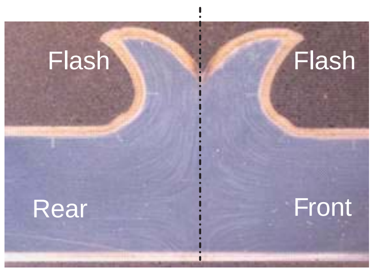


Fig. 6 Nitride of FW section

3. Results

A weight reduction of 0.8 kg was achieved, going from 6.55 kg (10.3 kg with counterweight) for the solid crankshaft to 5.75 kg (9.5 kg with counterweight) for the hollow, and making this the first Honda Formula One crankshaft to realize a weight of 10 kg or less. The effects of enhanced lubrication with the hollow crankshaft include recovery of 500 kPa in the instantaneous minimum oil pressure, which had shown negative pressure because of oil pressure pulsation, and prospects for durability.

4. Conclusions

The technology was established for manufacturing a hollow crankshaft that would satisfy the properties required for use in a high-speed, high-power Formula One racing engine.

The crankshaft was made hollow using a highly reliable FW technique, and production engineering capable of guaranteeing crankshaft accuracy as well as the cleanliness to use the hollow portion for oil passages was established.

It has been verified that the developed technology reduced weight 8% relative to the conventional crankshaft and had the effect of oil pressure pulsation reduction that could not be obtained with the conventional structure, yielding the prospect of durability for practical use. Regulations prohibiting welded crankshaft structures went into effect in 2006, and this hollow crankshaft was never used in actual competition.

Acknowledgements

This occasion is taken to express the warmest gratitude to Takayuki Ohnuma and Nobuki Matsuo of Honda Engineering Co., Ltd., for their cooperation with FW joining technology and quality assurance.

■ Author ■

Kiyonobu MIZOUE

Yasushi KAWAHITO

Ken MIZOGAWA

Development of Metal Matrix Composite Piston

Kazuhisa ISHIZUKA*　Katsuhito KUSAKAI*　Nobuyuki IMAI*

ABSTRACT

A metal matrix composite that can be expected to have outstanding strength at elevated temperatures was applied to reduce the piston weight. In composites, it is generally difficult to cope with both strength and toughness. To solve this issue, a powder alloy was applied using the mechanical alloying process, and the manufacturing process and surface treatment were optimized. This achieved a 16% weight reduction compared to the conventional material, AA2618. In addition, this enabled the Formula One engine speed to be increased by 400 rpm.

1. Introduction

Reducing the inertial weight of reciprocating systems is the most important subject to increase engine speeds, and it was necessary to develop new materials to realize lightweight and high strength parts for Formula One pistons.

Aluminum alloy metal matrix composites (MMC) strengthened by ceramic dispersion can be expected to have high specific strength, high stiffness, and better characteristics at elevated temperatures. However, it was necessary to overcome the issues that toughness and ductility are generally low, and strength anisotropy which comes from the restrictions on the forging process. To solve these issues, a powder alloy was applied using the mechanical alloying process, and the characteristics were enhanced by modifying the forging and heat treatment processes.

2. Developed Technologies

2.1. MMC Materials (AMC225XE) for Pistons

Table 1 compares the physical properties of MMC and the conventional material, AA2618[1]. MMC uses AA2124 as the base alloy, and adds SiC with a particle size of 3 µm in a 25% ratio by volume as the dispersed ceramics. The SiC is dispersed and alloyed with the base alloy powder for the prescribed time using a high energy mill. Figure 1 shows a metallographic image of the MMC, and a mapped image of the carbon from the SiC obtained using EPMA. Here, a countless number of submicron size particles can be observed, dispersed uniformly between large particles with several µm. These submicron particles are thought to be the SiC crushed by the mechanical alloying process and dispersed within the matrix. This material realizes high fatigue strength at elevated temperatures and better toughness and ductility by not only precipitation of the base alloy and the law of mixture, but also dispersion strengthening mechanism of the nanosize particles.

Table 1　Properties of piston material

Material	Modulus	Density	CTE	CTC
	GPa	g/cm³	ppm/K	W/mK
AA2618	74.6	2.76	22.6	165
MMC	113.0	2.89	15.5	129

2.2. Forging and Heat Treatment Technologies

In general, powder metal materials are worked after sintering to achieve good mechanical properties. In early days, pistons were forged using previously extruded bars, but there was the issue that anisotropy in the piston roof was serious, and the expected characteristics could not be obtained. The issue of anisotropy was successfully resolved by forging pistons directly from HIP (Hot

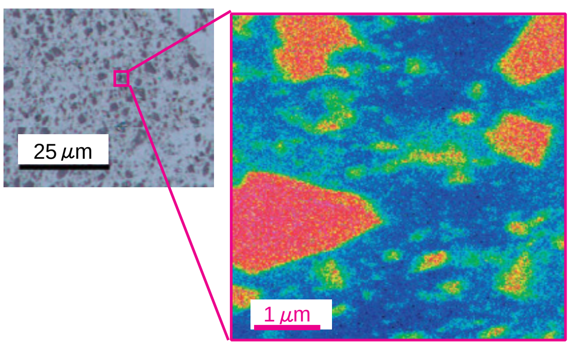


Fig. 1　Metallograph of MMC and distribution of carbon

* Automobile R&D Center

Isostatic Pressing) materials for which 3-dimensional isotropy can be expected.

The required forgeability was achieved by the low-speed hydraulic forging press. High cooling rate during quenching was ensured by using thin materials with a near-net shape, which realized good mechanical characteristics. Application of near-net materials also helped reduce machining of MMC, which is a difficult-to-cut material.

2.3. Surface Treatment

The lower coefficient of thermal expansion due to SiC dispersion leads to an increase in the clearance between the piston and the cylinder. This causes an issue in securing reliability of these sliding parts. This issue was overcome by applying an electroplated hard film and a resin coating containing dispersed hard particles to the skirt and top land, which secured the required durability and reliability.

3. Effects on Performance

Figures 2 and 3 show the mechanical characteristics of a test sample taken from the roof section of the piston material. Both high fatigue strength and ductility were achieved in the piston operating temperature range of 200 to 300 °C.

Use of this material realized lightweight and high strength pistons that were applied to races from the first GP of 2004.

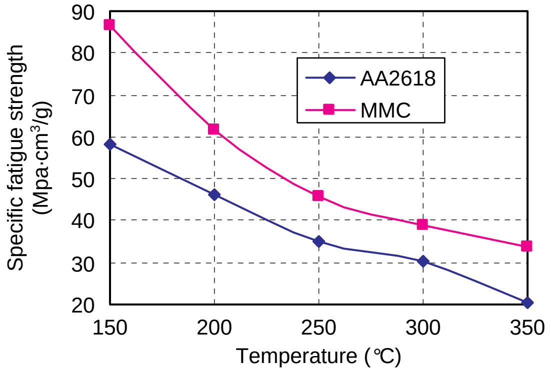


Fig. 2 Specific fatigue strength of developed material

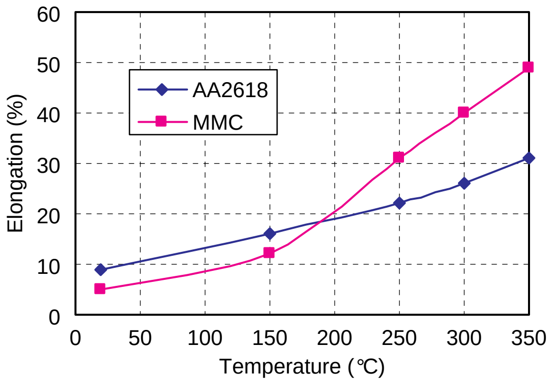


Fig. 3 Elongation of developed material

4. Conclusion

A high strength piston material was successfully developed by optimizing the composition and manufacturing method of MMC manufactured using the mechanical alloying method. Application of this material reduced the piston weight by 16% compared to the conventional material AA2618, and achieved an increase in the engine speed of 400 rpm.

Reference

(1) http:/www.amc-mmc.co.uk/

■ Author ■

Kazuhisa ISHIZUKA

Katsuhito KUSAKAI

Nobuyuki IMAI

Development of Lightweight Titanium-aluminide Piston Pin

Sadami MINATO* Yasunori ONAHA*

ABSTRACT

An intermetallic titanium aluminide material displaying increased fracture toughness and fatigue strength, and a process enabling the formation of the material into components, have been developed. Exploiting the material's low specific gravity and high modulus of elasticity, it was employed to manufacture a piston pin, enabling the achievement of a 17% reduction in weight and a 28% increase in rigidity against a conventional nitriding steel piston pin. The developed piston pin has contributed to increased speed, power and long-distance reliability in V10 engines, which have been required to complete two race distances since 2005.

1. Introduction

In addition to possessing high specific stiffness at the upper limit of Formula One material regulations and excellent high-temperature strength, titanium aluminide (TiAl) also displays excellent fatigue strength at ambient (or room) temperatures. For these reasons, manufacturers have been attempting to extend the use of the material to the main reciprocating components, which are the subject of a constant quest for weight savings. However, there have been concerns over the low fracture toughness and the quality of the European materials employed in engine valves(1), and other significant issues have arisen, including insufficient resources for development due to oligopolistic supply.

The aim of the project discussed in this paper was to develop an original high-quality TiAl material by balancing fracture toughness with strength, and to contribute to the achievement of increased engine speed by reducing reciprocating mass through the application of the material in piston pins.

2. Developed Technology

2.1. Material Design

In order to increase the strength and fracture toughness of TiAl, the first important step is to use an extrusion process to refine the coarse lamellar microstructure produced by ingot casting. The piston pin material must be produced at a diameter of 19 mm, and therefore necessarily possesses an insufficient extrusion ratio. For this reason, refinement of the microstructure during working was promoted by adopting a chemical composition design in which an intermetallic γ phase (TiAl), α phase (hcp-Ti), and β phase (bcc-Ti) coexist at 1150 °C, the final hot working temperature.

In addition, the exploitation of the metastable β phase in the final microstructure, which promotes strength and fracture toughness, is a factor in the achievement of increased ductility. Therefore, the chemical composition was designed to produce large quantities of the β phase material in a stable state at the annealing temperature of 1000 °C. Figure 1 shows the results of a comparison of microstructures and phase volume fractions using Thermo Calc(2).

2.2. Forming Process

A process of cold crucible induction melting and continuous casting, in which the molten developed material is solidified while being subjected to electromagnetic mixing, is used to obtain a uniform ingot microstructure.

TiAl possesses a high level of resistance to heat deformation. Stainless steel sheets are generally used for canning to control the decline in the work temperature and the material undergoes hot plastic working between 1200 °C and 1350 °C(1), (3). For the developed material, on the other hand, a dedicated antioxidant was selected.

* Automobile R&D Center

The use of this antioxidant in combination with a die glass lubricant reduces the material's deformation resistance, enabling hard working at an extrusion ratio of 58 at a lower temperature than specified above, between 1150 °C and 1200 °C. Refinement of the microstructure during the formation process is promoted in this range.

Finally, the material is subjected to annealing at 1000 °C for 2 hours, followed by forced cooling (Ar gas cooling) in order to obtain a β phase between 20% and 35%. By this means, stable elongation of 2% or more at ambient temperatures has been balanced with fatigue strength of 800 MPa or more (rotating bend) at ambient temperatures. Figure 2 shows the microstructure of the material (phase volume fraction) and its effect on the material's mechanical properties under each of the conditions used in the formation process.

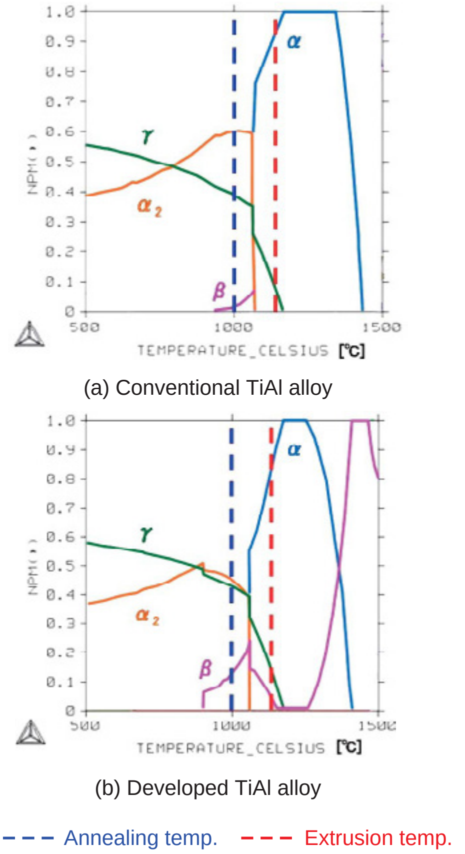


Fig. 1 Comparison of phase ratio (Thermo-Calc)

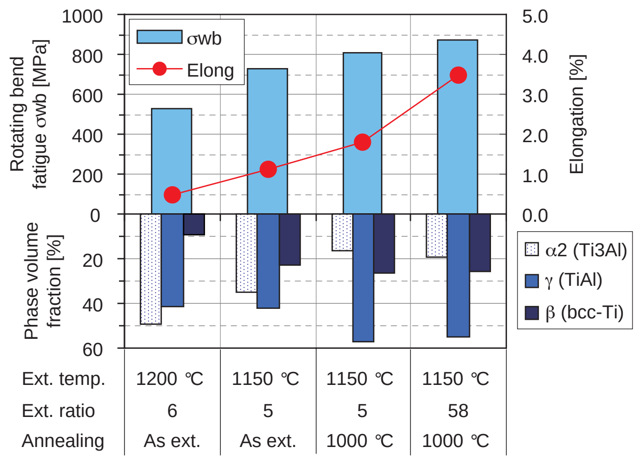


Fig. 2 Effect of phase structure on mechanical properties

3. Component Specifications

CAE and stress measurements were conducted to determine the piston pin design in view of the strength of the piston pin itself and the effect to the piston, and the component design was changed from the conventional tube type used with nitriding steel to a solid type. The solid design increased rigidity in the collapsing direction by 28% and reduced stress on the piston pin boss, which had previously represented a reliability issue, by 6%, while the application of the new material enabled the achievement of a 17% weight saving. Figure 3 shows comparison results of deformation in CAE analysis.

In addition, given the decline in hardness of the base material with the use of TiAl, a high-hardness diamond-like carbon (DLC) coating with a 7 μm Cr_2N film formed by PVD sputtering as a backup layer was employed as the surface treatment, enabling adhesion strength and scratch resistance to be increased.

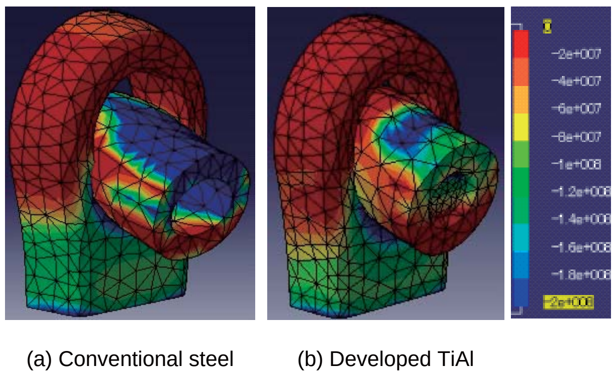
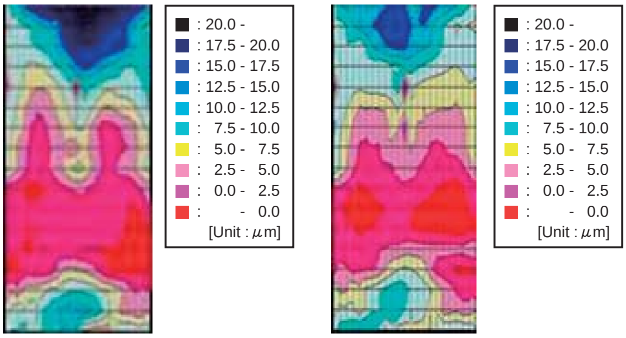


Fig. 3 Piston pin deformation (Catia V5 CAE)

4. Performance Results

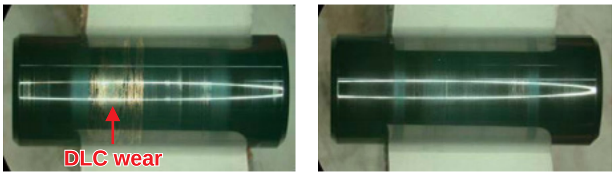
The solid TiAl piston pin was evaluated in high-load durability mode in an engine. As an effect of the achieved weight savings, the level of conrod bearing wear was equivalent to that associated with a conventional nitriding steel piston pin, even though the engine speed had been increased by 200 rpm.

In addition, the high rigidity of the developed material reduced the level of piston pin DLC wear and piston boss cracks, which had previously represented a concern in terms of reliability over long distances. Figure 4 compares the film thickness of the remaining overlay layer on the surface of conrod bearings produced using the developed and conventional materials following the endurance test. Figure 5 shows a comparison of the condition of the piston pin DLC for the developed TiAl and conventional steel following the endurance test.



(a) Conventional steel (18800 rpm/1503 km)   (b) Developed TiAl (19000 rpm/1485 km)

Fig. 4 Overlay thickness of connecting rod bearing after endurance test



(a) Conventional steel (18800 rpm/1503 km)   (b) Developed TiAl (19000 rpm/1485 km)

Fig. 5 DLC condition of piston pin after endurance test

5. Conclusion

Control of microstructure by means of chemical composition design and management of the formation process has enabled the development of a TiAl material that balances high fatigue strength with excellent fracture toughness (displaying an elongation of 2% or more). The use of this material to manufacture a piston pin has resulted in the achievement of a 17% reduction in weight, and has contributed to a 200 rpm increase in engine speed, and, by means of a reduction in the level of deformation of the pin, to reduced DLC wear and increased reliability in the piston pin boss section.

Acknowledgments

The authors wish to take this opportunity to offer their sincere thanks to the staff of the Titanium Metals Division of Kobe Steel, Ltd., and the Kobelco Research Institute, Inc., who generously assisted in the development of the material and product discussed in this paper.

References

(1) Clemens, H., Kestler, H., Eberhardt, N., Knabl, W.: PROCESSING OF TiAl BASED ALLOYS ON AN INDUSTRIAL SCALE, The Minerals, Metals & Materials Society, Gamma Titanium Aluminides 1999, p. 209-223 (1999)

(2) http://www.thermocalc.se/, 2009/04/02

(3) JP, Patent Application No. 2000-24748

■ Author ■

Sadami MINATO

Yasunori ONAHA

Development of Hollow Titanium Connecting Rod

Sadami MINATO*　Ken MIZOGAWA*

ABSTRACT

It is necessary to reduce the reciprocating mass in order to increase the engine speed and power of Formula One engines. The project discussed in this paper therefore set out to increase the section modulus of the shaft of the connecting rod while maintaining its rigidity and achieving weight savings. To this end, the diffusion bonding method was optimized, and a process of manufacturing a hollow connecting rod was developed. The developed connecting rod is lighter in weight and higher in rigidity than a rod with a conventional I-type section produced by forging, and has contributed to enabling engines to be increased in speed.

1. Introduction

As one of the main kinetic components enabling the operation of high-speed and high-power Formula One engines, connecting rods (conrods) are the subject of a constant quest for weight reductions and increases in strength and rigidity. For this reason, titanium alloys displaying high specific strength were applied in their manufacture. In 2000, weight savings were achieved through the use of a β-rich α+β titanium alloy, SP-700[(1)], which possesses 25% higher fatigue strength than that of the formerly used 6A14V titanium alloy. However, responding to demands for further weight savings exclusively by means of increasing strength was bringing materials close to the limit of rigidity design, a situation which necessitated a new technological breakthrough.

The potential for the use of a hollow conrod structure as a means of achieving weight savings while maintaining a geometrical rigidity was therefore studied.

2. Developed Technology

2.1. Study of Method for Hollowing Conrod

A variety of potential methods of realizing a hollow conrod structure were studied. One suggested method was to form a hollow shaft extending from the big end by means of electrochemical or mechanical machining, which would then be cover-welded using electron beam welding (EBW), thus forming a hollow structure. However, this method was unable to resolve the issue of the strength of the joints. Issues of reduced strength also arose in the cases of casting and wax soldering.

Diffusion bonding, as employed in the manufacture of aircraft turbines, involves the diffusion of a solid phase, and therefore does not affect the base material by heating. In addition, titanium displays a high oxygen solubility limit, so oxide layers easily diffuse and disappear on titanium surfaces. Diffusion bonding was therefore focused on, and manufacturing methods for the component were studied on this basis.

2.2. Mechanism of Diffusion Bonding

Diffusion bonding is a bonding method in which the temperature of the materials to be bonded is maintained at 0.7 Tm (Tm = melting point) or more in a vacuum or reductive gas environment, and pressure is applied in order to promote diffusion. Figure 1 shows a model of the diffusion process[(2)]. In the initial stage of the process, the asperities on the surface to be bonded are deformed and their close adherence promoted by increasing pressure and heat. Next, diffusion causes the grain boundaries at the interface between the materials to migrate and vacancies to disappear. In the final stage of the process, the remaining vacancies disappear through

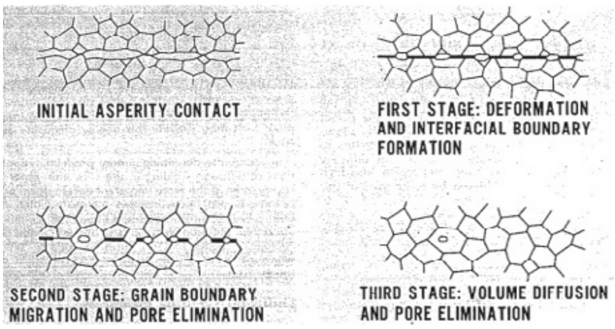


Fig. 1　Model of diffusion bonding process

* Automobile R&D Center

volume diffusion, and bonding is completed. Figure 2 shows the bonded microstructure realized in the conrod in this project. A continuous metallic microstructure with no remaining asperities at the bonding interface has been obtained.

2.3. Conrod Bonding Process

Because the conrod is solution-aged at a temperature lower than the β transformation temperature (870 °C) in order to obtain a predetermined level of strength, the bonding temperature was set at 830 °C, equivalent to the solution treatment temperature. The maximum pressure was set at 4.0 MPa, and the diffusion time kept for 5.0 hr. A hot press vacuum furnace owned by Kinzoku Giken Co., Ltd., capable of independent load control in 16 axes, was employed in the diffusion bonding.

Figure 3 shows the process of manufacture of the hollow conrod. A rolled sheet is roughly blanked using a water jet, after which it is machined into a half blank, forming a hollow shaft. These half blanks are superimposed and diffusion bonded.

The use of dowels positioned at the big and small ends controls relative displacement during bonding to within 0.13 mm at the upper limit of standard deviation. The amount of crushing in the direction of thickness was set at 4% of the initial thickness of the material, based on the height of the carbon stopper plates during hot pressing.

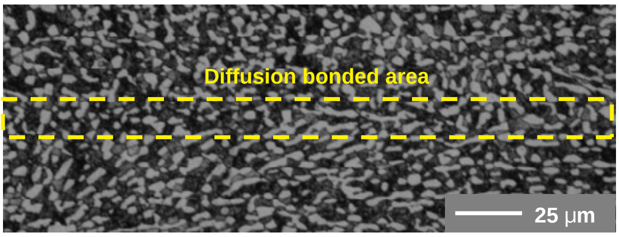


Fig. 2 Microstructure of diffusion bonded area of connecting rod

The level of roughness and cleanliness of the bonding surface affects the mechanical properties of the bonded section. Tests were therefore conducted to determine the effects of these factors using tensile test pieces bonded by means of two joint types (Fig. 4). Figure 5 shows tensile properties for different levels of bonded surface roughness. The level of surface roughness had a particular effect on the elongation and reduction area of the T-joint, and was therefore set at Rt1.6 or below in order to obtain tensile properties equivalent to those of the base material.

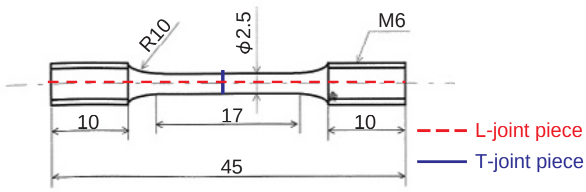
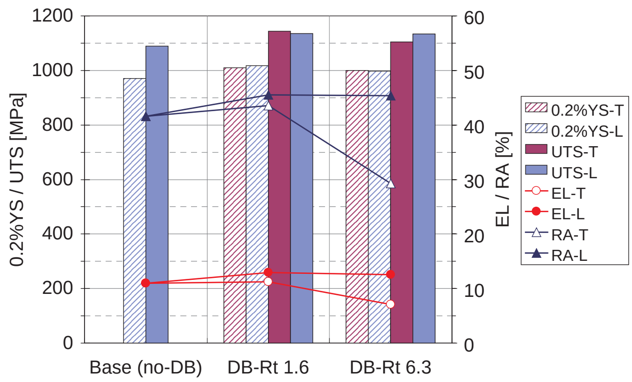


Fig. 4 Diffusion bonded tensile test piece



• 0.2%YS : 0.2% yield strength
• UTS : Ultimate tensile strength
• EL : Elongation
• RA : Reduction area

Fig. 5 Effect of surface roughness on tensile properties of diffusion bonded test piece

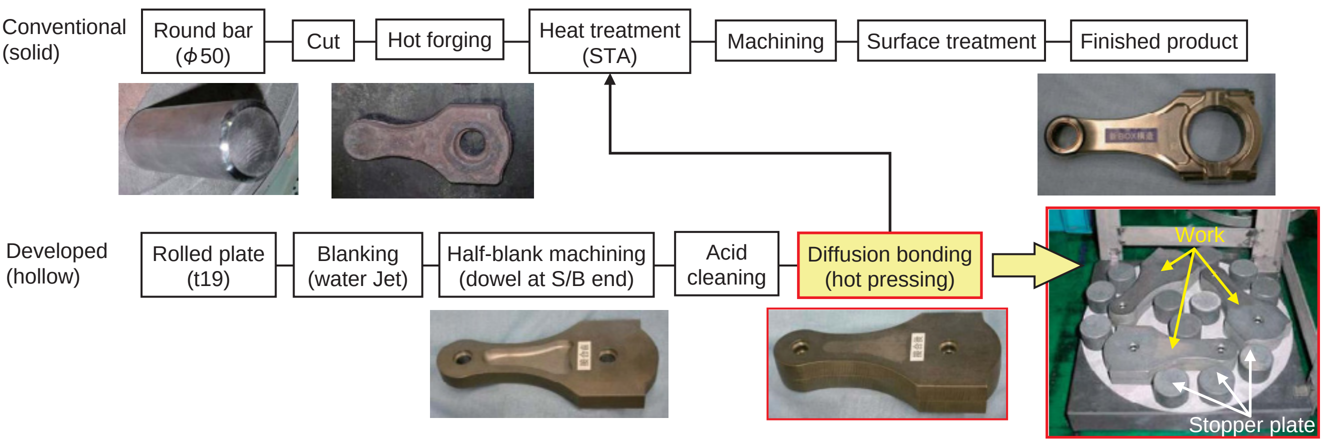


Fig. 3 Developed process of manufacture of hollow connecting rod

3. Achieved Performance

A hollow structure in which diffusion bonding is employed in the central section of the conrod thickness has been developed, as shown in Fig. 6. This has increased the modulus section of the shaft of the conrod while enabling thickness to be minimized. Compared to a conventional I-type section, an 8% reduction in weight, 2.5 times increase in the rigidity of the shaft, and 18% increase in the rigidity of the circulation of the big end have also been achieved. In addition, as a result of the reduction in the load on the conrod bearings, the potential for a 250 rpm increase in engine speed has been demonstrated in durability tests in a real engine.

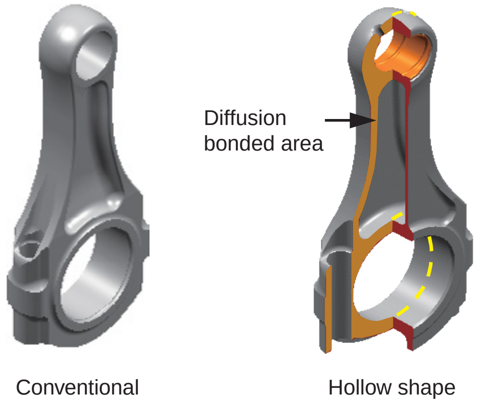


Fig. 6 Comparison of conventional and hollow conrod

4. Conclusion

A method of manufacture of a hollow conrod using diffusion bonding has been developed. The weight savings achieved enabled engines to be increased in speed and power, and the technology was introduced to race engines in 2003.

References

(1) Ouchi, C., Minakawa, K., Takahashi, K., Ogawa, A., Ishikawa, M.: Development of β-rich α+β Titanium Alloy SP-700, NKK Technical Review, No. 65, p. 61-67 (1992)

(2) Owczarski, W. A., Paulonis, D. F.: Application of Diffusion Welding in the USA, Welding Journal, Vol. 60, No. 2, p. 22-33 (1981)

■ Author ■

Sadami MINATO

Ken MIZOGAWA

Development of Titanium Aluminide Valve

Masayuki TSUCHIYA* Sakae TSUNASHIMA* Hisashi HASHIMOTO*

ABSTRACT

Titanium aluminide intermetallic compound has a high specific stiffness close to 40 GPa/g·cm^3, the maximum allowed under the regulations. The material also has high temperature strength and its use in the cylinder head valves of Formula One engines has contributed to achieving higher engine speed. Vacuum annealing has been developed to try to enhance fracture toughness, and a technique has been established to prevent products with micro-cracks from being shipped if they occur during processing, by optimizing eddy current inspection technology to overcome quality issues.

1. Introduction

Titanium aluminide (below, TiAl) has a Young's modulus of 155 GPa, density of 4.05 g/cm^3 and great strength under high temperatures. Substituting with this material in cylinder head valves (below, valves) that use titanium has enabled the achievement of higher engine speed by giving them more stiffness at lighter weight. However, the fact that this material is an intermetallic compound means that valve breakage often occurs because of the low fracture toughness and the quality of production (Fig. 1), increasing the need for high fracture toughness and enhanced quality.

Valve blank is produced through a process of twice extruding with the use of vacuum arc remelting (VAR) material[1]. This blank is produced by only one company in Europe[2], and negotiations and development that include quality assurance and process sharing up to the completion of the valve have progressed in tandem with the material's application in racing engines with the aim of achieving higher engine speed.

Fig. 1 Broken valve

Table 1 Chemical composition of TiAl (atomic%)

	Al	Cr	Nb	Ta	B	Ti
2002	46.5	2.5	1.0	0.5	0.1	Bal
2003-2005	42.0	2.5	1.0	0.5	0.1	Bal

2. Development Technology

2.1. Annealing for high fracture toughness

Since 2003, materials regulations have prohibited material with specific stiffness of greater than 40 GPa, so Honda engineers have switched to a composition with less Al, as shown in Table 1. Because of electrode dripping during VAR, valves initially have a segregation structure as shown in Fig. 2, and in 2003 valves were sorted by quality based on observation of the structure of all valves. Subsequently, melting rods for VAR were shortened and a quality sorting technique was established that used samples showing structural limits.

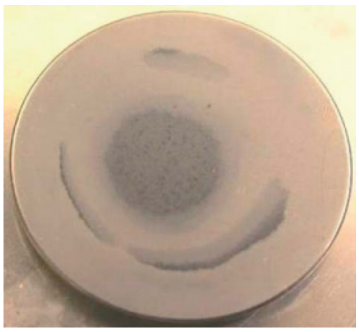


Fig. 2 Segregation structure of valve

* Automobile R&D Center

On the other hand, X-ray diffraction phase analysis of the material determined that the structure consisted of α2-phase (Ti_3Al) / γ-phase (TiAl) / β-phase (bcc-Ti). Because the β-phase is a metal phase, it contributes to fracture toughness, but the amount is very small and the valve material is made with a heated extrusion production method, so it is considered to be a semi-stable phase. Moreover, the amount of β-phase did not stabilize because of segregation and variance in extrusion conditions when producing the material.

Annealing was used to stabilize the amount of β-phase and enhance material elongation. Vacuum annealing was performed with a soaking heat treatment at 1000°C, in which the three phases of α2 + γ + β exist, and subsequently the amount of β-phase was raised to more than 20 atomic% by forced cooling with argon gas. Mechanical characteristics evaluation results are shown in Fig. 3. This elongation of more than 2% at room temperatures and rotating bending fatigue strength of more than 800 MPa at room temperatures, and ensures both high strength and high fracture toughness in the valve.

2.2. Micro-crack Scanning Non-destructive Quality Assurance Technology

During the TiAl extrusion process, the material is covered with a stainless sheath. For this reason, during machining both the highly malleable sheath and the TiAl with low fracture toughness have to be planed, and there is a possibility of cracks resulting from the vibration of machining. In addition, there is residual stress in the material during the extrusion process, so it was also conceivable that cracks could occur during finishing. If there are cracks open to the surface, they can be detected by a dye penetration test such as fluorescence dye penetration, but in some cases there are cracks with compression stress that are not open to the surface, and these can slip through without being detected by dye penetration testing. As can be experienced with Co-Ni bolts, there are also cases where high-strength materials manufactured by extrusion processing experience cracks with compression stress in the grinding process.

The technology of eddy current scanning, which detects changes in eddy current where cracks exist, was optimized, and non-destructive quality assurance was conducted. Eddy current scanning is done twice: after blank machining with the sheath and TiAl removed; and after complete machining. This enhanced detection precision. The blank machining status, from which there is a certain amount of material yet to be removed, allowed detection of relatively large cracks and was effective. For eddy current scanning on valves that have completed machining, a new sensor probe was developed, positioning was done by robot arm, the gap between sensor and valve was optimized, and noise was prevented by optimizing the signal processing filter. This made it possible to detect micro-cracks 100 μm deep in actual valves, as well as detecting micro-cracks 10 μm deep in test pieces.

Figure 4 shows the appearance of eddy current scanning.

2.3. Effects

In 2002, techniques to make inlet valves lighter helped cut 6.0 g from the weight of earlier titanium valves while raising engine speed by 600 rpm, and in 2003, valves became 2.4 g lighter and engine speed 240 rpm faster.

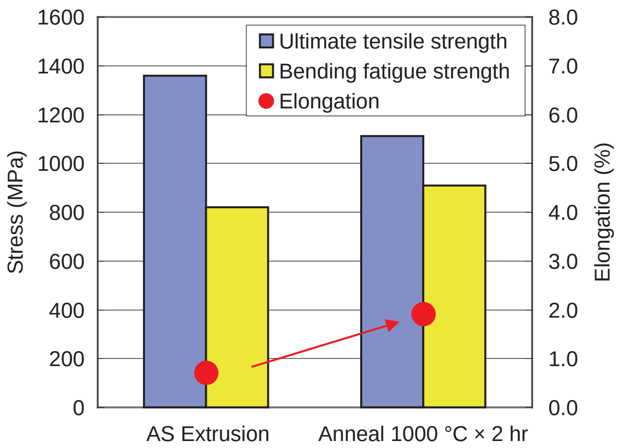


Fig. 3　Mechanical properties of TiAl

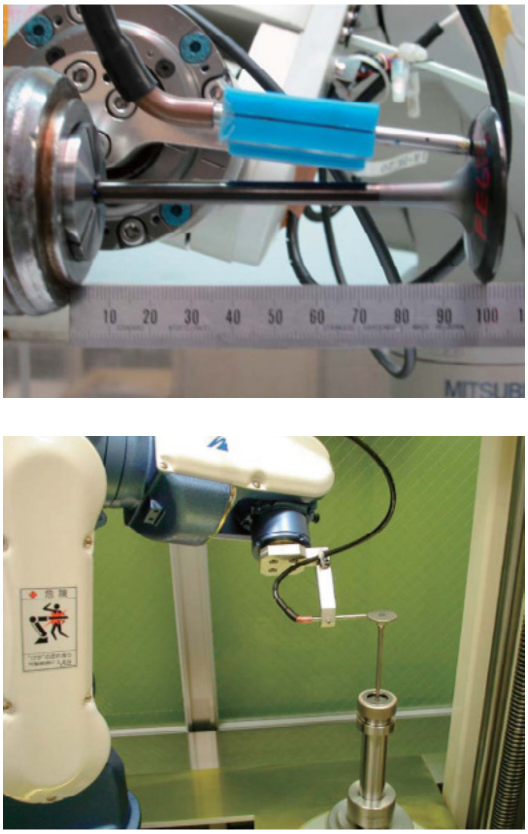


Fig. 4　Method of eddy current inspection

3. Conclusion

Using vacuum annealing heat treatment and eddy current scanning, Honda developed lightweight TiAl valves with long-distance reliability. This technology was applied to inlet and exhaust valves to help create engines able to produce higher engine speed.

References

(1) Publication of unexamined patent application, 2000-24748 (2000), Japan
(2) http://www.mwracing.eu/

■ Author ■

Masayuki TSUCHIYA

Sakae TSUNASHIMA

Hisashi HASHIMOTO

Development of Titanium Hollow Valve and Study of Sodium-potassium Valve

Taihei SANADA*

ABSTRACT

Valve specifications with the largest hollow area possible while still assuring the strength of welded areas were established, achieving a weight reduction of 4.4 g per valve, and applied to races. In addition, a high-quality oxidation resistant process was established for filling titanium valves with sodium-potassium, and technology that cools the high temperature areas of the valve by 100°C or more was developed. These cooling effects enabled expansion of the hollow area, further reducing the weight by 4.8 g per intake valve and 4.6 g per exhaust valve. Engine 2-race event durability tests were completed, but the intake valve experienced a drop in output due to the effects of sodium-potassium heat exchange, and some issues with exhaust valve durability also remained.

1. Introduction

Formula One regulations prohibited the application of intermetallics to engine parts from 2006. Therefore, the titanium-aluminum (TiAl, density 4.05 g/cm^3, Young's modulus 155 GPa) valves had to be changed to titanium alloy valves (in case of intake valve, density 4.65 g/cm^3, Young's modulus 114 GPa).

To reduce the weight and increase the stiffness of titanium alloy (Ti) valves, and increase the engine rotations, it was essential to make the valve head and stem hollow. In addition, filling Ti valves with sodium-potassium was investigated to further reduce the weight by expanding the hollow space.

2. Development Contents

Lightweight, high-stiffness Ti valves were developed by creating a hollow valve structure and establishing a technology for filling it with sodium-potassium.

2.1. Development of Hollow Valve Structure

The valve head was hollowed by drilling out the valve head, lightly fitting and electron beam welding (EBW) a plug, and performing stress-relief heat treatment.

Figure 1 shows a photograph of the valve head section. EBW was performed from the combustion side of the valve head, and melted the periphery of the plug to a sufficient depth. The heat-treated microstructure of the welded area was destroyed and the fatigue strength dropped to 30% of the non-welded area, so the residual thickness of the non-welded area was determined by valve fatigue tests to assure a parts strength equivalent to that of a solid valve head. Figure 2 shows the test results. The residual thickness requirement was set at 1.27 mm or more.

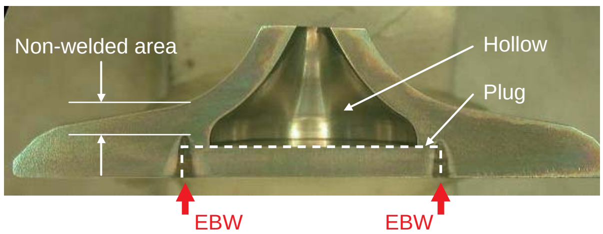


Fig. 1 Section of hollow head of intake valve

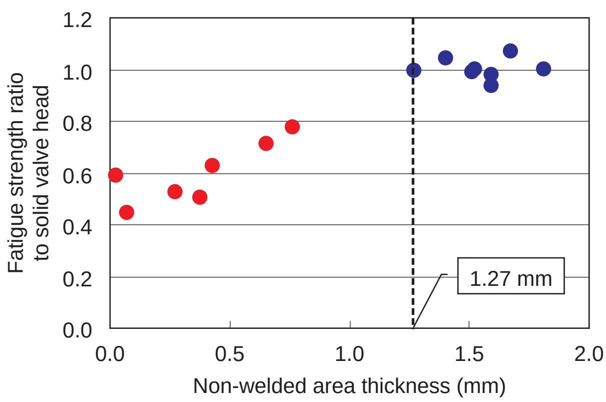


Fig. 2 Fatigue test results for welded valve head

* Automobile R&D Center

The valve head was hollowed to the greatest extent that satisfied the residual thickness requirement. Together with hollowing the stem, this achieved a weight reduction of 4.4 g compared to a solid valve.

2.2. Development of Titanium Sodium-potassium Valve

The sodium (Na) and sodium-potassium used as coolants are active, so they readily oxidize in air. In addition, titanium has a lower standard free energy of oxide formation than that of Na and potassium (K)[1], so when the inside of a Ti valve is filled with oxidized sodium-potassium, the inner walls of the valve oxidize and the strength drops. Therefore, a process for filling titanium sodium-potassium valves with non-oxidized sodium-potassium materials is necessary.

Figure 3 shows a photograph of the sodium-potassium exhaust valve section. The valve head and stem have a hollow structure, and the sodium-potassium is sealed by an inner plug and a stem end plug.

Figure 4 shows the sodium-potassium filling facility and filling operation. Before sodium-potassium filling, vacuum gas exchange is performed in the glove box to replace the air inside the hollow area with argon (Ar) to prevent oxygen from entering the hollow area. The oxygen density was set at 10 ppm or less. Na-78wt%K (NaK), which is liquid at room temperature, was selected as the sodium-potassium material, and the chances for contact with oxygen were minimized by filling the inside of the valve directly from the bottle via a filling nozzle. The filling amount was set at 50% of the hollow volume in consideration of fluidity inside the hollow area.

A press machine that fits the inner plug and stem end plug was located inside the glove box, and sealed the valve after NaK filling. Finally, the stem end was laser welded and completely sealed in a normal atmosphere.

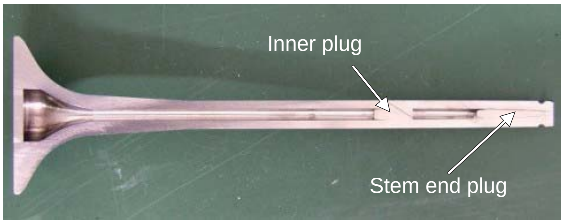


Fig. 3 Section of exhaust NaK valve

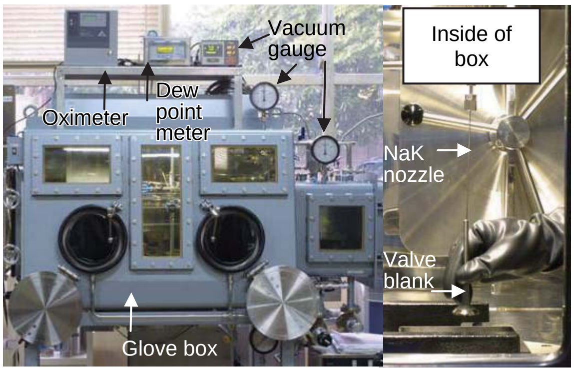


Fig. 4 NaK filling facility and filling operation

Table 1 Comparison of current and NaK valves

	Intake valve			Exhaust valve		
	Current	NaK	Difference	Current	NaK	Difference
Temperature (℃)	718	544	-174	840	737	-103
Fatigue strength (MPa)	227	615	+130%	126	211	+67%
Valve weight (g)	35.1	30.3	-4.8	32.7	28.1	-4.6

3. Engine Test Results

3.1. Temperature Reducing Effects of Sodium-potassium

To confirm the valve head temperature reducing effects, a quenched heat-resistant steel chip was embedded in the center of the valve head, and the valve temperature was estimated from the change in hardness after engine operation. Table 1 shows the results. The intake valve temperature was reduced by 174°C, which increased the material fatigue strength by 130%, enabling per-valve weight reduction of 4.8 g. The exhaust valve temperature was reduced by 103°C, which increased the material fatigue strength by 67%, enabling per-valve weight reduction of 4.6 g.

3.2. Engine Performance and Durability

Motoring head tests confirmed that the sodium-potassium intake valve increased the rotation upper limit by 500 rpm, but the output dropped by 5 kW in the dyno test. As the volumetric efficiency dropped, it is hypothesized that the valve head port side and stem temperatures rose due to the heat exchange of the sodium-potassium, which warmed the intake air and resulted in a drop in the intake air filling ratio.

The sodium-potassium exhaust valves finished a 2-race event as an engine durability test, but hairline cracks were evident in the welded area of the stem end. Since the submission deadline for engine spec homologation that would freeze the specifications was approaching and there was insufficient time to implement crack countermeasures, the exhaust valve was not applied.

4. Conclusion

(1) Valve-head welding technology was established that does not lower the parts strength, and hollowing achieved a weight reduction of 4.4 g compared to a Ti solid valve. The developed hollow valve was applied in races from the first race of 2006.

(2) A high-quality oxidation resistant process was completed for titanium sodium-potassium valves. Cooling effects of 100°C or more in high-temperature areas of the valve were confirmed, enabling weight reduction of 4.8 g per intake valve, and 4.6 g per exhaust valve.

(3) Sodium-potassium filling was confirmed to provide sufficient cooling effects. However, output dropped in some cases, and one factor for this is thought to be a drop in volumetric efficiency in the intake valve.

Reference

(1) Japan Society of Corrosion Engineering: Fusyoku bousyoku handobukku, Maruzen, p. 32 (2000) (in Japanese)

■ Author ■

Taihei SANADA

Development of DLC Coating on Camshaft and Rocker Arm

Naoaki ITO* Masaomi YONEHARA* Kazushige YAKUBO*

ABSTRACT

Among the Formula One engine valve train parts, the camshaft and rocker arm are subject to severe sliding conditions and constant demands for higher rotating speeds, so it was necessary to increase endurance reliability. Therefore, diamond-like carbon (DLC) coating films that strengthen the surface were developed, the film compositions and hardness balance were optimized, and the coating films were applied. As a result, friction was reduced by a total of approximately 5 kW, and endurance reliability was achieved that enables continuous camshaft use in 4-Race events.

1. Introduction

The camshaft and rocker arm are Formula One engine valve train parts that are subject to severe sliding conditions. The sliding conditions in the boundary lubrication area reach a contact pressure of 1 GPa or more, and a PV (contact pressure x sliding velocity) value of 20000 MPa m/s or more. Figure 1 shows the position of the sliding conditions on a Stribeck curve.

To secure sufficient endurance reliability under these conditions, it was necessary to cover the parts with a tough coating film that does not experience scuffing, even when the oil film breaks, and does not wear under high speed and high contact pressure sliding conditions. For this reason, DLC coatings have been used since 2002.

Engines are becoming increasingly high power and high speed, a pneumatic valve return system (J-VLV mechanism) using an orifice has been applied, and Formula One regulations also prescribe the continuous use of a single engine in 2-Race events, so further increases in durability were required. This increased the need to develop a DLC coating for the camshaft and rocker arm.

2. DLC Specifications

2.1. Approach towards DLC Layer Configuration

The sliding surfaces of both the cam robe and the rocker arm slipper were treated with DLC, and this was confirmed to increase wear toughness and reduce friction for both parts. Figure 2 shows the film configuration concept. To realize uniform contact, the top layer forms a low-hardness running-in layer. In addition, the DLC layer that serves as the main sliding layer achieves a balance between hardness and toughness, and the

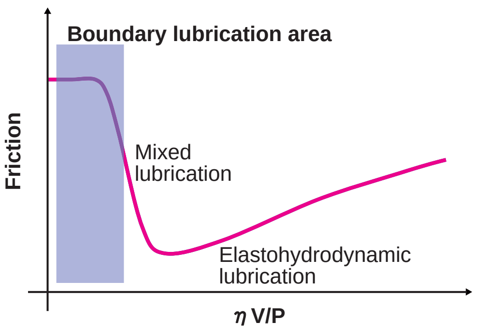


Fig. 1 Stribeck curve

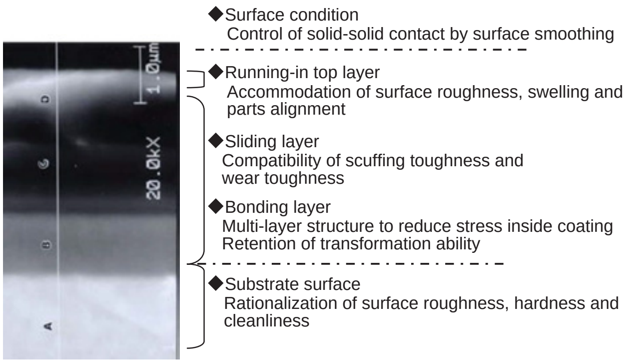


Fig. 2 Concept of DLC and bonding layer

* Automobile R&D Center

bonding layer, which has a hardness gradient, is formed by the spattering method to increase the scuffing toughness and adhesion strength of the DLC film.

2.2. DLC Treatment of Parts

For DLC to slide well against each other, the film hardness balance between them must be optimized. For example, DLC with the same hardness or that have topcoats that are too hard will attack each other, which is not desirable.

The combination of DLC specifications was evaluated and confirmed by basic sliding tests performed under high contact pressure conditions that simulated the actual sliding conditions. The results showed that a hardness balance of approximately 1.3 times in terms of the main sliding layer hardness reduced friction and provided high seizing durability. Figure 3 shows the hardness profiles of IDV38S, made by ICS Corporation, and DF1, made by Kobe Steel, Ltd., which were selected as the combination of DLC specifications.

The DLC treatment for each part was as follows. The camshaft was treated with the high hardness IDV38S, and the rocker arm slipper surface was treated with DF1, which has a lower hardness than IDV38S. This was done so based on the rule of thumb that when these parts slide, the camshaft experiences greater damage. Figure 4 shows the parts appearance after DLC treatment and an image of sliding. Figure 5 shows the DLC film configurations.

In addition, coating film quality is important for camshaft and rocker arm sliding, as DLC peeling trouble occurs for films with quality issues. Therefore, quality assurance methods were established for film uniformity, adhesion strength, wear toughness and other items; quality check samples were created that determine the allowable number, size and location of pinholes; and the quality was controlled.

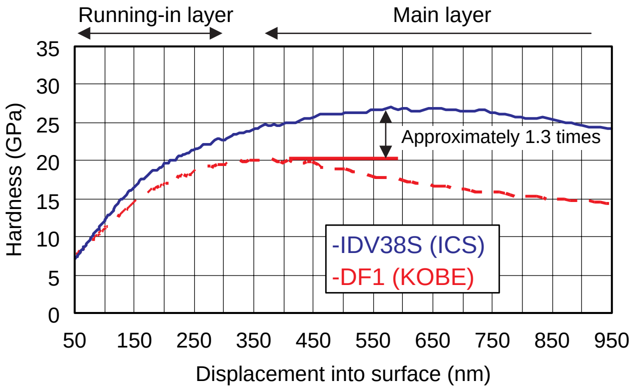


Fig. 3 Coating hardness of IDV38S and DF1

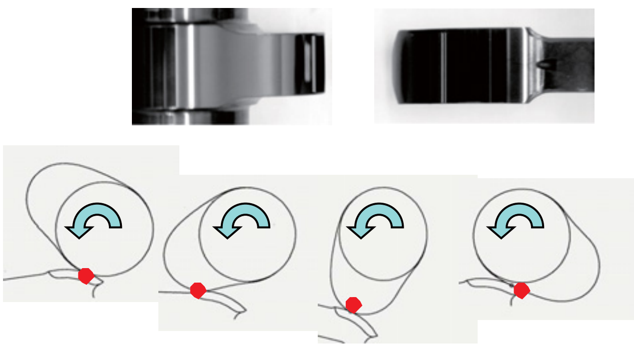


Fig. 4 Parts after DLC coating

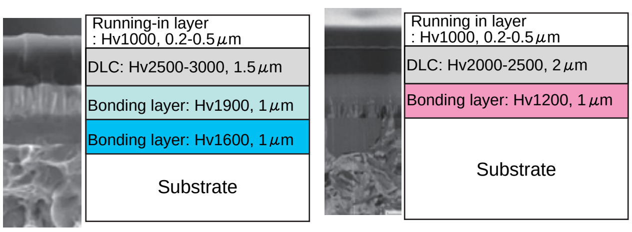


Fig. 5 IDV38S and DF1 layer configuration

3. Confirmation of Effects

An IDV38S-treated camshaft and DF1-treated rocker arm were introduced in the second half of 2006, and sliding reliability was assured under conditions of a contact pressure of 1.37 GPa and a PV value of 22 000 MPa m/s. Constant development led to yearly increases in the limit contact pressure, enabling high load and high speed operation at a contact pressure 17% higher and a PV value 32% higher than that of the combination of DLC specifications for 2004 (Fig. 6).

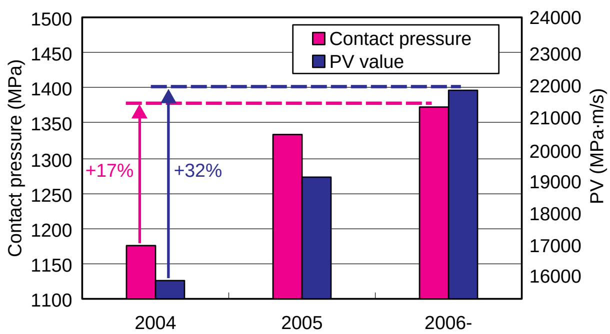


Fig. 6 Change in contact pressure and PV value

4. Conclusion

As Formula One engines evolve to higher output and higher speeds, steady progress was promoted to develop camshaft and rocker arm DLC that can withstand operation under harsh sliding conditions. As a result, performance was enhanced by a total of approximately 5 kW. In addition, these cam parts have been used in 4-Race events since 2007 with the aim of reducing costs, and the DLC coatings were able to assure sufficient endurance reliability.

Acknowledgements

The authors would like to express their deep thanks to ICS Corporation and Kobe Steel, Ltd., for their cooperation with the production technology and quality assurance associated with these parts.

■ Author ■

Naoaki ITO

Masaomi YONEHARA

Kazushige YAKUBO

Development of Laser Clad Welded Valve Seat

Nobuyuki IMAI*　Hiroaki NISHIDA**

ABSTRACT

Laser clad welded valve seats can expand the flexibility of layout compared to conventional press fit valve seats. The slot form, the shape and amount of the powder to be supplied, and the laser output parameters were optimized and balanced, quality assurance requirements were established, and the valve diameter was expanded by 1 mm. This achieved an increase in engine power of 6 kW at high water temperatures. In addition, the continuous use of one engine over the distance for two Formula One race events was supported by enhancing the powder materials.

1. Introduction

The valve seat (hereafter, "seat") pressed fit require sufficient wall thickness around the seat press fitting area of the cylinder head (hereafter, "head"), and also stiffness of annular rigidity of the seat itself. For this reason, the flexibility of the combustion chamber layout, including the valve diameter, valve pitch, and port angle, was limited. In addition, the area between the seat and the head is crimped, so heat transfer is insufficient between the valve, seat and head. This development realized a large diameter valve layout by establishing laser clad welded seat technology that forms the seat directly on the head, and also aimed to reduce weight by reducing the head wall thickness and increase engine power by enhancing cooling performance.

Figure 1 shows the press fit seat port section, and Fig. 2 shows the laser clad welded seat port section. Laser clad welding enabled a thinner water passage wall thickness.

Fig. 1　Layout of press fit valve seat

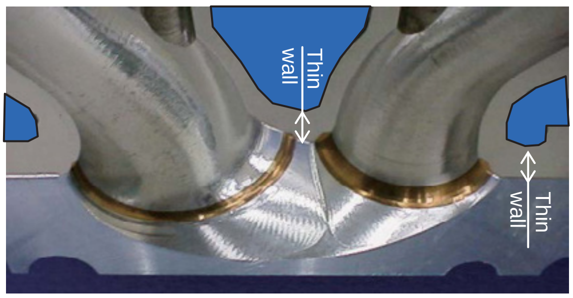


Fig. 2　Layout of laser clad welded valve seat

* Automobile R&D Center
** Aircraft Engine R&D Center

2. Establishment of Laser Clad Welded Seat Technology and Quality Assurance

The form of the slot used to clad the powder, the diameter of the powder to be supplied, the manufacturing conditions for welding the powder, and the quality requirements were set.

2.1. Laser Clad Slot Form

Figure 3 shows the laser clad slot form. The powder is collected in the center in a symmetrical form. When the clad thickness differs at the center and both edges of the slot, the welding time also differs. This may result in non-welds at the bottom of the slot, so the slot bottom form was an arc and the slot walls formed an obtuse angle. In addition, excess powder may likewise produce a difference in thickness, so the slot height was set at 4 mm to prevent unnecessary powder from collecting.

2.2. Powder Supplied for Laser Clad Welding

A copper alloy in the form of a spherical powder with a diameter of 80 ±5 μm was used as the clad powder. The powder was supplied using a vibration powder feeder, and supply was controlled at an accuracy of 0.005 g/s. This enabled the manufacture of homogeneous laser clad welded seats.

To support the continuous use of one engine over the distance for two Formula One race events, seat wear resistance was enhanced by mixing stellite and tribaloy powder into the copper alloy powder. Figure 4 shows the transition in the development specifications. Efforts were made to simultaneously achieve wear resistance and fracture toughness in accordance with changes in the valve train system specifications.

2.3. Laser Clad Welding Conditions

The welding process is an important process for the laser clad, and is controlled by the laser output and the amount of powder. In particular, the start point overlaps with end point, so start area of laser clad form must be inclined. Therefore, cladding start point was performed by gradually increasing the amount of powder according to the form, and increasing the laser output power in steps. Figure 5 shows this image drawing.

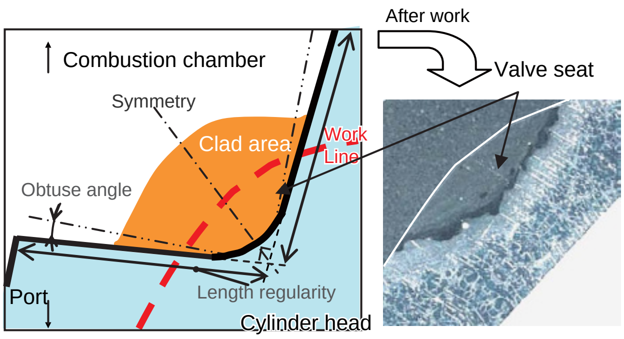


Fig. 3 Laser clad welding slot form

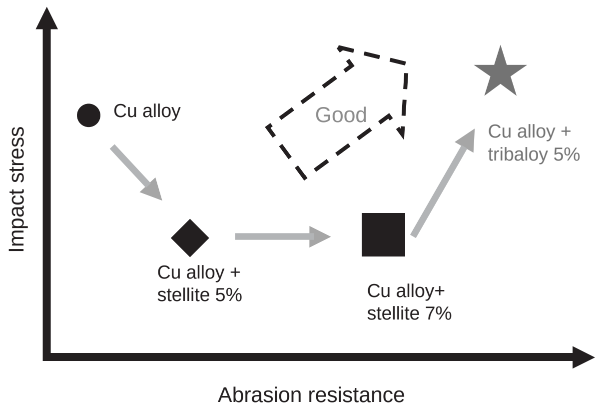


Fig. 4 Valve seat development corresponding to higher loads

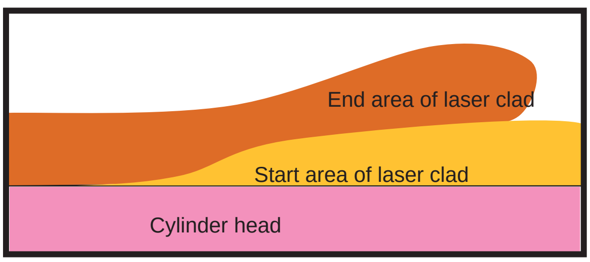


Fig. 5 Start and end form of laser clad

2.4. Quality Assurance of Laser Clad Welded Seats

Low quality in the start area may result in seat defects and lead to engine blowout or reduced engine power.

Non-welds can result from an excess powder amount or insufficient laser output, and brittleness can result from an insufficient powder amount or excessive laser output. Figure 6 shows examples of these cases. Non-weld and embrittlement layer limit standards of quality were prepared by analyzing the materials of seats in which defects actually occurred.

Limit standards were judged by analyzing the cutting planes of lot samples. The non-weld limit standard was no pinholes of diameter ø0.3 mm or more, and the embrittlement layer limit standard was no embrittlement 0.4 mm or thicker as obtained by measuring the hardness of the start area at the two points 1 mm and 4 mm from the start point. Figure 7 shows the limit standard of quality.

Fig. 6 Limit quality examples

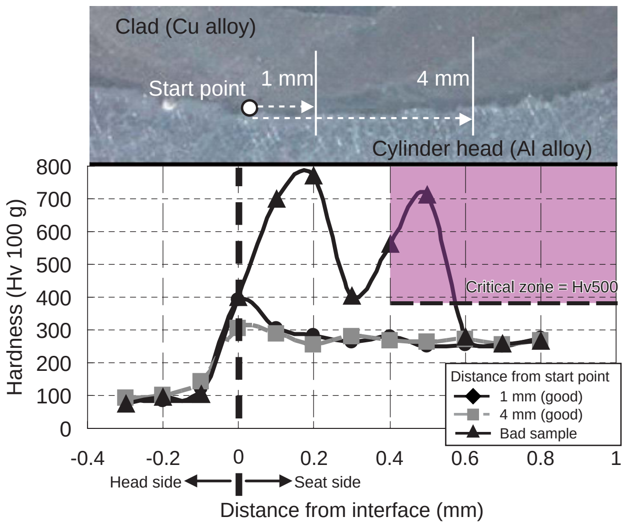


Fig. 7 Quality standard for dilution brittleness

3. Results

Laser clad welded seat technology increased the flexibility of the layout around the valve seat, enabled expansion of the valve diameter by 1 mm, and achieved a 6 kW increase in the engine power at a high water temperature of 120°C.

4. Conclusion

Laser clad welded valve seat manufacturing technology was established and applied to Formula One races from 2004. Hard particle powder was added and quality standards were established, which helped achieve the durability and reliability required to support the continuous use of one engine over the distance for two Formula One race events.

Acknowledgements

The authors wish to express their deep thanks to Tosei Electrobeam Co., Ltd., for cooperation in establishing the powder supply conditions.

■ Author ■

Nobuyuki IMAI

Hiroaki NISHIDA

Development of Titanium Exhaust Pipe

Takeshi MUNEMURA* Hiroshi YAMADA* Takayuki OHNUMA**

ABSTRACT

This paper discusses the use of titanium exhaust pipes for Formula One race cars in weight reduction. Newly developed processes for titanium material including heat treatment providing an acicular structure, oxidation resistant coating, sandwich press forming, and investment casting manufacturing have achieved a 20% weight reduction. This specification was not applied to race cars but provides technology for lightweight titanium exhaust pipes.

1. Background and Objective

Exhaust pipes are mounted on the upper part of Formula One race cars, and their component weight is high. Therefore, weight reduction has a large effect on enhancing chassis performance. Normally Inconel 625 (specific weight of 8.5 kg/cm^3), which has good high-temperature strength and workability, is used for exhaust pipes. In this research, titanium alloys, which have low specific weight (specific weight of 4.5 kg/cm^3), were selected as substitute material, in association with a variety of development efforts such as controlling the micro structure, enhancing oxidation resistance, and establishing casting and press forming method to reduce the weight of exhaust pipes.

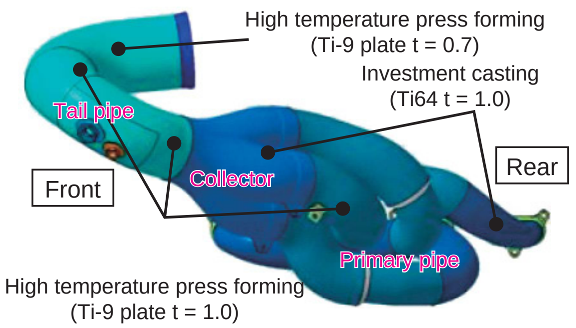


Fig. 1 Structure of exhaust pipe

* Automobile R&D Center
** Honda Engineering Co.,Ltd.

2. Technology Developed

2.1. High Temperature Strength

Figure 1 shows the structure of an exhaust system. The exhaust pipe is made up of a primary pipe, collector, and tail pipe. The materials used are KS Ti-9 plate material manufactured by Kobe Steel, Ltd. and investment casting Ti-64.

The strength of titanium alloys is lower in high temperature environments exceeding 600°C. In order to enhance high temperature strength and creep resistance, heat treatment providing an acicular structure was carried out at 1000°C on the final process and structural control was performed. As shown in Fig. 2, specific fatigue strength is higher than Inconel 625 at temperatures lower than 800°C and so material replacement was feasible.

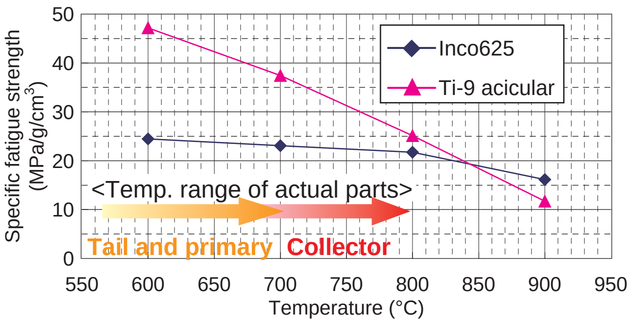


Fig. 2 Specific fatigue strength

2.2. Oxidation Resistance

The oxidation of titanium alloys accelerates in high temperature environments so an oxidation-resistant coating became necessary. An internal dispersion process of aluminum was developed, enabling a titanium aluminum layer to be formed on the surface, and providing good oxidation resistance characteristics for atmospheric exposure at 950°C. Figure 3 shows processes for applying both a stable thick oxidation-resistant coating over the entire surface, including the inside of the pipes, and heat treatment for material structure control. A 30 μm stable titanium aluminum oxidation-resistant coating as shown in Fig. 4 was achieved.

2.3. Workability

In order to reduce thickness and weight and enhance design freedom in the shape, press formed parts and investment casting parts were welded together.

2.3.1. Press forming

As the elongation of titanium alloys at room temperatures is small, hot press forming of 600°C or higher was necessary, under which the material elongation is equivalent to that of Inconel material. However, the heat capacity of the thin plate material is low and the temperature dropped quickly after it has been taken off from the furnace, so cracking could not be avoided. Press forming was enabled through sandwiching using stainless steel plates on both sides to hold temperature constant (Fig. 5). In order to achieve both holding temperature and press dimension accuracy, a thickness of 1.0 mm was used for the stainless steel plates. These stainless steel plates prevent direct contact between the die and the titanium alloy plates and so they also help to control galling.

2.3.2. Investment casting

There had been no major accomplishment in thin wall casting of a complex shaped collector and lost wax casting is unable to meet the short term delivery of shape modifications required for racing. The requirements for casting were met using a Rapid Prototyping (RP) model, and by applying the LeviCast method (suction method by negative pressure). A thickness reduction by chemical milling was also utilized, and together these methods made it possible to reach the target plate thickness of 1.0 mm and a 30-day delivery time. Figure 6 shows a summary of the process and Fig. 7 shows the external appearance of the parts.

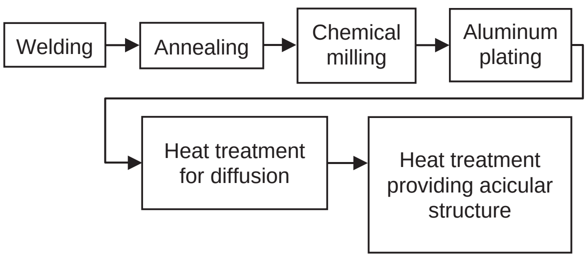


Fig. 3 Process of coating and heat treatment

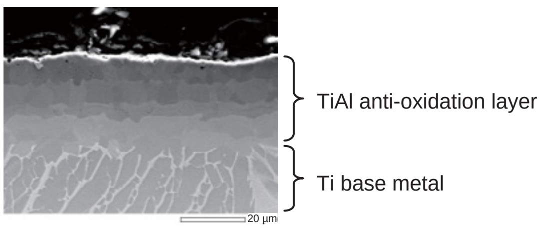


Fig. 4 Cross section of anti-oxidation coating

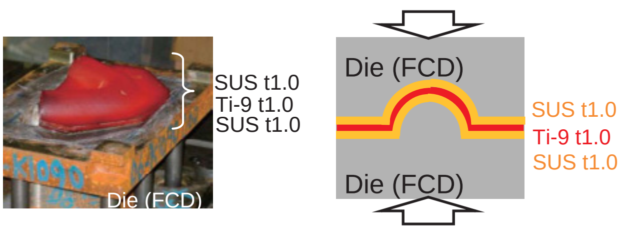


Fig. 5 Process of press forming

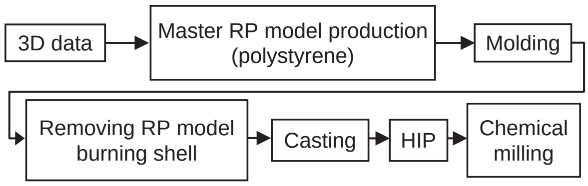


Fig. 6 Process of casting

Fig. 7 Casting collector and primary transitional pipe

3. Results

This development achieved a weight reduction of 1400 g (20%) per exhaust pipe assembly (Fig. 8).

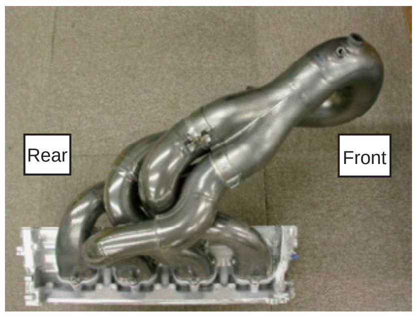


Fig. 8 Full titanium exhaust pipe assembly

4. Conclusion

The exhaust pipe for the 2007 specification had a unique layout: the exhaust pipe extending toward the engine in the forward direction, then loops back toward the rear. The loop-back area was broken during a track test and so this was not used for the race. However, in the 2006 specification (normal type; straight toward the rear of vehicle), the exhaust system based on this technology was used for 1200 km, confirming the durability of the technology.

This has shown that weight reduction is feasible by converting the exhaust pipe to titanium and optimizing the component shape.

■ Author ■

Takeshi MUNEMURA

Hiroshi YAMADA

Takayuki OHNUMA

Development of Lightweight and Low-friction Resin Materials for Oil Pump Rotor

Yu MURAI*   Masayuki TSUCHIYA*

ABSTRACT

The total weight of the scavenge and feed oil pump rotors fitted in Formula One V10 engines was previously more than 1 kg, leading to demands for weight savings and friction reductions. A polyethyl ethyl ketone material and a polyimide material have been used to manufacture the scavenge pump outer rotor in a two-stage injection molding process, and a polyethyl ethyl ketone extrusion material has been used to manufacture the feed pump outer rotor. This has resulted in a weight saving of 465 g and a friction reduction of 1.8 kW.

1. Introduction

Because dry sump lubrication is employed in Formula One engines, two oil pumps are used: A feed pump to increase pressure, and a scavenge ("scav") pump to scavenge oil. Both pumps are trochoid pumps. One set of rotors for the feed pump and seven sets of rotors for the scav pump are fitted in each V10 engine at the commencement of development, the total weight of which was previously more than 1 kg. The oil pump rotors have conventionally been manufactured from steel and aluminum (Al) sintered metal. The project discussed in this paper developed resin rotor materials in order to achieve weight savings and reduce friction.

2. Product Development

2.1. Material Evaluation

The materials used to manufacture pump rotors are required to display mechanical strength, dimensional stability, wear resistance, and sliding performance at oil temperatures of around 120 °C. Candidates for the resin material were selected from among high-strength and high temperature resistance super engineering thermoplastic resins and thermosetting resins. These were a polyethyl ethyl ketone (PEEK) injection material reinforced with 30% carbon fiber (CF), a PEEK extrusion material reinforced with 30% CF, a thermoplastic polyimide (PI) injection material reinforced with 30% CF, and a phenol resin injection material reinforced with 40% glass fiber (GF). High-temperature tensile tests and high-temperature tensile fatigue tests were conducted on test pieces formed into ASTM#1 dumbbells. The results of the fatigue tests are shown in Fig. 1. The materials selected were the PEEK injection material reinforced with 30% CF and the PI injection material reinforced with 30% CF, which displayed high strength in the temperature range in which the rotors would be used.

2.2. Development of Manufacturing Methods

Injection molding not only enables the achievement of high strength, but also obviates the need for the cutting process, thus reducing costs. However, because the outer rotor is thick, it was a challenge to achieve dimensional accuracy using normal injection molding, given the large surface sink. A two-stage process was therefore applied in which the inner part of the outer rotor is molded first, after which the outer part of the outer rotor is molded by enveloped casting. The two-stage injection molding method enables different

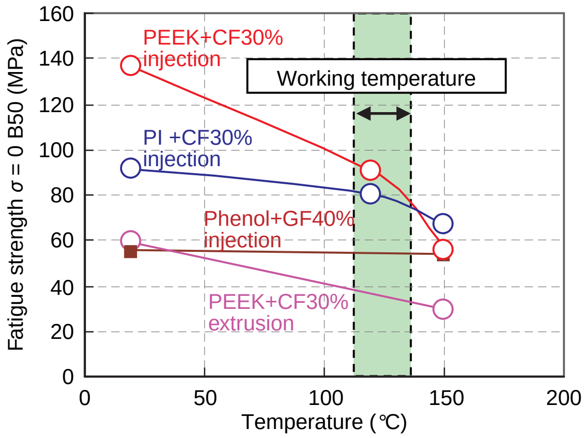


Fig. 1 Fatigue strength of resin materials

* Automobile R&D Center

materials to be used for the inner and outer parts. The PEEK material with its good wear resistance was employed for the inner part and the PI material with its good dimensional stability was employed for the outer part, enabling the issues of the surface sink and wear resistance to be resolved simultaneously. In addition, the method has enabled the accuracy of the teeth on the outer rotor to be controlled to within ±20 μm, and the teeth machining process can therefore be omitted (Fig. 2). The inner rotor is manufactured using a WE54 magnesium alloy extrusion material in order to achieve weight savings. A weight saving of 420 g (53%) was achieved against Al sintered materials in the case of the scav rotor.

The coefficient of thermal expansion of the resin materials varies with their method of manufacture, and they display strong anisotropy. For the injection materials, the coefficient of thermal expansion was measured in two directions to set the dimensions. For the extrusion materials displaying a high level of variation between lots, the coefficient of thermal expansion of different lots was measured in two directions in order to set the rotor thickness and the diameter dimensions.

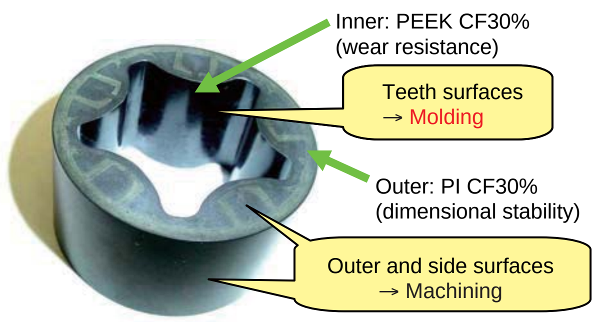


Fig. 2 Developed outer rotor manufactured using two-stage injection molding

3. Performance and Effects

An oil pump rig test was performed at an oil temperature of 100 °C using an injection molded resin scav rotor. As Fig. 3 shows, the results of the test indicated a 10% decline in torque. An engine test showed a decline in friction of 1.8 kW. No mechanical damage or significant performance declines were observed following engine durability tests.

There were concerns that the use of resin for the feed rotor would increase rotor wear due to increased pressure. The wear resistance of a variety of materials was tested in oil pump rig tests. The lowest amount of wear was displayed by a rotor with an outer rotor manufactured using the PEEK extrusion material, and an inner rotor manufactured using an Al sintered material. In addition, no significant performance declines were observed following engine durability tests using a rotor manufactured from these materials. A weight saving of 45 g (52%) was achieved in the feed pump.

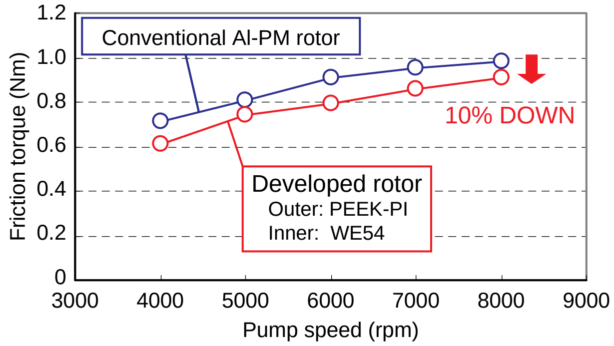


Fig. 3 Results of friction torque measurements

4. Conclusion

Resin oil pump rotors were developed, with the following outcomes.

(1) Scav pump rotor

The outer rotor is manufactured from PEEK and PI in a two-stage injection process, and the inner rotor is manufactured from WE54 magnesium alloy, resulting in a weight saving of 420 g and a friction reduction of 1.8 kW. The new rotor has been employed in races since the first race of 2004.

(2) Feed pump rotor

The outer rotor is manufactured from a PEEK extrusion material and the inner rotor is manufactured from the conventional Al sintered metal, resulting in a weight saving of 45 g. The new rotor has been employed in races since the 16th race of 2004.

Acknowledgments

The authors wish to take this opportunity to thank all the members of staff at Nichiei Co., Ltd., who made helpful proposals concerning the molding of the parts discussed in this paper, and who cooperated in the development of the manufacturing method for the parts.

■ Author ■

Yu MURAI

Masayuki TSUCHIYA

Research on Coolant Using Nano Additives and Heavy Water

Yu MURAI*

ABSTRACT

Research was done on coolants with the objective of increasing automobile radiator cooling efficiency and contributing to aerodynamics by enhancing the thermophysical properties of engine coolants.

The research confirmed the effectiveness of adding nano additives to enhance heat conductance, as well as using heavy water in solvent to increase specific heat of the coolant. Because evaluations of cooling performance in actual engines have not shown a clear difference from conventional coolants using water, a precise heat transfer measurement technology was established to clarify the factors affecting heat transfer, and from this, directions were established for the development of engine coolants.

1. Introduction

Formula One engine coolants have conventionally used water. Coolants circulate between the engine and radiator to radiate heat. Increasing the radiator's cooling efficiency makes it possible to reduce the surface area of the radiator duct aperture and lower aerodynamic resistance. Additionally, by making the radiator itself smaller, it is possible to design in ways that reduce weight and enhance aerodynamic characteristics, factors that could shorten lap times.

The target of the research was to enhance cooling efficiency by the equivalent of one rank of duct aperture surface area (equivalent to reducing coolant temperature by 4°C) by enhancing the coolant's thermophysical properties. This paper introduces research that, based on a patent[1] concerning heat transport fluids from research on mass-produced automobiles, sought to raise the coolant's heat transfer coefficient by producing experimental coolant whose thermophysical properties have been enhanced by adding nano additives at a high concentration and using heavy water.

2. Materials Development

The heat transport capacity of coolants is generally calculated from the specific heat per unit volume and from the heat transfer coefficient, as shown in Eq. (1). From Eq. (3), which is found by substituting Eq. (2) into Eq. (1), it is supposed that raising thermal conductivity, raising specific heat and lowering viscosity will cause heat transport capacity to increase.

Heat transport capacity =

$$\underbrace{[Cp \times \rho]}_{\text{Specific heat per unit volume}} \times \underbrace{[Pr^{1/3} \times \lambda \times (U/\nu)^{1/2}]}_{\text{Heat transfer coefficient}} \quad (1)$$

$$Pr = \frac{\nu \times Cp \times \rho}{\lambda} \quad (2)$$

Substituting (2) into (1),

Heat transport capacity =

$$[Cp \times \rho] \times [\nu^{-1/6} \times (Cp \times \rho)^{1/3} \times U^{1/2} \times \lambda^{2/3}] \quad (3)$$

Pr: Prandtl number; Cp: specific heat; ρ: density;
ν: dynamic viscosity
U: flow velocity; λ: thermal conductivity

The properties of the coolant experimentally produced are shown in Table 1. Based on a heat transport fluid patent, the aim was to increase the coolant's thermal conductivity by adding cup-stacked carbon nanotubes (below, CS-CNT). Coolant DW178 was test-produced in which dispersant was used in water to disseminate CS-CNT at a rate of 10 wt%, and the coolant was tested in a wind simulator[2], but the radiator became clogged and prevented evaluation. The cause is believed to be that the heat tolerance of the dispersant was insufficient in the coolant temperature zone of 100°C and up in which coolants are used in Formula One racing.

The aim in this research was to increase thermal conductivity by not using a dispersant whose heat tolerance was in question, and to increase specific heat

* Automobile R&D Center

Table 1　Thermal properties of coolant

Material	Additive	Dispersant	Thermal conductivity W/(mK)	Specific heat J/(cm³K)	Kinetic viscosity mm²/sec	Density g/cm³
H_2O (Base)	–	–	0.61	4.2	0.7	1.00
DW178	CNT 10%	$ROSO_3$-Na	0.74	4.0	1.4	1.06
$H_2O+Al_2O_3$	Al_2O_3 30%	–	0.73	4.0	1.5	1.27
D_2O	–	–	0.60	4.6	0.9	1.11
$D_2O+Al_2O_3$	Al_2O_3 30%	–	0.79 (+30%)	4.5 (+7.7%)	2.5 (+270%)	1.41

by using heavy water. Al_2O_3 nano additive (below, Al_2O_3), formulated by a sputter rapid cooling technique, was used as an additive since it offers better dispersion in water as compared to CS-CNT. To increase specific heat, heavy water (below, D_2O) was used that has 10% higher volume specific heat than ordinary water (below, H_2O). Adding Al_2O_3 to D_2O at a rate of 30 wt% increased thermal conductivity by 30% and specific heat by 7.7%. Viscosity, however, also increased by 270%.

Three formulations were tested in the wind simulator: Al_2O_3-added H_2O, Al_2O_3-added D_2O, and D_2O. The cylinder head temperature fell by 2 - 5°C, but at the same time engine output dropped by 4 - 8 kW, there was no correlation between test results, and reproducibility was low. Therefore, a technique was devised that evaluates the coolant, including the impact of the flow of each coolant, by separating heat conduction within solids and the heat conduction on the fluid/solid boundary.

3. Stand-alone evaluation results

A liquid-circulating heat exchange testing system was built as shown in Fig. 1. Figure 2 shows results of measuring heat transfer coefficient dependence on flow velocity. The development specifications suggest that cooling performance will be enhanced as compared to water in the flow velocity zone of 0.5 m/sec and below, but above 0.5 m/sec, cooling performance will decline. It is predicted that since the flow velocity in the cylinder head’s water channels is 2 - 6 m/sec, and that inside the radiator tubes is 1 - 2 m/sec, cooling efficiency will decline if this is used in Formula One coolants.

Based on the above, at low flow velocities, the impact of the liquid’s thermal conductivity will be great, but at high flow velocities, the impact of viscosity will

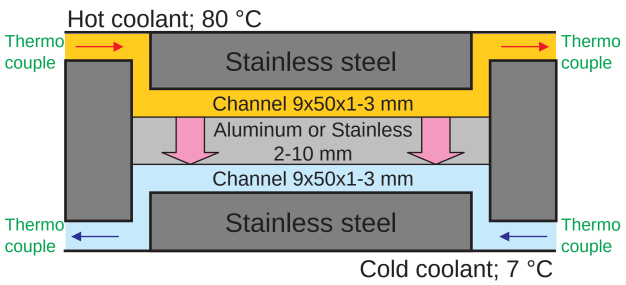


Fig. 1　Overview of heat transfer coefficient measurement apparatus

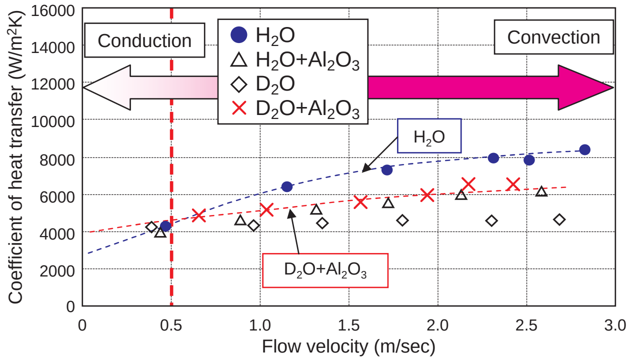


Fig. 2　Relationship between heat transfer and flow rate

be great. In other words, at low flow velocities, heat transfer within and among substances is dominant, but at high flow velocities, heat transfer from movement of the substances themselves is dominant. This made it clear that an automotive engine coolant must be formulated to strike a good balance between increasing thermal conductivity and reducing viscosity.

4. Conclusion

The newly developed nano fluid increases thermal conductivity, but at the same time causes an increase in viscosity because of the effect of interactions between additive particles, indicating that cooling efficiency would decline at the flow velocities at which Formula One engine coolants are used. Therefore, engine coolants need to be formulated to strike a good balance between increasing thermal conductivity and controlling increases in viscosity.

References

(1) JP, P2008-201834, A (2008)
(2) Nakamura, S., Motohashi, Y., Hayakawa, S.: Development of Wind Simulator Equipment for Analysis of Intake Phenomena in Formula One Engines, Honda R&D Technical Review, F1 Special (The Third Era Activities), p. 95-100

■ Author ■

Yu MURAI

Development of High-performance Gear Materials

Daiki KONAGAYA*　Tsutomu TANAHASHI*　Takashi TANAKA*

ABSTRACT

Two types of steel, a high pitting-toughness material displaying excellent performance in relation to the gear tooth fatigue characteristic and a high yield strength material displaying excellent tooth bending toughness, were developed for use in Formula One gearboxes. Optimization of the alloy composition increased pitting toughness, contributing to the achievement of a 740 g weight saving by enabling the thickness of the gear teeth to be reduced. The high yield strength gear material displayed a yield strength in excess of 1900 MPa, and was developed to ameliorate the issue of tooth root fatigue damage originating in bending of the gear teeth under the excessive input forces characteristic of Formula One racing. From 2008, regulations were changed to stipulate the use of a single gearbox for four race events, and the new material contributed to the achievement of increased long-distance reliability.

1. Introduction

At the commencement of the development program for Honda's Third Formula One Era, gears manufactured by Xtrac from a material corresponding to SNCM815VA were employed in the BAR Honda Formula One gearbox. However, in 2001, there was a frequent occurrence of tooth pitting on the final drive gear. The need for a high pitting-toughness gear material therefore increased, and attention was focused on the development of a new material.

Bending of the gear teeth due to extremely high input forces, as for example at the start of a race, also represented an issue. Changes in Formula One regulations in 2008 stipulated the use of a single gearbox for four race events, and a development program for a high yield strength gear material was therefore conducted in order to increase tooth bending toughness. The target value for tooth bending toughness was set 20% higher than the target for high pitting-toughness material, and the target value for 0.2% yield strength was set at 1900 MPa and above (40% higher than the target for high pitting-toughness material).

2. Development of High Pitting-toughness Material (LBHD-2E)

2.1. Developed Technology

In order to increase the pitting toughness of gear teeth, it is important that the hardness of the tooth surface should not decline (i.e. the tooth surface should not soften) even at high temperatures[1]. Table 1 shows the chemical composition of the developed steel, in which the amount of Si, Cr, and Mo, elements that increase resistance to temper softening, was optimized. A triple melt process was used to manufacture the steel in order to reduce the incidence of inclusions, which can be the origin points for tooth pitting, enabling production of a high-cleanliness material.

Table 1　Chemical composition

(mass%)

	C	Si	Mn	Ni	Cr	Mo
Conventional steel SNCM815VA	0.12-0.18	0.15-0.35	0.30-0.60	4.00-4.50	0.70-1.00	0.15-0.30
Developed steel LBHD-2E	0.30	1.50	0.35	2.00	1.50	0.75

2.2. Effects on Performance

Figure 1 shows the surface hardness profile of the conventional and developed steels after heating to 300˚C following carburization. The decline in surface hardness is controlled in the developed steel, LBHD-2E, in comparison to the conventional steel, SNCM815VA, and it displays a higher resistance to softening at high temperatures.

Figure 2 shows the results of a gear tooth pitting test using race gears. LBHD-2E displays more than twice the pitting toughness of SNCM815VA in terms of life cycles. LBHD-2E was employed in races from 2004 onwards, and enabled the achievement of a weight saving of 740 g through the reduction of the thickness of the gear teeth.

* Automobile R&D Center

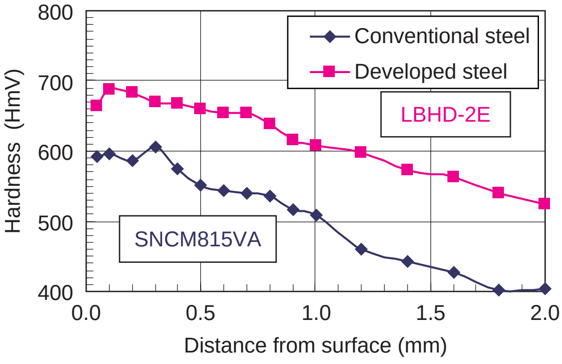


Fig. 1 Comparison of hardness profile of conventional and developed steel after 300 °C heating test

Table 2 Chemical composition

(mass%)

	C	Si	Mn	Ni	Cr	Mo
Conventional steel LBHD-2E	0.30	1.50	0.35	2.00	1.50	0.75
Developed steel 250STF5	0.52	2.50	0.70	1.80	0.80	0.40

Table 3 Comparison of mechanical properties

			0.2% YS MPa	TS MPa	EL %	Core hardness
Conventional steel	1	SNCM815VA	1029	1308	18.6	42 HRC
	2	LBHD-2E	1332	1795	15.9	52 HRC
Developed steel		250STF5	2137	2493	6.5	59 HRC

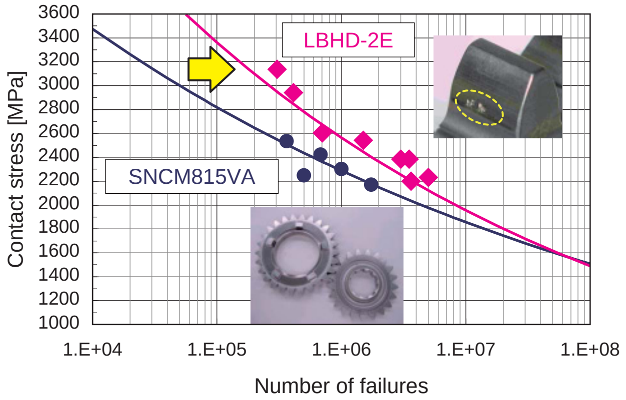


Fig. 2 SN diagram of gear tooth pitting

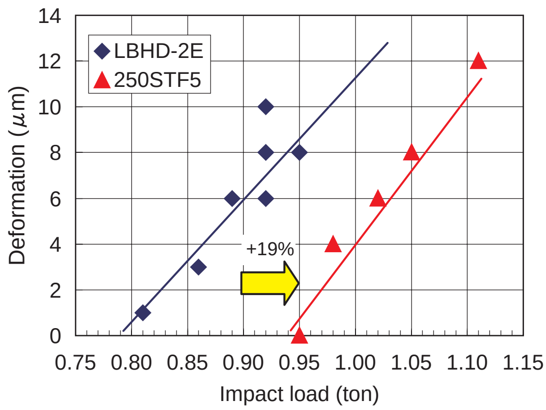


Fig. 3 Result of tooth bending test

3. Development of High Yield Strength Gear Material (250STF5)

3.1. Developed Material

The development of a high yield strength material was conducted with high-Si steel(2) as the base material, given that it enabled the achievement of a balance between high yield strength and high toughness, and was a material for which carburization could be used to harden the surface layer. The amount of C added was increased to achieve high hardness in the matrix, and the amount of Si added was increased to increase resistance to softening. Because Si also promotes grain oxidation, low pressure carburizing was selected as the carburizing method. Direct quenching was employed during the carburizing treatment and temperature conditions were optimized in order to prevent grain growth. Table 2 shows the chemical composition of 250STF5, the developed steel, and Table 3 shows its mechanical properties. The material satisfied the target of 1900 MPa or higher for 0.2% yield strength, reaching a figure of 2137 MPa.

3.2. Effects on Performance

Figure 3 shows impact loads and bending deformation in tooth bending tests conducted on a pair of gears. Compared to the previous steel, LBHD-2E, the load at which the teeth commenced bending was 19% higher for the developed steel.

Tooth bending tests in an actual gearbox also showed that the load at which the teeth commenced bending was 24.8% higher in the developed steel as a percentage of input torque. 250STF5 was employed in races from 2008.

4. Conclusion

A high pitting-toughness material was developed, and was employed for ratio gears in races from 2004. This helped to enable the thickness of the gear teeth to be reduced by 20%, contributing to the realization of a weight saving of 740 g in the gearbox as a whole. The material was later also employed for the final gears and the dog rings, contributing to the achievement of increased reliability against gear tooth pitting.

A high yield strength gear material was developed that exceeded development targets, achieving a yield strength of 2137 MPa, 60% greater than that of the previous steel. When the developed steel was employed in a gearbox, the load at which the teeth commenced bending was increased by 24.8%. This material was used for the 1st gears and shaft from the opening race of 2008, and contributed to the achievement of increased reliability over long distances, as necessitated by the regulation stipulating the use of a single gearbox for four-race events.

Acknowledgments

The authors wish to take this opportunity to thank the staff of Daido Steel Co., Ltd., for their generous assistance in materials development and gear tooth bending tests during the course of the projects discussed in this paper.

References

(1) Hanyuda, T., Nakamura, S.: Properties of Newly-developed Pitting Resistance Steel for Gears,“DPG1”, Electric Furnace Steel, Vol. 71, No. 1, p. 59-64 (2000)
(2) Kurebayashi, Y., Hatano, A.: Influence of Silicon and Vanadium Contents on Fatigue Properties of Carburizing Steels, Electric Furnace Steel, Vol. 67, No. 1, p. 34-43 (1996)

■ Author ■



Daiki KONAGAYA Tsutomu TANAHASHI Takashi TANAKA

Development of DLC for Transmission Gears

Tsutomu TANAHASHI*　Yasunori ONAHA*

ABSTRACT

A diamond-like carbon (DLC) coating has been developed for the transmission gears to increase their transmission efficiency. Using the carbon sputter method increased topcoat hardness and suppressed the formation of interfaces within the coating film, which enhanced durability under high surface pressures. This DLC was applied to the shift gear, final gear and bevel gear, and contributed to reducing friction. In addition, it secured the reliability needed to enable continuous use in the 4-Race events that Formula One regulations prescribe.

1. Introduction

Transmission loss is mainly due to losses from the bearing and gear, and gear sliding loss accounts for a large percentage of this. Therefore, DLC coatings that have good friction characteristics are thought to effectively increase transmission efficiency. However, the transmission gears (hereafter, "gears") have a high surface pressure load, so the DLC used by engine valve train parts lacked sufficient durability and could not be applied.

Therefore, to increase transmission efficiency by utilizing the DLC's good friction characteristics, the following targets were newly set, and a DLC coating for gears was developed.

(1) A coating film composition that can withstand surface pressure loads up to a maximum 2.2 GPa, enabling continuous use in the 4-Race events that Formula One regulations have prescribed since 2008

(2) Establishment of a coating method that forms a uniform film over the complex gear tooth shape

2. Development Concept

Engine valve trains that use a DLC coating perform reciprocal motion, so there is an oil film break point with a sliding speed of 0 m/s. This means that a hard bonding layer is required below the coating film to enhance scuff resistance.

On the other hand, the gear sliding environment has a high maximum surface pressure of 2.2 GPa, but the sliding speed does not go to 0 m/s, and there is no oil film break such as in a valve train. For this reason it was thought that a bonding layer is not required for the gear DLC, so efforts focused on developing a topcoat and coating film composition with an emphasis on wear resistance.

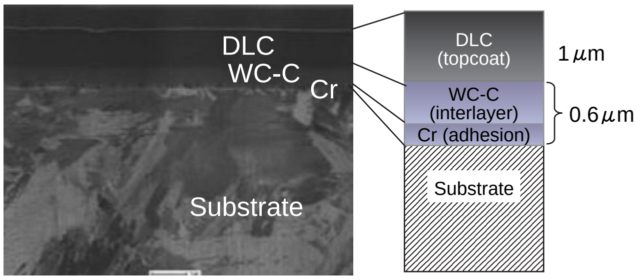
* Automobile R&D Center

3. Developed Technology

To increase adhesive strength between the coating film and the substrate, which is an issue under high surface pressures, the substrate surface roughness was finished to approximately 0.1 Ra, and surface contamination (oil components and minute corrosion) was eliminated by enhanced cleaning. As a result, this has resolved issues rooted in the substrate surface condition.

The nano-indenter hardness of the topcoat was increased to 60 GPa to enhance the wear resistance. From there down to the substrate, the coating film consists of an interlayer metal-carbon coating film (hereafter "WC-C") with a hardness that changes gradually until the interlayer joins with the chrome (Cr) adhesion layer. Figure 1 shows the DLC coating film composition.

The carbon sputter method was used continuously from the DLC topcoat to the WC-C. Hydrocarbon gas is generally used in many cases, but continuous sputter enabled the formation of interfaces to be suppressed, which prevented interlayer peeling under high surface pressures.



(a) FIB-SIM image　(b) Schematic diagram

Fig. 1　DLC cross section

In addition, gears have an uneven shape (especially the bevel gear), so uniformity of the coating thickness between the tooth tip and the tooth root is an issue. However, efforts were made to secure a uniform coating thickness by adjusting the bias voltage and work placement inside the DLC chamber. Figure 2 shows the coating thickness reduction ratio in the direction towards the tooth root, using the coating thickness at the tip of the bevel gear tooth as the reference. This shows that the developed DLC has a more uniform coating thickness compared to the conventional DLC.

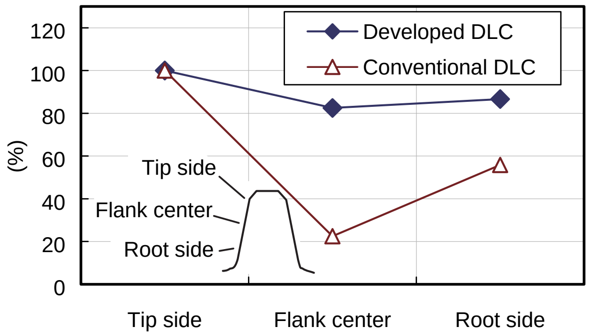


Fig. 2 Coating thickness distribution at tooth flank

4. Confirmation of Effects

Pin on disk seizing limit tests and actual transmission tests were performed. In the pin on disk seizing limit tests, oil was applied, wiped off, and then the load at which the friction coefficient rose due to seizing measured. As shown in Fig. 3, the seizing load increased by 40%. This is thought to be due to an increase in the adhesiveness between layers. In addition, engine tests using an actual transmission verified an increase in both performance and durability. Friction was reduced by a total of 3.3 kW as shown in Table 1, and a transmission efficiency of 97.0% was achieved.

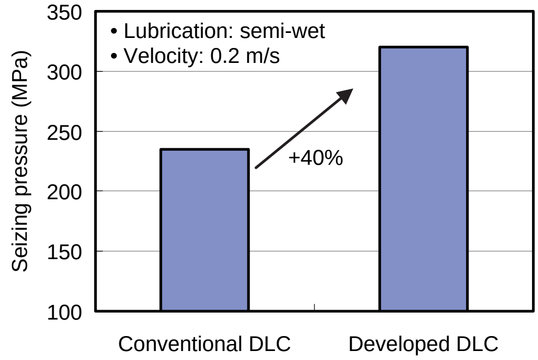
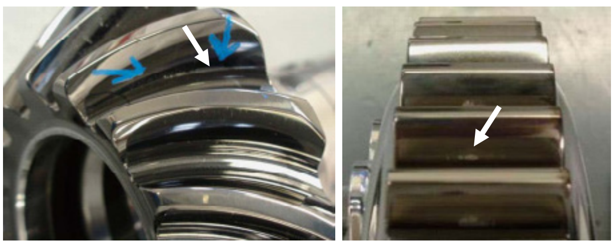


Fig. 3 Result of pin on disk test (DLC disk-DLC pin)

Table 1 Friction reduction of each DLC gear in actual gear box

	Shift gear	Bevel gear	Final gear
Friction	–0.8 kW	–1.3 kW	–1.2 kW

Figure 4 shows the condition of the bevel gear and the final gear after running in a 4-Race event. The figure indicates that the damage at the tooth roots is miniscule (indicated by the arrows in the figure), and that there is no issue with durability.



(a) Bevel gear (b) Final gear

Fig. 4 DLC after 4-race event

5. Conclusion

A DLC coating for transmission gears has been developed that has good durability under high surface pressures. This DLC was applied to the shift gear and the final gear from 2007, and to the bevel gear from 2008. Friction was reduced by a total of 3.3 kW, and a transmission efficiency of 97.0% was achieved. In addition, the developed DLC was confirmed to have sufficient durability for use in 4-Race events.

Acknowledgements

The authors wish to express their deep thanks to ICS Corporation for their cooperation with the hard DLC condition settings and gear coating production process.

Reference

(1) Suzuki, H., Ikenaga, M.: Jireidemanabu DLC seimakugijutsu, Nikkan Kogyo Shimbun, p. 44-50 (2003) (in Japanese)

■ Author ■

Tsutomu TANAHASHI

Yasunori ONAHA

Development of High Thermal Conductivity Material for Heatspreader

Yu MURAI* Masatoshi OKUMURA*

ABSTRACT

The DC-DC converter is key to developing small, lightweight power control units for Formula One hybrid systems. A new material consisting of a silver-impregnated diamond powder compact, was developed in order to enhance the heat dissipation performance of the DC-DC converter. The coefficient of thermal conductivity of the developed material is three times that of a conventional aluminum material. The use of the silver-diamond composite material in a heatspreader has reduced the temperature increase of the switching elements by 33%, offering a good prospect for the extension of continuous boost from 1.4 to 6.6 seconds, the maximum regulation figure.

1. Background and Aims

From 2009, the use of the Kinetic Energy Recovery System (KERS), a hybrid system for Formula One vehicles, is allowed by Formula One regulations. The KERS systems enable 60 kW of motor power and 400 kJ of regenerated energy to be used per lap to provide boost-up. Extension of the time for which continuous boost can be provided will help in overtaking on the straights, and will increase the degree of freedom in planning race strategies.

The KERS power control unit (PCU) consists of a DC-DC converter (VCU) that controls voltage, and an inverter that controls motor current. The heat load is highest on the switching elements in the VCU, which have a high operating frequency. The project discussed in this paper set out to enhance the heat dissipation performance of the VCU heatspreader in order to increase motor output to the regulation upper limit of 60 kW and enable 6.6 seconds of continuous boost (“full boost”).

2. Developed Technology

2.1. Study of Material

In order to enhance heat dissipation performance, the heatspreader material would be required to display a high coefficient of thermal conductivity and a coefficient of thermal expansion similar to that of the insulating plate of a circuit board. Aluminum nitride (AlN), a material with good thermal conductivity, is used for insulating plates. If the thermal expansion coefficient of the heatspreader was similar to that of AlN, the buffer used to relieve heat stress could be removed, and thermal resistance would be reduced (Fig. 1). As Table 1 shows, the production of a material from a composite of diamond and silver (Ag), both materials with high coefficients of thermal conductivity, enabled the achievement of a coefficient of thermal conductivity more than three times higher than the conventional material. In addition, the volume fraction of diamond, which has a coefficient of thermal expansion of 1.1 x 10^{-6} K^{-1}, has been set at 62% in the composite material, giving it a coefficient of thermal expansion of 5.5 x 10^{-6} K^{-1}, close to that of AlN (Fig. 2).

Table 1 Properties of materials for heatspreader

Material	Thermal conductivity W/mK	Coefficient of thermal expansion 10^{-6}/K	Density g/cm³	Volume fraction of diamond %
A6063	200	23.4	2.7	–
Ag-CD	630	5.5	6.0	62

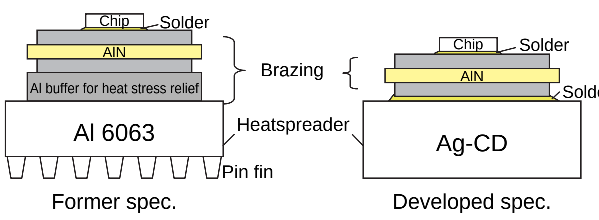


Fig. 1 Cross-section of power device

* Automobile R&D Center

2.2. Manufacturing Method for Material

A fine crystalline diamond powder with a mean grain diameter of 0.1 mm was used in order to enable maximal exploitation of the characteristics of diamond, which possesses a high coefficient of thermal conductivity. The diamond powder was pressurized inside the mold to increase the filling factor. The metal was impregnated into the material and a heat treatment was applied to prevent distortion, following which the compact was demolded. Figure 3 shows the microstructure of the Ag-CD material.

2.3. Development of Component

The diamond composite material displays a high level of hardness, making cutting and grinding extremely challenging. Water jet machining was used to drill the through holes used to fit the case and to cut the exterior, procedures which do not require high precision. Electric discharge machining was employed to produce the grooves used for the mounting of the thermocouple, the taper holes and the blind holes, for which precision is essential.

In addition to reducing the coefficient of thermal expansion of the heatspreader, specifications were employed for the potting resin, which gave it a low coefficient of thermal expansion of 10 x 10^{-6} K^{-1}. No performance decline was observed following tests of thermal shock resistance in power cycle tests.

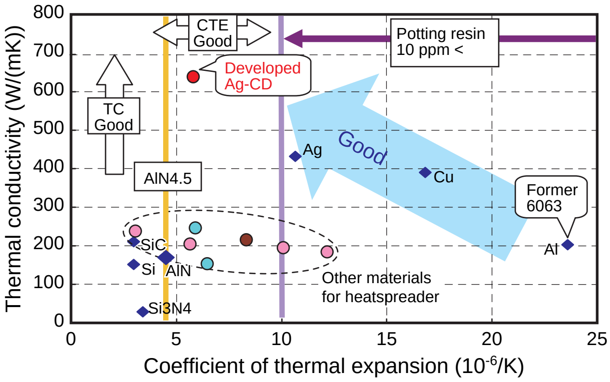


Fig. 2 Thermal conductivity and CTE of materials

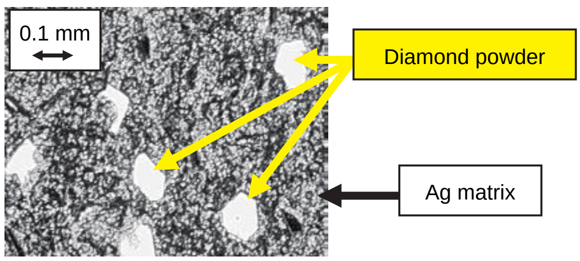


Fig. 3 Microstructure of Ag-CD material

4. Results

Heat dissipation performance was evaluated in thermal resistance tests of prototype power modules using heatspreaders manufactured from Al6063 and the Ag-CD material.

In a comparison using the same finless shape, the temperature increase of the switching elements was 35% lower in the power module using the Ag-CD heatspreader than in the module using the conventional Al6063 heatspreader. As Fig.4 shows, when the developed material was applied in the manufacture of a heatspreader that also incorporated a change in shape, the temperature increase of the switching elements was reduced by 33%. This indicates a solid potential for the extension of continuous boost time from 1.4 to 6.6 seconds per lap ("full boost") at a cooling water temperature of 80 °C or below, as assumed for an entire racing season.

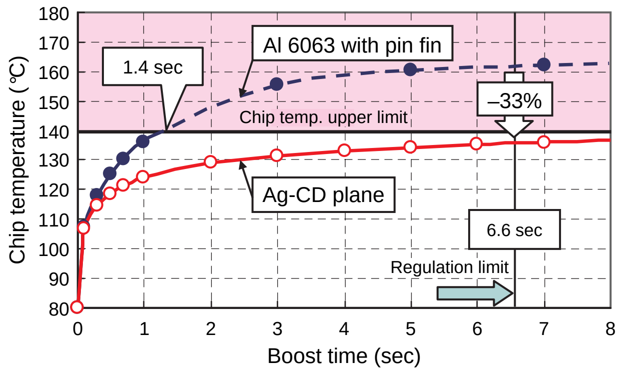


Fig. 4 Chip temperature after boost

5. Conclusion

An Ag-CD material displaying a higher coefficient of thermal conductivity and lower coefficient of thermal expansion than a conventional Al material was developed. The use of the developed Ag-CD material in the VCU heatspreader enhanced heat dissipation performance by 33%, offering good prospects for the ability to provide full boost for 6.6 seconds per lap for an entire racing season.

■ Author ■

Yu MURAI

Masatoshi OKUMURA

Development of High Induction Stator Core

Takehiro MIYOSHI*

ABSTRACT

Stator core iron loss was reduced by 60% by optimizing the thickness and microstructure of iron-cobalt alloy, which shows a high induction, and by developing a new adhesive and surface treatment technology. A manufacturing process was developed that satisfies the required magnetic characteristics and dimensional accuracy, and a new production system was constructed for laminated parts. The newly developed stator enabled a 20% reduction in the motor size.

1. Introduction

Application of hybrid systems to Formula One racing cars was planned from 2009, and the increased weight and effects on aerodynamics of adding motor-related parts was an issue. Reducing the size and weight of motor parts is essential for maintaining competitiveness, and the aim was to reduce the stator core size by applying a high induction permendur (Fe 49%, Co 49%, V 2%, hereafter "FeCo materials"). FeCo materials are used in the military, aerospace and other fields, but the steel type specifications are limited and there is also a long manufacturing lead time. Higher induction and lower iron loss were pursued for racing use, and efforts were also made to establish material specifications and a stator core manufacturing method that satisfy the required dimensional accuracy and productivity.

2. Developed Technologies

2.1. Material Development

Figure 1 shows the target material property values. Compared to Fe-Si ("Base"), existing FeCo materials ("original") have high iron loss, so the target was set at an iron loss equal to or less than that of Fe-Si. Iron loss is classified into the two types of eddy current loss and hysteresis loss, and eddy current loss is proportional to the square of the thickness, so eddy current loss can be reduced by reducing the thickness[(1)]. However, reducing the thickness raises the issues of a lower lamination factor, lower rolling and pressing quality, and lower productivity due to an increased number of layers. The iron loss values of cold rolled materials with a thickness of 10 μm to 150 μm were measured under conditions of 1.0 T and 400 Hz. The magnetic characteristics were measured using a toroidal-shaped laminated ring with an outer diameter of 25 mm and an inner diameter of 15 mm. Figure 2 shows the structure after rolling, and Fig. 3 shows the magnetic characteristics. The lowest iron loss value occurred at a thickness of 50 μm. Reducing the thickness lowers the eddy current loss, but at the same time the structure becomes finer, resulting in an increased hysteresis loss. This is thought to be why the total iron loss is not reduced at a thickness of 50 μm or less. The lamination factor drops when lamination is performed with a thickness of 50 μm, so the thickness was set at 100 μm in consideration of the balance between the lamination factor and iron loss.

An insulation layer is needed between the steel sheet layers to reduce eddy current loss, so thinness, insulation performance, and productivity are demanded. It was difficult to coat thin plates with a zinc phosphate insulating film for Fe-Si, and the required insulation performance could not be obtained. Various insulation treatments were evaluated, and the results showed that eddy current loss was most reduced by the oxide film. This film was formed by adding a heat treatment process after magnetic annealing.

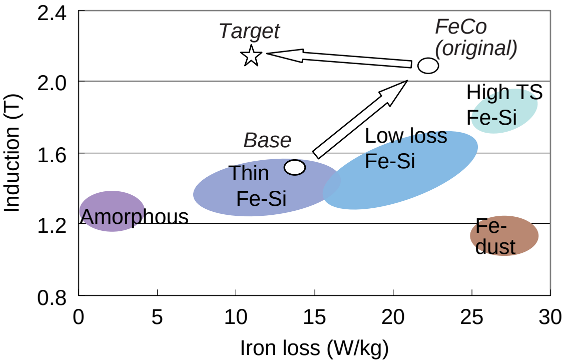


Fig. 1　Target properties

* Automobile R&D Center

Calking and welding produce strain that leads to a rise in iron loss, so the laminated materials were fixed using adhesive. This adhesive required adhesive strength, thinness, heat resistance and oil resistance, so a newly developed heat resistant adhesive with low viscosity was applied. Figure 4 shows the insulation film layers. The adhesive layer together with the insulation layers on both sides have a combined thickness of just 3 μm, which enabled a stator lamination factor of 97% or more.

2.2. Heat Treatment and Lamination Process

The heat produced by magnetic annealing causes the fixing jig to adhere and deform, so deformation was suppressed by adding a permeable ceramic. This work method enabled simultaneous heat treatment of multiple stators.

FeCo materials show order-disorder transformation by magnetic annealing, resulting in different dimensions before and after treatment. Press accuracy was achieved by using a die that is adjusted for heat deformation.

Securing flatness of the edge surfaces after adhesive bonding was difficult, but parts accuracy was achieved by increasing the jig accuracy, use of vacuum de-bubbling, and optimizing the core-turning and lamination procedure.

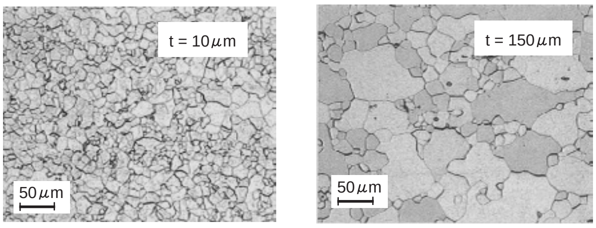


Fig. 2 Microstructure

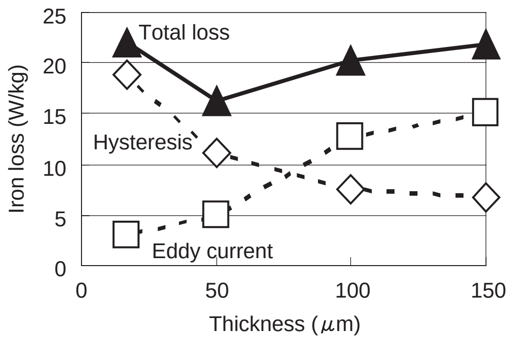


Fig. 3 Iron loss and thickness

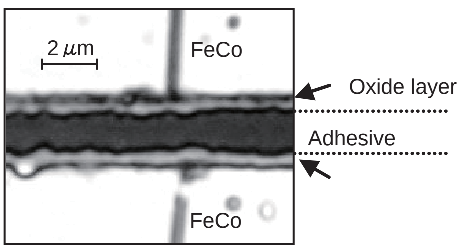


Fig. 4 Surface treatment and adhesive layer

3. Effects on Performance

Copper wire was wrapped around a core yoke, and the DC BH characteristics and iron loss value were measured to confirm the magnetic characteristics of an actual stator core. Figure 5 shows an exterior view of the test sample. The induction B50 was 2.0 T or more, and iron loss was 12 W/kg or less at W10/400, which met the targets. A motor using the newly developed materials achieved 15% higher torque with a size 20% smaller than that of a motor using Fe-Si materials.

Fig. 5 BH measurement sample

4. Conclusion

(1) Using existing FeCo materials as a base, the thickness and structure were optimized and a new surface treatment was developed. This simultaneously achieved high induction and low iron loss.
(2) A machining process for ultra-thin laminated materials was developed that both satisfies dimensional accuracy and has high productivity.
(3) The motor size was reduced by 20%.

Acknowledgments

The author wishes to express his deep thanks to related parties at Jeftec Co., Ltd., for their cooperation in establishing the manufacturing process for this part.

Reference

(1) Matsuoka, H.: Wakaru Denjikohan, Nippon Steel Corporation, p. 40 (1985) (in Japanese)

■ Author ■

Takehiro MIYOSHI

Weight Reduction Study for Side Panel of Formula One Monocoque

Yasuhiro YAMADA*

ABSTRACT

This paper describes the research conducted on Carbon Fiber Reinforced Plastic (CFRP) for the side panel of a Formula One monocoque.

The elongation ratio of a matrix resin was increased, thus increasing intrusion load 30% by load distribution. This enabled weight reduction by 1.1 kg for only a segment of the side panel.

In addition, weight reduction research was further conducted on single-sheet side panel, an alternative option to aluminum-honeycomb sandwich side panel, and the resulting technical issues were clarified.

1. Introduction

A several-mm-thick material called CFRP is used for the Formula One monocoque. CFRP is made by laying up multiple 100 μm-thick "prepreg" sheets, which is a carbon fiber fabric impregnated with matrix resin, and hardening the layered sheets. In addition, sandwich structures in which an aluminum honeycomb is sandwiched between CFRP increased strength and rigidity, and weight reduction is further required.

The Federation Internationale de l'Automobile (FIA) sets regulations for intrusion resistance in side impacts to help ensure side panel safety[1].

The FIA updates side panel regulations every few years in order to enhance safety, and each time the monocoque weight is increased.

Weight increases are expected to continue in the future following regulation updates, so the mechanism of intrusion resistance was studied, a matrix resin for side panels was developed, and weight reduction research using optimized panel structures was initiated.

2. Side Panels and Intrusion Regulations

In Fig. 1, the side panel is shown as the shaded portion. The side panel takes up over 50% of the surface area of the monocoque, thus it is a component that benefits greatly from weight reduction technology.

Figure 2 shows the setup for the intrusion test of the side panel.

For the intrusion test, a test panel is used to simulate a laminate on the side panel section of the monocoque. This test panel, which has a 550 mm x 550 mm outer dimension framed with a 25 mm wide metal frame for affixing bolts, is penetrated at a rate of 2 ± 1 mm/min. The FIA stipulates three regulation parameters: intrusion load, energy absorption, and fracture morphology. In 2005, all three of these parameters were increased to the following: intrusion load: "during the first 100 mm of displacement by the penetrator, the load must exceed 250 kN"; energy absorption: "during the first 100 mm of displacement by the penetrator, energy absorption must exceed 6 kJ"; and fracture morphology: "after the penetrator displaces 150 mm, there should be no damage to the border or to the fixture."

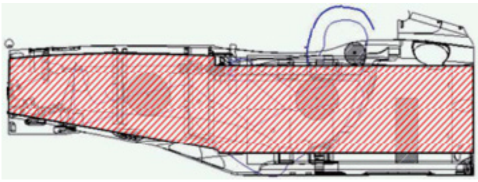


Fig. 1 Side panel area of monocoque

Fig. 2 Intrusion test setup

* Automobile R&D Center

Table 1 Side panel regulation transition

Year	Maximum load; first 100 mm of displacement (kN)	Energy absorption; first 100 mm of displacement (kJ)	Destructive morphology; after 150 mm displacement	Others
2001-2004	150	6	No destruction to frame	–
2005-2006	250			–
2007-				Additional extra secondary panel

Each racing team is conducting research to clear these regulations by weight reduction applications. Moreover, since 2007, the FIA has ordered the addition of FIA-designated, 6.2 mm-thick laminates to portions of the exterior side panels that satisfy the above requirements[2] (Table 1).

3. Technology Elements

The impact of carbon fiber-type, panel structure, aluminum honeycomb density, matrix resin of CFRP was confirmed.

3.1. Carbon Fibers

Intrusion tests were conducted on CFRP made from commercial carbon fibers. The carbon fiber series T1000G manufactured by Toray Industries, Inc. was confirmed as offering the highest tensile strength and best intrusion resistance among available carbon fibers.

3.2. Panel Structures (Sandwich Panel and Single Sheet)

Formula One monocoque is designed to receive the highest intrusion load when the inner wall of the panel is penetrated. Figure 3 shows intrusion test results for the sandwich panel and single sheet. For the sandwich panel, the top panel that corresponds to the outer wall panel of the monocoque was penetrated with 142 kN of load. Then, the bottom panel that corresponds to the inner wall panel of the monocoque was penetrated with 261 kN of load. Also, the single sheet was penetrated with 200 kN of load, 61 kN less than the bottom panel of the sandwich panel, despite being 1.3 mm thicker than the bottom panel of the sandwich panel. Furthermore, if the 10 mm thickness of the honeycomb is subtracted, the displacement of the single sheet is found to be 4 mm less than bottom panel penetration.

For the sandwich panel, the top panel and aluminum honeycomb is considered to disperse the load in the in-plane direction, so less stress would be generated at the bottom panel under the same load. In other words, the load distribution was expected to be an integral technical element for increasing intrusion resistance.

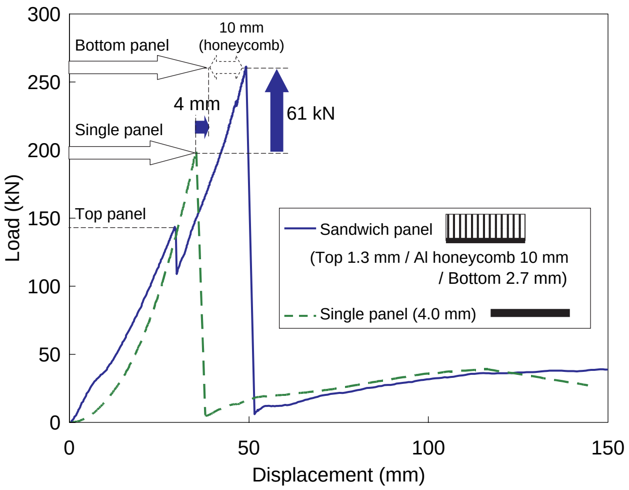


Fig. 3 Intrusion test results of sandwich / single panel

3.3. Aluminum Honeycomb Density

With sandwich panels that use low-density aluminum honeycomb for weight reduction, the intrusion load is decreased. That is, honeycombs with low density have low compressive strength and shear strength, and consequently, is unable to distribute the load acting on the top panel over a wide area of the bottom panel (Fig. 4). Technology is required that supports reduced intrusion loads for low-density honeycomb.

3.4. Matrix Resin

Pre-impregnated matrix resin, used as CFRP materials, takes full advantage of the special characteristics of carbon fiber, including high strength and high elastic modulus. However, the high strength and high elastic modulus characteristics of CFRP were expected to adversely affect intrusion resistance because of low load distribution. Therefore, CFRP trials were made with two kinds of epoxy resins, one with a high elastic modulus and another with a high elongation ratio. Then, intrusion resistance tests were conducted on the side panel. As anticipated, the results showed that matrix resin with high elongation ratio has greater intrusion resistance (Fig. 5).

Thus, it was confirmed that low-density honeycombs and matrix resin with excellent load distribution

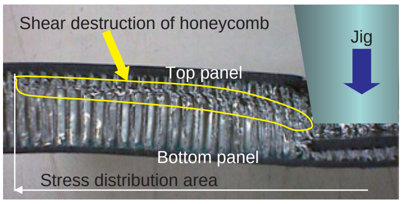


Fig. 4 Side panel section before bottom panel intrusion

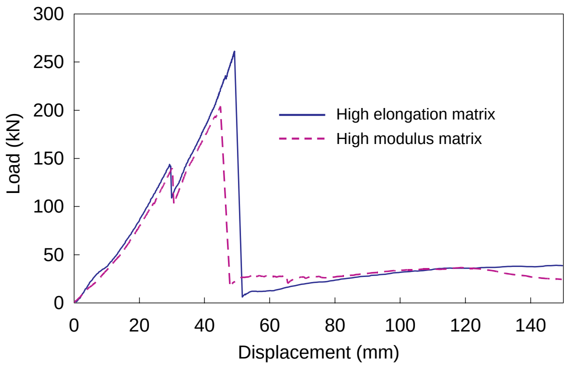


Fig. 5 Intrusion test results of high modulus/ elongation matrix CFRP panel

properties are a necessary technical element for side panel weight reduction.

4. Research on Matrix Resin for High Intrusion CFRP

An aluminum honeycomb with a density of 10 lb/ft^3 has been used for the side panel and weight reduction was conducted by using 8 lb/ft^3 of material. Research was then conducted under the concept of compensating for reduced intrusion loads due to decreased load distribution by increasing the elongation ratio of the matrix resin.

By optimizing the resin components, RH421 epoxy resin with a 4% elongation ratio manufactured by Nagase ChemteX Corporation was obtained for the final specification. Figure 6 shows the mechanical properties.

An intrusion resistance test was conducted on a test sandwich panel that used RH421 and an aluminum honeycomb with a density of 8 lb/ft^3. The results showed a 30% increase in intrusion load over the existing Cycom2020[(3)] product specification.

Figure 7 shows the panel intrusion holes. The Cycom2020 specification shows breaks at sharp angles around the intrusion hole area as opposed to the gentle bends exhibited by the RH421 specification.

It is thought that, because of reduced CFRP bending and the reduced compression elastic modulus (Fig. 8), bending and compression stress concentration, which are CFRP weak-points, did not occur and these stresses were successfully converted into tensile stresses. Thus, the high elongation ratio of the matrix resin allowed weight reduction of the aluminum honeycomb and the increased intrusion load to be achieved simultaneously.

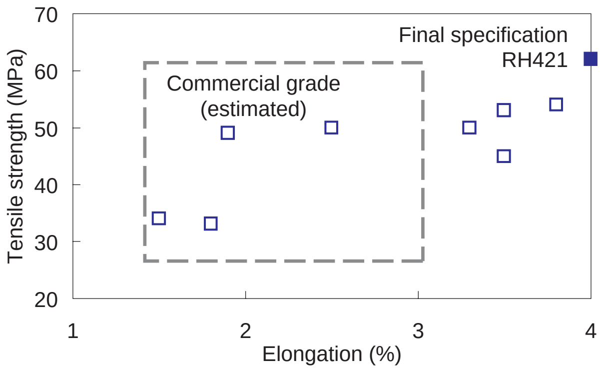


Fig. 6　Physical properties of trial epoxy resin

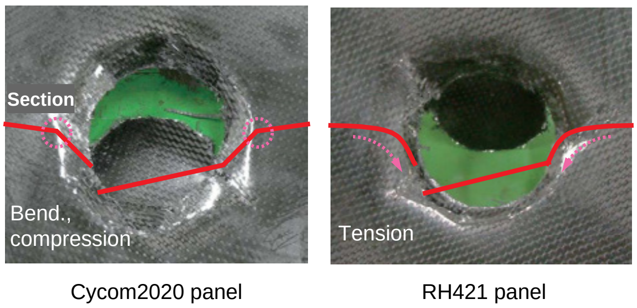


Fig. 7　Panel appearance after intrusion test

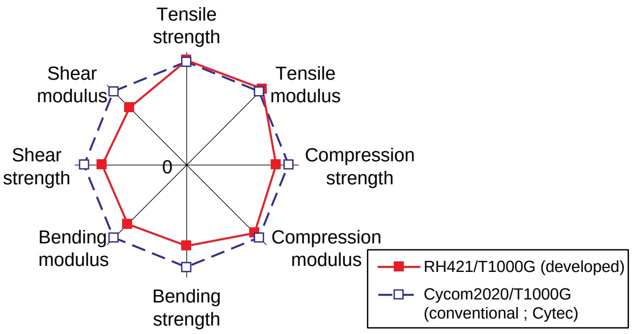


Fig. 8　Mechanical properties of CFRP

5. Research on Single-sheet Side Panel

Research was conducted on further weight reduction with the goal of conforming single-sheet side panels to regulations without using aluminum honeycombs.

5.1. Verification of Composite Results for Low Elastic Modulus Layers

Figure 9 shows the intrusion load for single-sheet CFRP. When the panel thickness was increased, the intrusion load increased but the intrusion load efficiency, obtained by dividing intrusion load by the panel thickness, decreased. This was thought to be caused because load distribution was decreased together with the increase in panel rigidity.

Functional materials similar to the aluminum honeycomb of the sandwich panel were researched to help distribute the load. Single-sheet side panels were made for trial purposes (Fig. 10). For the panels, an organic fiber layer with an elastic modulus less than that of carbon fiber was arranged in between the CFRP layers or on the CFRP outer layer. As a result, the intrusion

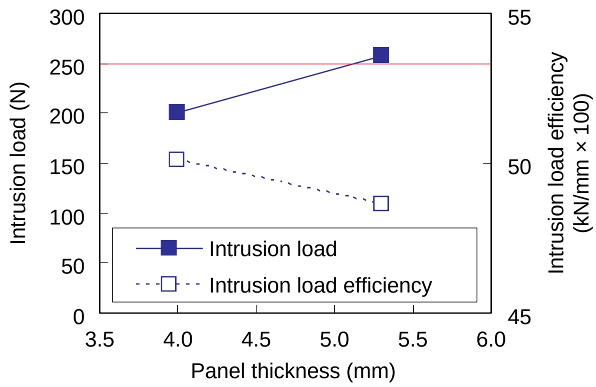


Fig. 9　Intrusion load and intrusion load efficiency of single panel

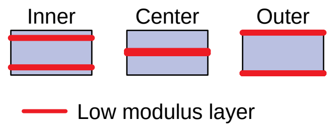


Fig. 10　Example of low modulus layer arrangement

load was increased more than for the single-sheet CFRP panel, with the exception of one specification, even though there was variation depending on the difference in the kind of organic fiber and arrangement (Fig. 11).

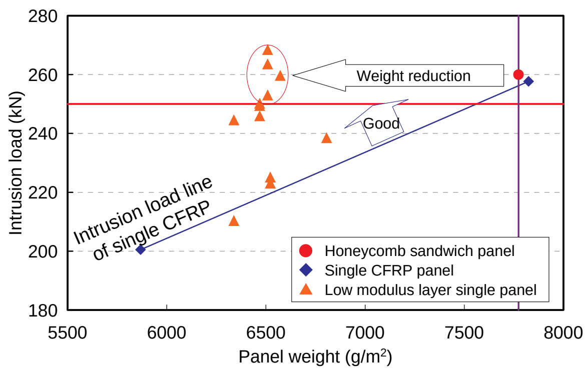


Fig. 11 Effect of low modulus layer arrangement for intrusion

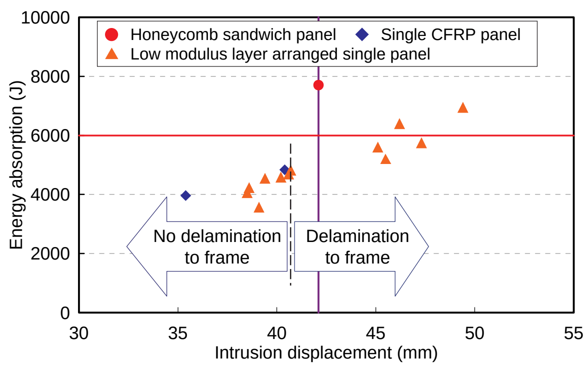


Fig. 12 Energy absorption of low modulus layer single panel

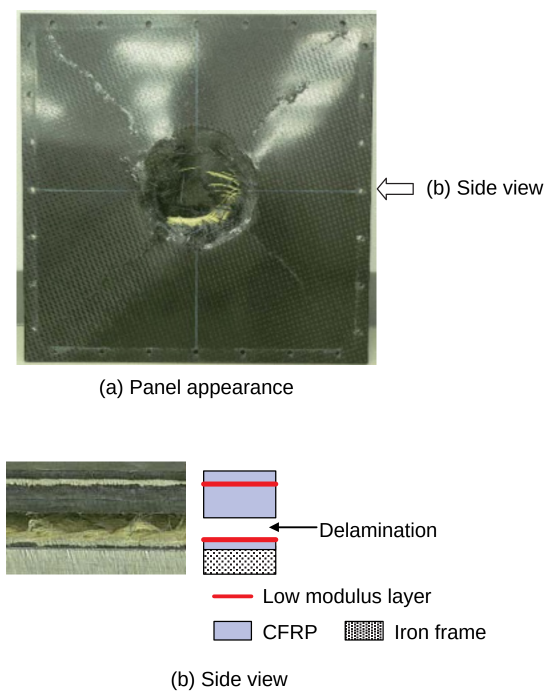


Fig. 13 Appearance after intrusion test of low modulus layer single panel

5.2. Issues of Single-sheet Low Elastic Modulus Composites

Figure 12 shows the remaining issues related to energy absorption and the fracture morphology for the side panel. As the amount of the energy absorption increases the intrusion displacement increases; however, because single sheets do not undergo a two-stage intrusion like the sandwich panel, the amount of the energy absorption is low. Also, when intrusion displacement increases, delamination is generated at the organic fiber layer and the carbon fiber layer interfaces and extends to the edge of the panel (Fig. 13). Because of this fracture morphology, regulations have not been achieved. Additional control technology for interface delamination is required in single-sheet manufacture for side panels.

6. Conclusion

The technology that maintains intrusion resistance with sandwich panels having low-density aluminum honeycomb was established by producing a high elongation ratio CFRP matrix resin.

Side panels with RH421 applied allowed the bottom panel to be thinner, and in combination with a low-density aluminum honeycomb achieved a total weight reduction of 1.1 kg. These specifications were officially approved by the FIA.

An actual test using a real vehicle with these side panels was performed and verified with no decrease in driving stability performance.

On the other hand, it was clear that interface delamination control technology is necessary regarding single-sheet construction that will allow further weight reduction.

References

(1) Federation Internationale de I'Automobile (FIA): Side intrusion test procedure, Appendix to the 2007 FIA Formula One Technical Regulations, Article 18. 6, p. 34-35 (2006)
(2) Federation Internationale de I'Automobile (FIA): Additional side intrusion panels, Appendix to the 2007 FIA Formula One Technical Regulations, Article 15. 4. 7, p. 26-27 (2006)
(3) http://www.cytec.com/

■ Author ■

Yasuhiro YAMADA

Development of Brake Caliper Production Process with High Strength Al-Li Material

Akihiro YANASE* Hiroshi YAMADA*

ABSTRACT

A production process was developed that uses high-strength Al-Li materials to reduce the brake caliper weight. Anisotropy of material strength due to metal flow was eliminated with 3-axis multi forging, and an 8% increase in fatigue strength was achieved by T6 heat treatment with 3-stage aging. This is expected to increase the service life to 4 times that of current calipers.

1. Introduction

Reducing a vehicle's unsprung weight contributes to enhancing its vehicle dynamics, so brake caliper weight reduction is demanded. A caliper development project was started with the aim of reducing the weight by 150 g per caliper (600 g per vehicle). As the current caliper materials, the 2099 materials of aluminum-lithium (Al-Li) alloy is used, which are high stiffness, low density aluminum materials. However, these materials have anisotropy of material strength, which posed design restrictions. Therefore, a forging process was established with the aims of eliminating anisotropy of material strength, and a heat treatment was developed with the aim of increasing strength.

2. Developed Technology

2.1. Selection of Materials

Figure 1 shows the material properties of the Al-Li alloy, and Fig. 2 shows the anisotropy of material strength from the fatigue strength in the longitudinal (L) and transverse (T) directions of the materials. The fatigue strength data of 2618 materials for the piston is also shown for comparison reference.

Under Formula One regulations, caliper materials are prescribed as aluminum materials with a Young's modulus of 80 GPa or less. As the current material, a 2099-T83 extrusion bar of Al-Li alloy is used that displays low density, high strength, and high Young's modulus. However, this material has anisotropy of material strength, and the low T direction strength restricted the design.

* Automobile R&D Center

2.2. Forging Process for Eliminating Anisotropy of Material Strength

Anisotropy of material strength could not be eliminated by 1-axis forging of the 2099 extrusion bar. Therefore, 3-axis multi forging was applied, which is an effective method for eliminating anisotropy of material

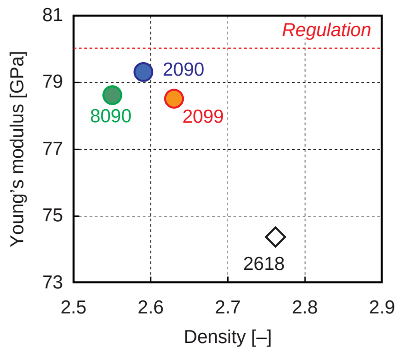


Fig. 1 Modulus and density

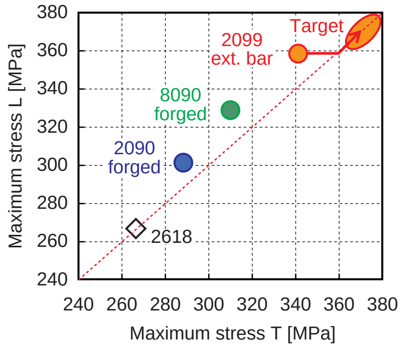


Fig. 2 Fatigue property

strength. Before die forging, 50% upset forging from the three X, Y and Z axes was performed to eliminate anisotropy of material strength by a hydraulic press. Figure 3 shows the set forging process.

2.3. Heat Treatment Conditions for Increasing Strength

The 2099 materials are a deposit strengthening type alloy, with σ' deposit phases (Al_3Li) and T_1 phase (Al_2CuLi) deposit phases. Cold compression to introduce a dislocation that serves as the deposit phase core is an effective means of increasing strength, but this is difficult to apply to calipers, which have a complex shape. Heat treatment with multi stage aging is also effective in generating a core for the deposit phase[(1)], so T6 heat treatment with 3-stage aging was developed with solution treatment for 2 hours at 549°C, followed by aging (treatment) for 48 hours at room temperature, 24 hours at 120°C, and then 100 hours at 165°C as the heat treatment conditions for the 2099 forged materials.

The status of each deposit phase was analyzed using a transmission electron microscope (TEM) and differential scanning calorimetry (DSC), and the following mechanisms of deposit strengthening were confirmed: 1) formation of a GP zone by room temperature aging, 2) σ' phase deposition by 120°C aging, and 3) T_1 phase deposition with the GP zone as the core by 165°C aging.

2.4. Evaluation of Material Properties of 2099 Forged Materials

Prototype 2099 forged materials for calipers were manufactured using the 3-axis multi forging process and T6 heat treatment with 3-stage aging. Test pieces were sampled in the L and T directions of the 2099 forged materials as shown in Fig. 5, and tensile-tensile fatigue tests were performed.

Figure 6 shows the results. The current 8090 and 2099 extrusion bars exhibit anisotropy of material strength, with the fatigue strength differing according to the location from where the test piece samples were taken. However, the developed 2099 forged materials introduced in this paper had the same fatigue strength regardless of sample location, so anisotropy of material strength was eliminated. In addition, an 8% increase in fatigue strength was confirmed relative to the T direction of the 2099 extrusion bars, which had been a design restriction. As a result, the strength in the weakest area of the part was increased, giving prospects for a four-fold increase in durability.

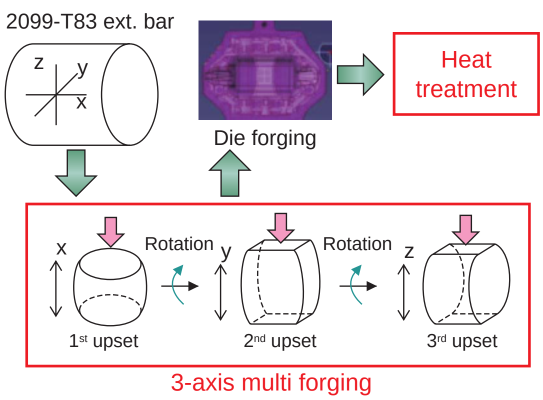


Fig. 3 Caliper forging process

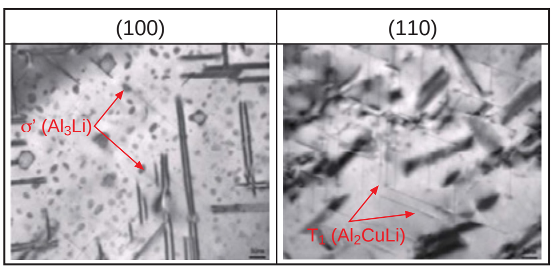


Fig. 4 TEM photograph of 2099 after 3 stage T6 (x 150000)

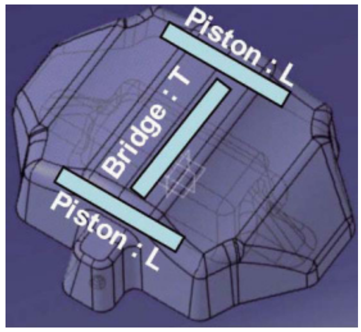


Fig. 5 Sampling regions

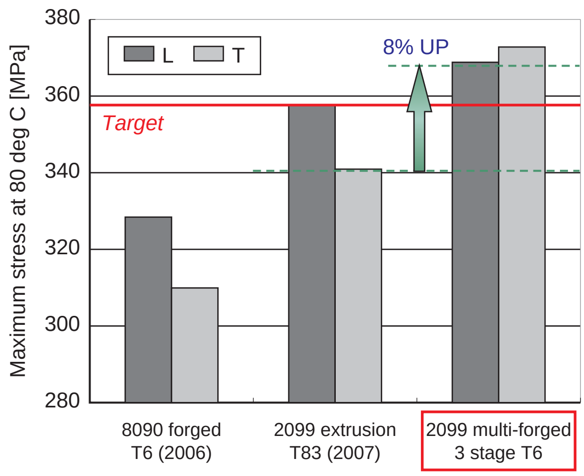


Fig. 6 Fatigue property of caliper body materials

3. Conclusion

This development established production process technology that eliminates anisotropy of material strength by 3-axis multi forging, and increases fatigue strength by 8% compared to the current materials by T6 heat treatment with 3-stage aging. Converted to service life, this increased durability by 4 times.

Reference

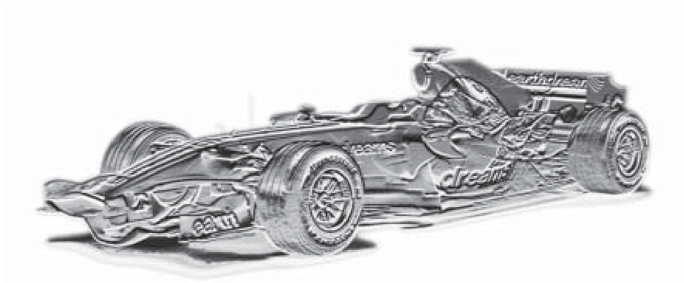
(1) Romios, M., Tiraschi, R., Parrish, C., Babel, H. W., Oqren, J. R., Es-Said, O. S.: Design of Multistep Aging Treatment of 2099 (C458) Al-Li Alloy, Journal of Materials Engineering and Performance, p. 641(2005)

■ Author ■

Akihiro YANASE

Hiroshi YAMADA

Descriptions of Race Management Technologies



Development of Lifecycle Management System for Racing Parts

Hidetada TANIGUCHI* Katsunori TAGUCHI* Masataka NODA*
Shigeo MIYAJIMA* Osamu WATANABE**

ABSTRACT

In racing operations, parts development and supply must be constantly reviewed in a short time with regards to changing circumstances in order to simultaneously promote upgrading and supplying of parts with the aim of maintaining competitiveness relative to other teams. For this reason, a part lifecycle management system was constructed to increase development speed and effectively utilize resources. This system aimed for centralized management of part plans and results data for all stages, from investigation of part design specifications to drawing issue, production plans, arrangement and purchasing, procurement and manufacturing, receiving inspection, inventory control, assembly, usage history, and disposal. Centralized data management enabled instant sharing of information across the Racing Division, and provided prospects for creating an environment that enables accurate and prompt decision making even at racing sites.

1. Introduction

The Racing Division of Honda's Automobile R&D Center (Tochigi) developed, assembled and supplied engines, developed advanced chassis technologies, and also developed and supplied internal gears and gear shifting parts for the gear box to the Honda Racing Formula One Team (HRF1). Engines were also assembled and supplied by Honda Racing Development Ltd. (HRD) in England, and in addition to parts development and supply to race sites from Tochigi, locally procured parts were supplied from HRD.

In racing operations, development speed is demanded to maintain competitiveness, and this assumes a unique operations flow (Fig. 1). Parts with multiple specifications are constantly designed, arranged and tested, then just before assembly the test results are reviewed to select the optimum specifications from among the current inventory and parts scheduled for delivery, and the selected parts are assembled into products and supplied. In addition, arrangements must be made concurrently for development parts and racing supply parts, and assembly specifications also change frequently. This means that a diverse and large volume of inventory is essential in securing a constant stock of parts that might be used.

Assembly specifications and inventory conditions change constantly, including support for races that are held every other week, and it is also necessary to accurately record part assembly results and usage histories to provide feedback on trouble information. Parts shifting and plan changes based on tests and race results also had to be shared between Tochigi and HRD, so speed and accuracy of information were also demanded.

In addition, the recent application of regulations limiting engine usage had tended to increase the need to manage parts with the same part number by quality rank, and to perform individual part lifecycle management.

The system constructed for production car development operations was also used to manage racing parts until recently, but no fundamental changes had been made to suit the environmental changes and operation contents unique to racing operations.

In the operational flow of production car development, the prototype assembly specifications and assembly schedule are determined, and then the parts resources needed for assembly are secured before purchasing (Fig. 2). For this reason, the constructed conventional system lacked the flexibility required by racing operations, and constantly changing information could not be accurately reflected.

To overcome this issue, the information needed by each department was provided separately using other system ledgers and tools, and had to be collated manually based on experience or by using large numbers of people. In addition, when the need for coordination arose between Japan and England, which are separated by both physical distance and time difference, atypical

* Automobile R&D Center
** FUJITSU LIMITED

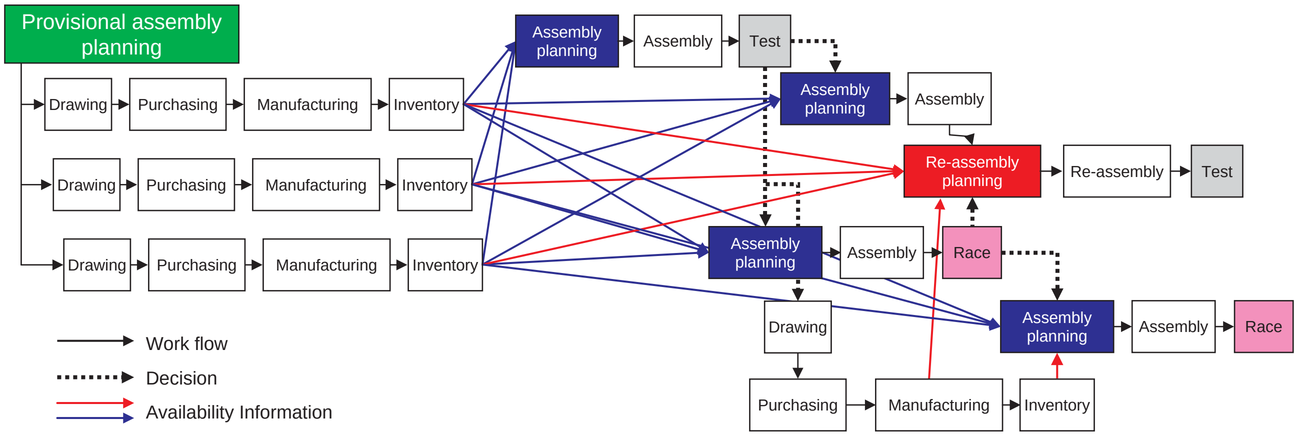


Fig. 1 Racing operations

Fig. 2 Production car development

and detailed exchanges via telephone or e-mail consumed much time and manpower.

As a result of the above, it was sometimes difficult to make optimum decisions promptly, finding replacements for lacking parts took time and effort, and excess inventory sometimes occurred, all of which were factors hindering the effective use of development resources.

2. Development Policy

The newly developed system aimed to share accurate parts management information to enable the timely and accurate decisions needed for racing operations. In addition, construction of an environment enabling the maximum utilization of development resources was also expected as a result.

To achieve this, the target was set as centralized management of constantly up-to-date plans and results for all stages, from the investigation of part design specifications to drawing issue, production plans, arrangement and purchasing, procurement and manufacturing, receiving inspection, inventory control, assembly, usage history, and disposal.

The specific development indexes were determined as follows.

(1) Faster verification of assembly plan feasibility
(2) Increased purchase order speed
(3) Enhanced parts management information

Construction of the new system required that the operations flow be restructured, so members selected from all related departments thoroughly verified the work flow, and reviewed operational roles to achieve the optimum overall balance.

Due to the nature of racing operations, speed and flexibility are demanded, so the division between contents that should be decided by humans and areas that can be left to the system was thoroughly discussed, and the specifications were determined.

In addition, measures were taken to promote accuracy and maintain up-to-date information, including systems that provide tangible effects, such as letting not only people make decisions based on centrally managed information, but also letting people in charge of entering information view the necessary information.

An advance project was commenced in Spring 2006, and work began to understand the current status of all operational issues involving racing parts management at Tochigi. Based on these results, operational roles were redefined starting from Spring 2007, and construction of the new system began from the start of 2008.

3. Function for Verifying Assembly Plan Feasibility

An integrated system was constructed that links and verifies the control number (unit number) information assigned to each assembled product (A), assembly schedule (B), assembly part configuration sheet (C), arrangement, purchasing and delivery information for each part (D), inventory information (E), assembly results (F), and usage history (G) (Fig. 3).

The system uses alert displays to automatically indicate which part impedes the assembly flow when the assembly plan for each unit number is changed, and all related persons can share information instantly. This lets people make comprehensive judgments regarding responses to various issues, and these decision results are reflected to the system, enabling the latest plan information to be shared.

The individual functions that comprise this system are described below.

3.1. Assembly Part Configuration Management for Each Completed Product

An assembly part configuration sheet function was developed as a core function for verifying plan feasibility. This function was created by extracting the plan management know-how accumulated by the Racing Division, and managed the parts configuration for each completed product. This function generates the original data that is the basis for instructions to all subsequent processes, and features the ability to change the configuration even after parts are purchased and manufacturing starts.

Assembly part configuration management for each unit number was achieved by linking data such as noted below (Fig. 4).

(1) Registration of parts by each specification

Multiple specifications are defined for each function unit, and the components used in each specification are registered. This enabled the selection and use of arbitrary parts (selected parts) from among multiple types.

(2) Registration of specifications applied to unit numbers

The combination of specifications to be applied to each function unit is registered for each unit number.

(3) Assembly part configuration sheet

The data from (1) and (2) above are linked to create a list of all parts required to assemble each unit number.

3.2. Delivery Management of Purchased Parts

Functions enabling the swift provision of delivery date information were strengthened based on split delivery and delivery date changes according to the supply capability of suppliers, which is common in racing operations.

(1) Multiple delivery date response management

The system that links suppliers and purchase order information was modified and a function was developed so that multiple delivery date responses can be registered

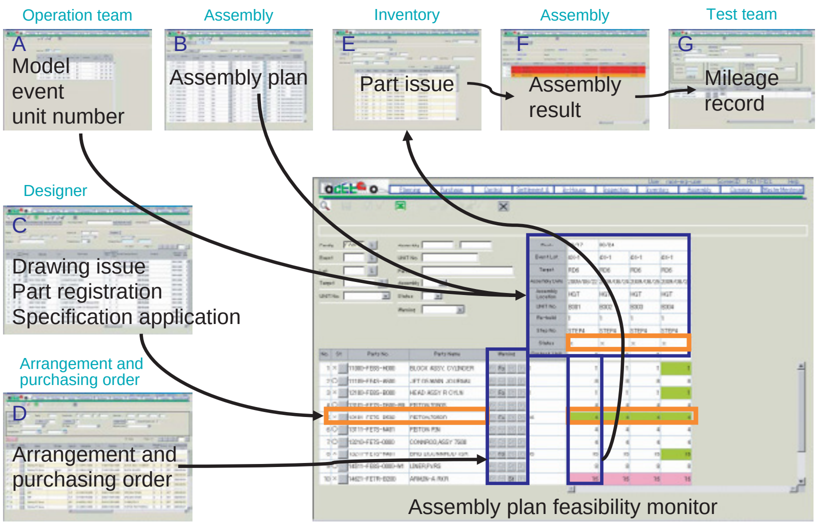


Fig. 3 Assembly plan feasibility monitoring

Part registration for specifications

Part	Quantity	Common	Piston L1	Piston N5	Funnel M	Funnel S
A	4		○			
B	3	○				
C	2			○		
D	2			○		
X	1	○				
E1	4	○				
E2	4				○	
F1	4					○
F2	4	○				

Specification application for unit No.

	Round1		Bench
Region	R001	R002	Performance
Quantity	1	1	5
Piston			
L1	○	○	10
N5		○	20
Funnel			
M		○	10
S	△		10
memo			

Assembly part configuration sheet

			Assembly date	2007 06/01	2007 06/02	2007 06/10	
			Unit No.	R001	R002	Performance	
		Part	Quantity	1	1	5	Quantity
Piston	L1	A	4	4		10	14
Piston	N5	B	2		2	20	22
		D			2		2
Common		C	3	2	2		4
Common		X	1	1	1	5	7
Common		E1	4	4	4	20	28
Funnel	M	E2	4		4	10	14
Funnel	S	F1	4	4		10	14
Common		F2	4	4	4	20	28

Fig. 4 Assembly part configuration sheet

for a single purchase order.

(2) Confirmation of delivery date response statuses

Various supplier delivery schedule statuses can be confirmed, such as delivery date responses not yet received, delivery date delay responses, or changes to delivery date responses. In addition, information concerning delivery dates (fixed delivery dates) agreed upon between the person making the purchasing order and the supplier can be shared.

3.3. Process Progress Management for Manufactured Products

When manufactured parts are completed by machining materials that pass between the in-house Manufacturing Division and multiple suppliers, unfinished parts may be transferred directly between suppliers to shorten the schedule. Process progress management formerly relied on the experience of the person in charge, and was performed by understanding the status of only representative priority management parts. Therefore, management was systematized and the following functions were developed to increase information accuracy and enable the sharing of information.

(1) Process progress management

Process design and machining process schedules and results can be displayed for all parts.

(2) Understanding progress using receipts

Unfinished parts do not necessarily move in purchasing order units. Therefore, progress status was understood by creating receipts and loading the receipt information to the system when unfinished parts are received by the next process.

3.4. Reserves Used to Calculate Arranged Quantities

To calculate the proper amount of parts required (arranged quantity) amidst frequently changing circumstances, it is necessary to compare the quantity required by the assembly plan with the quantity remaining to be ordered, current inventory, the quantity scheduled for assembly, and other factors. Therefore, a reserves function was developed to obtain the proper arranged quantity.

First, the free inventory is obtained using Eq. (1) (Fig. 5).

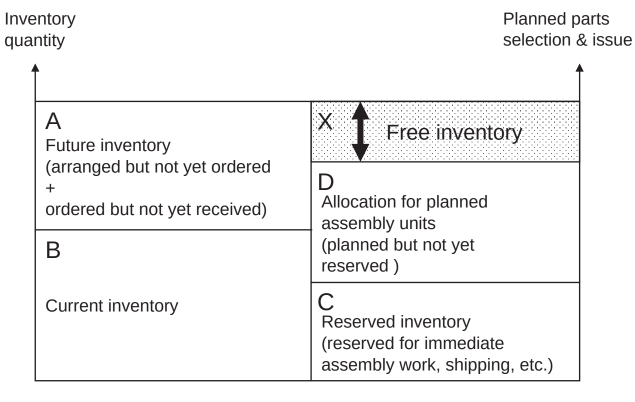


Fig. 5 Free inventory

$$X=(B+A)-C-D \tag{1}$$

X: Free inventory
B: Current inventory in parts warehouse
A: Future inventory (Sum of quantity arranged but not yet ordered and quantity ordered but not yet received)
C: Reserved inventory (Quantity ready and awaiting issue from inventory)
D: Allocation for planned unit numbers not yet reserved (Quantity required by unit number plans but not yet reserved for issue from inventory)

The final arranged quantity can be determined by experienced personnel by taking into account irregular factors such as the feasibility of reusing old parts based on the shared part management information described hereafter, and the number of race events remaining in the season, and then adjusting the quantity with respect to the free inventory obtained in the manner described above. This enabled more accurate arrangements.

3.5. Inventory Sorting

Labels noting the unit number information based on the assembly plan at the time of purchasing order placement were formerly affixed to parts, but this was not useful for racing part management in which assembly plans undergo frequent changes. Therefore, an inventory sorting function was developed to increase the sorting speed and accuracy.

This function automatically creates prioritized sorting proposals for available inventory based on the latest assembly parts configuration sheets, assembly dates, unit number purposes, applied specifications, and quality rank. Humans then make a comprehensive decision to determine the final sorting destination. This enabled highly flexible operation.

4. Purchasing Order Related Functions

Various pieces of information are required when ordering parts, and the links between this information affect the actual speed required to place purchasing orders. The speed required to place purchasing orders was increased by strengthening and enhancing the functions and processes described below.

4.1. Information Entry Prior to Drawing Issue

In the conventional system, determination of the arranged quantity, supplier, tentative price and other purchasing order operations were performed successively after drawing registration (hereafter, "drawing issue"). Centralizing information enabled more accurate advance information to be shared, which made it possible for these operations to proceed concurrently prior to drawing issue (Fig. 6).

(1) Registration of drawing issue schedule

Various pieces of information concerning parts scheduled for drawing issue are managed. The scheduled drawing issue date, material quality, base parts and other information that affects purchasing orders is entered.

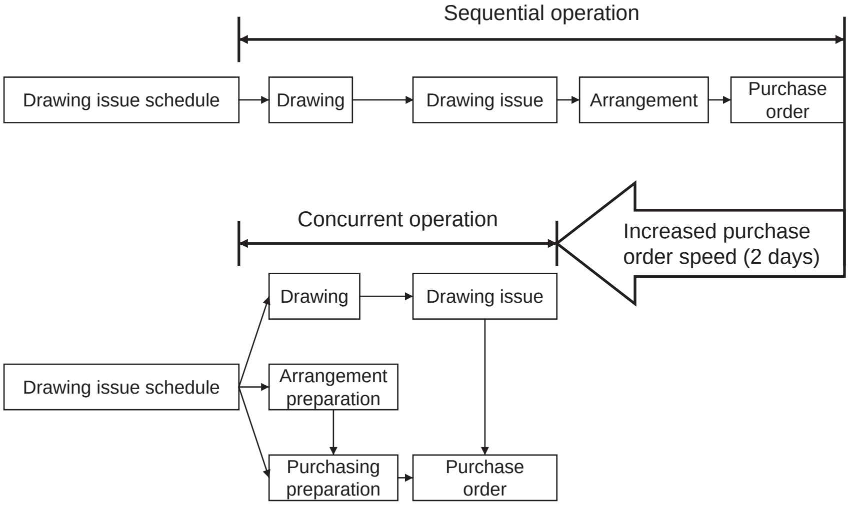


Fig. 6 Revised process of purchasing

Coordination issues are prevented by enabling part numbers to be entered only from the drawing issue schedule, which is the initial source of information, and then sharing that information with subsequent drawings creation and issue.

(2) Arrangement preparation

Information on the arranged quantity based on the assembly plan, and the department requiring the parts, is entered.

(3) Purchasing preparation

Suppliers and procurement methods for supplies are investigated based on the drawing issue schedule information, and the results are entered.

4.2. Enhancement of Drawing Issue Process

In the conventional drawing issue process, information sometimes stalled partway through the information transfer and other processes that relied on paper-based approval and distribution. In contrast, the newly developed system enabled instantaneous information transfer using the following functions.

(1) Compatibility check when issuing drawings

The conventional system had a downstream process whereby drawings were checked and revised by humans, but the check function was strengthened so that part configuration and part history compatibility checks can be completed in the drawing issue stage.

(2) Specification change notices

Specification change notices can be automatically created by extracting data from the drawing issue schedules and advance information.

(3) Issue of specification change notice numbers

Specification change notice numbers are issued immediately after a drawing issue is approved in order to strengthen compatibility check functions when drawings are issued.

5. Enhanced Parts Management Information

5.1. Inventory Control

Separate ledgers kept outside the system thus far and information managed by other systems without data links were integrated to realize the following functions.

(1) Individual part management

Individual part management was enabled by assigning a unique ID number to each part during the receiving inspection, and printing or laser-marking a 2D barcode on the label or directly on the part.

(2) Management by quality rank

This function enabled better inventory control by ranking parts based on actual measurement results, and grouping parts into ranks.

(3) Reserved and allocated inventory part list

This corresponds to C + D in Eq. (1), and is a list of existing parts not yet issued from the inventory, from among the parts required for assembly of each unit number according to the assembly part configuration sheet. When a design change occurs, it is automatically reflected in this reserved and allocated inventory part list.

(4) Usage history management

Managing the part usage history enabled a more detailed understanding of part replacement periods based on lifecycle.

(5) Management of inventory outside the part management warehouse

Total part quantities and locations can be clarified by centrally managing the storage location and inventory in each department, from the time parts are issued from the part management warehouse until the parts are discarded.

(6) Management by assembly and unit number

Parts can be managed not only by part number, but also by the arbitrary groupings of assembly and unit numbers, which are used during storage and transfer (Fig. 7).

5.2. Assembly Results Management

Part assembly results were formerly managed by a separate system that is not linked with the inventory control system, and entering the results took time. However, the newly developed system enabled accurate recording in a short time. Engines in particular are composed of approximately 1000 parts, so methods were devised to efficiently and accurately record the results for each unit.

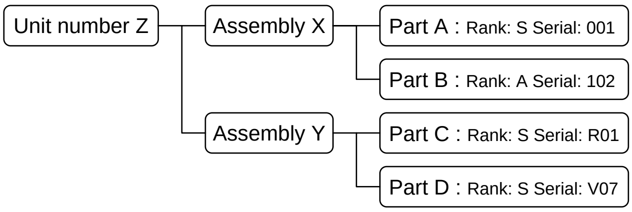


Fig. 7 Unit number and assembly

(1) Increased assembly results in entry efficiency

The new system enabled quick and accurate entry of assembly results in a semi-automatic manner instead of manually, by scanning a 2D barcode of the individual part ID number. In addition, automatic carryover of inventory issue information for each part prevented entry errors and at the same time reduced the data entry workload.

(2) Easier assembly results verification

Assembly part configuration sheets and assembly results can be compared, enabling instant understanding of differences and easy comparison of different assembly results.

(3) Link with manufacturing quality information

Serial number information is received from a separate system that manages manufacturing quality information, and linked with individual part ID numbers. This enabled detailed information on manufacturing quality to be referenced from the assembly results.

6. System Application Technologies

The new system is not a customization of a commercially available package, but was constructed through dedicated development in consideration of maintainability, expandability and feasibility of the requirements unique to racing operations such as described thus far.

When developing a system, the policies and goals of systemization must be commonly held, or individuals will devise arbitrary requirements, making unification of specifications difficult. Conversely, excessive standardization may affect ease of use. An open application framework was used as the base for the system, but feasibility of operations was given priority, and construction aimed for a balance between standardization and individuality.

The technical contents applied to each level are described below.

6.1. Approach towards User Interface

Basic window configuration patterns such as search and lists were provided for the user interface, but the layout of each window was investigated thoroughly together with users to realize demands for operability that differs for each operation. As a result, the following window configuration patterns were created to match individual requests.

(1) Monitor panel type

The assembly plan feasibility verification window shown in Fig. 8 displays the arrangement status for each part in unit number units, and enabled identification of up to eight different color-coded alerts for easy understanding of the status at a single glance. In addition, the layout provides launch buttons for each part to enable the necessary measures to be taken according to the alert type.

(2) Operation intensive type

The purchasing order information creation window is used constantly, so all necessary operations should be realized in a single window. Therefore, the layout was divided into three areas, with basic information displayed in the upper part, and the purchasing order form and

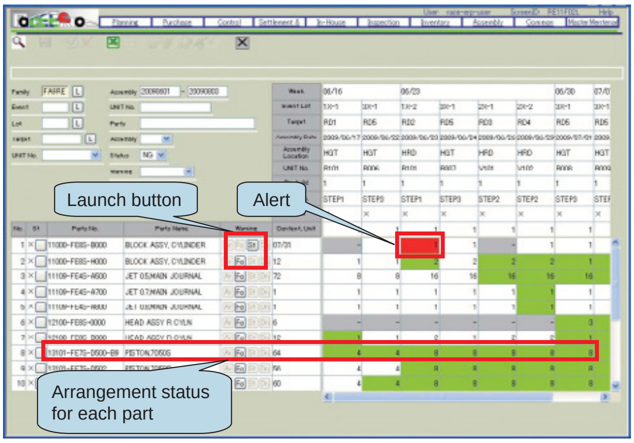


Fig. 8 Assembly plan feasibility monitor

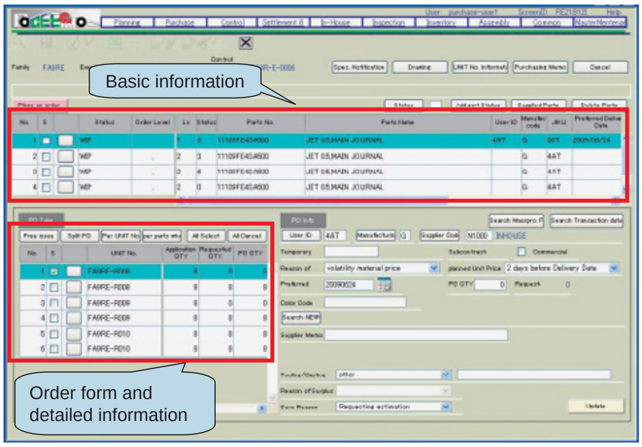


Fig. 9 Purchase order entry

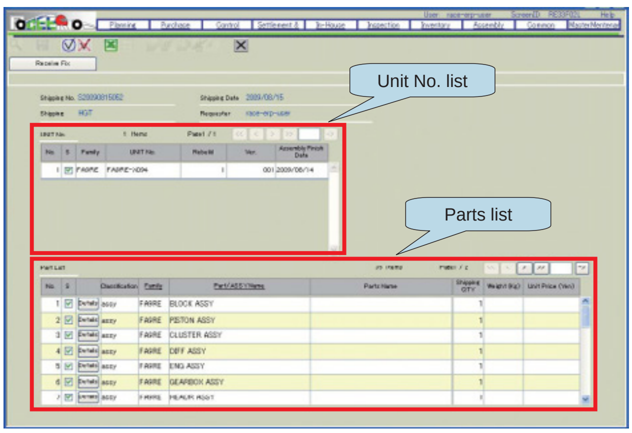


Fig. 10 Receiving confirmation

Table 1　Common functions

	Common functions	Explanation	Usage pattern
a. Display	• Table column customization • Structure expansion • Memo	Enable each user's preferred display, structure and additional information	Sorting or selecting data on user's demand
b. Multi location	• Multi language • Multi location	Switch displayed language for menus, drop-down lists, etc.	Usable by multiple offices, both domestic and overseas
c. External interface	• Excel input • Excel output	Bulk operation by using a spreadsheet is possible.	Mass data inputs and outputs essential for fast operations
d. Security	• Menu privilege • Data privilege	Can be set for each user based on their role and screen mode	Privileges control based on department and/or title

detailed information displayed in the lower part of the window (Fig. 9).

(3) Simultaneous list display type

Both the unit number list and the parts list must be checked at the same time, so two areas with separate page switching control were implemented in the upper and lower parts of the receiving confirmation window shown in Fig. 10.

6.2. Common Function Technologies

Common functions were split from the operations logic and implemented on the framework to increase user convenience and system development efficiency. Frequently-used functions were made into common functions that could be accessed from any window, and this enabled system developers to focus on the operations' logic design, which increased both quality and development efficiency. Table 1 summarizes the common function technologies applied from the viewpoints of (a) display, (b) multi location, (c) external interface, and (d) security.

6.3. Approach towards Data Model

The data model did not use the data structure of the conventional system, and was instead newly designed to realize centralized management of information. Design departments create design drawings and part configurations in part number units, but the basic information unit for ordering departments is the purchasing order form. In addition, after parts are received, important parts must be identified and managed individually. Therefore, a database was constructed that links the assembly part configurations, purchasing order forms, and individual parts handled by each department

Application specification

	Unit001	Unit002	Unit003
	May/20/2009	Jun/10/2009	Jun/20/2009
Part A	2	1	
Part B		1	2
Part C	2	2	2

Arrangement

Part	Arranged quantity	Arranged date
Part A	3	May/10/2009
Part B	3	May/30/2009
Part C	6	Jun/10/2009

Test

	Serial No.	Mileage
Part A	E0001	50
Part A	E0002	50
Part A	E0003	0

Purchase order

Order No.	Part	Ordered quantity	Requested delivery date	Agreed delivery date
0010	Part A	3	May/10/2009	May/10/2009
0020	Part B	3	May/30/2009	Jun/1/2009
0030	Part C	6	Jun/10/2009	Jun/10/2009

Unit No.	Part	Serial No.	Quantity
Unit001	Part A	E0001	1
Unit001	Part A	E0002	1
Unit001	Part C		2
Unit002	Part A	E0003	1
Unit002	Part B	E0010	1
Unit002	Part C		2
Unit003	Part B	E0011	1
Unit003	Part B	E0012	1
Unit003	Part C		2

Received and inventory

Part	Serial No.	Quantity	Received date	Order No.
Part A	E0001	1	May/10/2009	0010
Part A	E0002	1	May/10/2009	0010
Part A	E0003	1	May/10/2009	0010
Part B	E0010	1	Jun/1/2009	0020
Part B	E0011	1	Jun/1/2009	0020
Part B	E0012	1	Jun/1/2009	0020
Part C		6	Jun/10/2009	0030

Fig. 11　Data model

in terms of arranged quantity, received quantity, and assembly results (Fig. 11). This enabled each department to handle information in the units suited to the operations performed by that department, and organically linking these units of information enabled information spanning the entire part lifecycle to be instantly understood.

7. Confirmation of the Effects of System Application

The decision to withdraw from Formula One was made immediately prior to application of this system, and the system has not been used for actual operations. However, the system was still completed and integrated tests were conducted. The results confirmed that each department could carry out operations under an environment in which all data related to part manufacturing and supply was centrally managed, and that timely and accurate decisions could be made, which was the goal of development.

7.1. Faster Verification of Assembly Plan Feasibility

Formerly, the feasibility of the overall assembly plan first became apparent when persons in each department compared the information held separately by each department. This made a process to investigate countermeasures essential, and checking the location of a quantity of specific parts for this sometimes took as much as two days. However, this system enabled instant identification and information sharing for the part which impedes the assembly flow. In addition, the latest information is centralized and can be easily referenced, which is thought to enable swift decisions even when coordinating between Japan and England.

7.2. Increased Purchase Order Speed

Accurate advance information enabled advance preparations to proceed concurrently, which together with enhancements to the drawing issue process, shortened the time required from drawing issue to place a purchase order by two days. In addition, centralization of information can also be expected to prevent confusion and duplicate work due to differences in information.

7.3. Enhanced Parts Management Information

Inventory control from the time parts are entered into the inventory until disposal can be performed not only in part number units, but also by individual parts themselves, assembly, unit number, or quality rank. This enabled an accurate understanding of inventory quantities as well as measures for part excesses and shortages according to the application. In addition, understanding part replacement periods based on individual part usage history information can also be expected to help understand the part quantities required and achieve proper inventory levels.

Assembly results records up to and including actual work such as at a racing circuit can also be shared in real time, simplifying the process of checking which of the selected parts have actually been used. In addition, links with a separate system that manages manufacturing quality information enabled swift and accurate decisions to be made regarding defective parts and whether to exclude parts from the same manufacturing lot.

8. Conclusion

As of 2006, HRF1 was already collectively managing racing sites and factories for non-engine parts, using the part usage history management function and 2D barcodes with an originally developed system based on commercially available software known as the ERP (Enterprise Resource Planning) Package. The starting point for this system development project was the finding that HRF1 was able to respond immediately to inquiries from Tochigi regarding gear box parts, exhaust pipes, and other parts.

Centralizing part lifecycle management required wide-ranging investigations and reviews to optimize overall operations. Approximately 3 years were needed from the start of advance investigations in 2006, but the know-how generously provided by HRF1, thorough discussions by project members involved, and the clear sharing of goals and targets across the Racing Division enabled completion of the system.

This system was designed for direct application to both Tochigi and HRD, and aimed for centralized management of engine parts. This system was not applied directly to HRF1, but preparations were underway to enable links with chassis part and gear box part information, such as a part number read-as function and measures to avoid duplication of key management numbers, to enable future links with existing HRF1 systems.

The indexes of “faster plan feasibility verification, increased purchase order speed, and enhanced part management information” are themes that are also constantly pursued in the development of mass production cars. Application of this system to the development of production cars will be investigated in the future.

■ Author ■

Hidetada TANIGUCHI

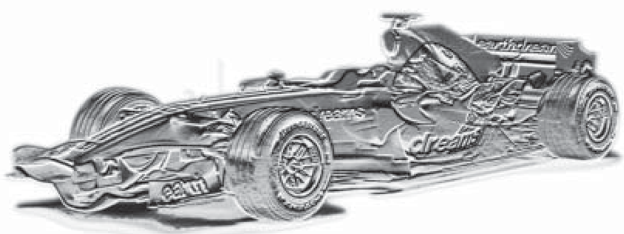
Katsunori TAGUCHI

Masataka NODA

Shigeo MIYAJIMA

Osamu WATANABE

Activity Reports



Looking Back on Assembly Activities –Racing Spirit–

Tsuyoshi ISHIHARA* Naoto SUNAKO*
Atsushi NORIHARA* Motoaki ISHIZAKA*

ABSTRACT

Numerous technologies were developed and put to use in races during Honda's third-era Formula One activities. This paper introduces assembly activities from development tests to the racetrack.

1. Introduction

Honda's third-era Formula One assembly activities began with several goals: "no dropping out of a race because of human error;" "no supply delays;" "reflect development issues in blueprints;" and "pass down technological skills." Based on Honda's second-era experience, these four elements were essential for race management, and activities were always pursued with a consciousness of these targets as the path to victory.

This article introduces these activities by dividing them into those that are in the factory domain, taking place at Tochigi R&D Center (HGT), which oversaw development of new technical items and the assembly and supply of race and test vehicles, and at Honda Racing Development (HRD); and those in the circuit domain, which supported race and running test sites.

2. Factory domain

2.1. Startup of Honda Formula One Third Era

The third era started in 1999 by organizing the team strength so that primarily those members who had supported the golden era of Honda Formula One in the second era could pass down their technical skills to new members, increase development speed and pursue supply of equipment for races. Passing on everything to the new members involved a lot more than just writing a manual and making them read it; it was a lot of work!

This section first gives a simple explanation of how technological skills relating to the work of the assembly crew were handed down.

At the development stage, one repeatedly assembles and dismantles previously developed single-cylinder engines, demonstration bench engines, durability engines and running test engines, after which one checks the new specifications, looks for issues and feeds back results of countermeasures to try to bring blueprints to fruition. One assembles an engine built to the latest specifications and then sends it to the racetrack. Engines used in races are dismantled, malfunctions are found and then this goes into the next cycle of activity. In this way, the assembly domain is always involved, from the beginning of development until race day.

Although one casually talks about engine assembly, a Formula One engine is not simply put together using as-is components that are in inventory. These are very delicate engines: with the emphasis on performance, they have only the minimal amount of durability and are subject to approximately five times the engine vibration as mass-produced engines, and the slightest error or even one component quality defect can easily cause them to break down.

To assemble such an engine, there must be no errors. Components must be checked assiduously one-by-one so that not even a fine burr is overlooked, and so that each component can perform to its maximum extent. The assembly crew is part of the final process, where one can touch actual articles and check them with one's own eyes, and quality must be protected here so that the team can win the race. Similarly, during dismantling, one visually senses malfunctions and feeds back information to bring blueprints to fruition.

Assembly is furthermore required to supply equipment for development and races with enough speed to keep up. In fact, the authors cannot remember a single season in Honda's Formula One activities in which there was not a need for emergency component replacement because of malfunction, or no repairs on engines experiencing trouble. It is always a battle against time, so one has to work with efficiency, speed and reliability.

To achieve this, the biggest issue is giving new

* Automobile R&D Center

members the ability to spot malfunctions as well as speed and work precision.

It is hard for new members to learn these things in a short period of time, and there was a noticeably large experience gap between them and Honda's second-era Formula One members. The new members started with their experience at HGT, got more experience from being sent to HRD, and finally grew into track mechanics.

These members prepared on average 1000 engines per year, including new assembly, rebuilds and reassembly (Fig. 1), with the number of reported troubles averaging 182 per year, thus contributing greatly to bringing blueprints to fruition.

2.2. Development of Quality Enhancement Measures

During nine years of activities, the regulations have changed several times to cut costs: the usage restrictions on engines have changed so that the guaranteed usage distance is 1 500 km, up from 300 km; and whereas there were non-restrictive rules allowing practice, qualifying and race engines to be changed, this was changed to require one engine per race-event, and then one engine per two race-events. The regulations have also changed in ways that affect race strategy, with a penalty now being assessed for changing engines. This has increased the requirements for engine quality and durability.

Originally, our members thought that the old ways of doing things were sufficient, but extending guaranteed engine usage distances was a new frontier for our Formula One activities up to that point, and also represented a great opportunity to further enhance quality and durability.

First, members strengthened quality checks of each component. They anticipated the possibility of impurities getting into places invisible from the outside, such as oil passages, and then before assembly used an endoscope to do a full check. For phenomena such as injuries on the sleeve, it was decided to do a bench check prior to shipping, then check piston and sleeve status and look for impurities with an endoscope immediately before shipping, and then supply equipment to the racetrack. Besides this, the number of check items was increased

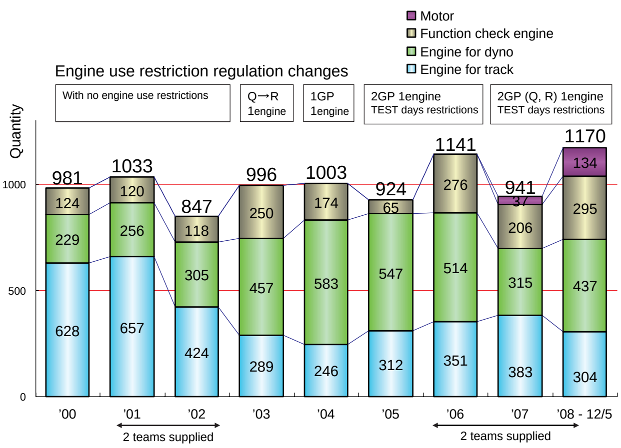


Fig. 1 Record of complete engines and motors supplied

for all assembly processes and countermeasures were added to prevent troubles before they could occur, which strengthened the members' efforts to eliminate defects that were their fault and prevent faulty components from shipping. This further enhanced members' awareness of quality and made them more mindful of the engines they themselves assembled.

During assembly, more malfunctions occurred even as performance was being enhanced. Development had proceeded on material substitutes to reduce friction of the air-valve system movable seal, but air leaks during assembly increased. Every time an air leak occurs, it is a major issue in terms of supplying equipment for races on a tight schedule: it takes time to repair, and the crew has to wait for specifications adopted in response until it is almost too late for the race. There were also concerns that an engine might blow out during a race.

The team originally suspected a quality flaw in components on the periphery of the air valve, but could not come to a definite conclusion, so they performed analysis starting at the point where the air leak occurred and found that a minute amount of dust had gotten caught on the movable seal face. Under specifications up to that point, there was no impact even if dust got caught, so material substitution was believed to be the cause. Components were cleaned and then checked prior to assembly, and assembly was done with great care, but it did not solve the issue and small amounts of dust and dirt still got caught.

As a countermeasure, impurities that came from within components and which could not be removed by ordinary cleaning were eliminated by implementing an ultrasonic cleaning system. To eliminate airborne dust and dust from work clothes, a clean room was set up and workers wore anti-dust clothing over their work clothes and anti-dust caps over their hair as they assembled air-valve systems. Movable seal faces, which can be damaged by even a light abrasion, were put on with great care, and the engine assembly process was nerve-wracking.

The result was complete prevention of air leaks and the ability to supply more competitive engines.

2.3. Supply of Motors Made by Honda

In 2007, the FIA presented a regulation permitting the use of hybrid systems in races starting in 2009, after which it was decided that the motor would be built by the assembly crew.

This was an area in which the assembly crew had no experience, meaning they had to start from the very beginning. Up to that point, the assembly crew had overcome many obstacles and felt that it would surely find a way, but there was much unease with the transition from things one could tangibly observe to something electric, which could not be seen at all.

The crew's education began with the study of motors for mass-produced vehicles. Getting advice from experts on mass-produced motors, the crew began to master this area.

Actually, even the stage of checking the building process was of greater difficulty than imagined. Structurally, the job of coiling wire is something that only one worker can do on one motor. It is difficult to divide up tasks as one can with an engine and thereby shorten the production timeline, and when one is planning to supply equipment for race use, one cannot wait until just before the race to decide on the best specifications. Additionally, if some trouble occurs, the equipment would not be on time, and this increased the sense of unease.

Being lightweight and compact while providing high output are absolute conditions for a motor used in racing, so winding technology was required offering a super-high space factor. The issue of greatest priority was to establish a method of winding a coil: specifically, how to wind it so that it would be within the required size.

An examination and winding experiment were begun, but Tries 1-3 could not meet the requirements at all (Fig. 2). The coil end was always too high. At this stage, about 50 hours had been spent on the work. On Try 4, a jig was devised and experimented with, but as it still did not meet requirements, some other solution was necessary. Other attempts were made, such as enhancing the jig or changing the winding process, and on Try 5 a breakthrough seemed to be in sight, but while progress had been made, this still did not meet requirements. With more innovations, the crew finally succeeded in meeting requirements with Try 6. It had taken two months from the start of examination until trial winding.

For the original winding, the crew thought to reduce work time by implementing equipment and winding mechanically, but eventually concluded that meeting the requirements would necessitate a very complex winding method, and as it was just not possible with a machine, it had to be done manually.

Once the winding work was over, the coil ends were molded to fit the coil in the case. At this point, a very big problem came up. During molding, the wires rubbed against each other and shorted out. After all the long hours it took to wind a coil, in the end it was not usable.

The crew understood that the cause was the relationship between the winding method and molding load, but since the work was done manually, it was impossible to make the way the wires come out match perfectly, so a technique was selected that controlled load during molding.

Fig. 2　Try 1

While the crew somehow managed to succeed, it took many staff hours and there were still concerns about being able to provide the motor on time. After examining many ideas for reducing work time, about 30 types of jigs were created.

Eventually, this effort succeeded in reducing winding work time from about 50 hours to 15 hours, in part because of overall skill enhancement of the workers.

As a result, the crew's goals were in sight for supplying equipment for durability testing, road testing, and all races, and supplying system became highly organized.

Production also got up and running, and the know-how acquired during production for the 2009 winter running tests was a very valuable asset. However, Honda announced it was withdrawing from Formula One and, unfortunately, this motor was never used in a race.

Motor development that had begun with a lot of fumbling and no clear idea of how it would turn out ended up being a source of great confidence and pride for the assembly crew.

2.4. HRD, where Honda Engines were Born

To offset the risk of distance from the engine development base HGT, HRD, the base for onsite engine maintenance (the rebuilding team) and the running test team during Honda's third-era Formula One activities, was established to inherit functions from Honda's second-era Formula One activities and serve as the front line of support for each circuit in the EU. When first launched, the site supplied engines during Japan's long holiday periods (May and August), and subsequently its role expanded, such as making an initial determination of durability and reliability following running tests of new-specification engines, making emergency replacements of rejected lot components, and having materials analysis members onsite. This allowed them to make more detailed analyses and assessments of the situation when malfunctions occurred at the running test site and thus greatly accelerate development. Additionally, when supplying engines to two teams the second time, the scope of the crew's responsibility grew to include all engines for the Super Aguri Formula One Team (SAF) as well as engines for the Honda Racing Formula One Team (HRF) (50% of running engines supplied). The increasing number of engines supplied, along with the need to deal with malfunctions, made it necessary to strengthen and expand the assembly system. Along with this, activities were also transformed by having one person in residence and persons dispatched from Japan taking up long-term residence so that it became their regular work, an arrangement that supported race activities. Engines prepared by the rebuild team were handed over to the site mechanics with the sense of saying, "Please take care of this engine," which, though the words were not actually spoken, bound both sides in a relationship of trust. Every member of the rebuild team had a sense of pride and self-confidence that the engines they were shipping out were perfect. The engine rebuild specialists dispatched from Japan lived three or four in a home close to HRD during their five- or six-month stay in the UK. In addition, this provided a venue for new members to learn the mindset and spirit of accepting challenges in their day-to-day lives as well as in races, because these people were sharing a home with the veterans several times a year, and it provided the sole place where they could relax while in the UK. It was HRD that prepared the Honda engine that enabled Jenson Button to win the Hungarian Grand Prix in 2006.

2.5. Collaboration with Local Members

There were changes to the system in the rebuild field from Honda's second-era to third-era Formula One activities. In Honda's second-era Formula One activities, only persons dispatched from the R&D Center conducted the work, and they were able to work the same way where they were dispatched as in the R&D Center, almost as if the site were an extension of the R&D Center. In the third era, a number of local staff members worked side-by-side with members dispatched from HGT from the time this arrangement was established. The culture and language were different, of course, and the authors remember how difficult it seemed to stay in step with each other. But while fusing our different cultures and launching a system to build equipment together were big challenges for us in conducting race activities in the UK, there was also a great sense of eagerness to do this as a new experiment in Formula One activities.

The local staff members who were hired were typical race engine builders who had only assembled parts out of inventory, and so had no experience with engine development to deal with the development issues from the quality side. By contrast, Japanese crew members acted with a constant sense of responsibility as the final check-persons in every domain (inspection, processing, materials). They also were aware at all times that this place was an R&D center, not an assembly plant, and as such it was up to them to guarantee the quality of the finished engines and it would be the assembly crew's fault when defective components went into their assemblies. When this happened, there were always discussions about why the defective component was not noticed beforehand.

The veterans said, "Get development information from the R&D Center and local information about the circuit, but in any case look closely at the actual engines and components in front of you." There was never any wavering from this principle, which was a tradition handed down from Honda's second-era Formula One activities. Local staff members were partnered with persons from Japan who instructed and guided them to help them develop this way of doing things, so that after several seasons had passed, the work could finally proceed smoothly. Local members were given roles with responsibility and thereby learned a sense of responsibility, so that the two sides came to understand each other better and both could absorb new knowledge from the other.

In 2006, Honda began supplying engines to two teams for the second time, and it was decided that all SAF engines would come from HRD. Until then, HRD had been in charge of preparing Friday practice engines (P) and race spare engines (RS), but now it would also be in charge of race engines (R), and this would greatly transform the arrangement in which HGT assembled and supplied the R.

Starting that year, all engines from HRD had "Made in UK" stamped on them. This boosted the motivation of local members and was taken as a sure sign of their independence. The authors recall how this established a solid relationship of trust between the two sides in Honda's Formula One endeavors.

2.6. The Unknown Story of How the Winning Engine was Prepared

In the latter half of July 2006, to make up work during the summer holidays at HGT, assembly was planned and carried out at HRD on two engines for the Hungarian Grand Prix of August 6-8. Preparation of one engine proceeded smoothly and was completed on schedule, whereas the other was plagued by troubles. During assembly, air started leaking from the valve-train system air-valve seal face and would not stop. Studies were done on the situations deemed likely to cause this (damage to the seal face, the interposition of impurities, deformity and form defects, hardness, etc.), but nothing could be identified. Repairs were made with a great deal of work time spent, for example switching out components from different lots, and the two engines were finally done with little time to spare. A plan was organized to assemble and supply Rd13 engines, a total of three engines including the one assembled at HGT. The HGT engine, with slightly greater output, was mounted on Button's machine, while one of the engines assembled by HRD went on Rubens Gonçalves Barrichello's machine. As for the remaining engine assembled by HRD (the one that had an air leak during the assembly work), it was to be used as the race spare to the disappointment of the local crew, but then the Hungarian Grand Prix opened and Button's engine blew during practice on the second day. The spare engine from HRD was quickly mounted on his machine. Originally slated to start fourth in the qualifying, Button was given a 10-slot penalty for change of engine and had to start in 14th place and would have to work his way up from there. While this would have been fatal under dry conditions at the Hungarian, which makes overtaking difficult, the race day began under wet conditions. Vehicle speed, weather changes, race strategy and veterans' dropping out were among factors that let Button, starting in 14th place, gradually raise his position and ultimately finish the race far ahead of the other vehicles.

At that moment, Honda had won its first third-era Formula One victory. For HRD, winning the race with an engine they had built with one British member and one Japanese member was a great confidence-builder.

3. Mechanical Domain

3.1. Supplying Equipment to Two Teams

From the start of Honda's third-era Formula One activities until Honda began supplying more engines to two teams in 2001, the number of staff members increased and, as a result, Honda started supplying two teams with a staff consisting of 13% experienced persons and the rest not experienced. For the mechanics, of course, inexperienced personnel had to be used, but under the guidance of experienced staff from Honda's second-era Formula One activities, these people received practical training in on-site trouble-analysis techniques.

Experience taught us that the ability to evaluate sudden troubles cannot be learned in a very short time, so all sorts of means were considered and put into effect to help the inexperienced staff make up the gap between themselves and the experienced staff. An attempt was made to enhance their knowledge by, for example, reading up on past troubles, conducting simulations of trouble with an actual vehicle in front of them, and having groups of young mechanics practicing FMEA amongst themselves. Sometimes, however, they became overwhelmed by the atmosphere of the site and the sense of urgency and were unable to act.

We felt that the ability to judge and act at a moment's notice in extreme circumstances could not be acquired unless we always had an engine in front of us and could put our hands on it.

Even as their knowledge and competency increased, it did not help at the racetrack unless they had tools, spare parts (including, for example, proven replacement components, in cases where a malfunction occurred with a newly introduced component) and so on.

The veterans often said, "Preparation occupies 90% of work."

The most essential thing needed was the practice of keeping the tools, equipment and components needed at the site in a neat and orderly way so that they could quickly be found when in a hurry and nothing would be missing. This is all very basic. But for the young mechanics, this was where the work began.

The following relates what happened at one event. An engine that had been running smoothly suddenly blew out without warning. The blown engine hurriedly underwent emergency dismantling by two of us, a veteran and myself (Ishihara); when we did primary analysis of the cause, the veteran asked me to pass him a battery-powered impact wrench, but when I picked it up, the battery was dead.

This led to a severe scolding from the veteran. I realized that my negligence in preparation had come to light in that instant. After that incident, the first thing we did at the site was to keep a battery charger ready.

The engines supplied to the race track are supposed to have undergone a very strenuous final check at HGT or HRD, but all engines underwent another engine-specification check at the hands of the mechanics, with each bolt checked for torque, harness wiring checked, etc., time and time again.

Oil-pressure adjustments and air-intake trumpet-length adjustments were made in keeping with the usage environment at the site.

From time to time, components were replaced to deal with troubles in time for an event and the crews constantly tried to extract the maximum performance, durability and reliability in each race.

On the starting grid of the final round, a number of final quality checks on the hardware side of the engine were performed right up to the start of the formation lap, including checks of whether oil, water, fuel and air pressure, air and oil consumption and oil and water temperature were acceptable, whether there were any leaks, whether all cylinders were firing and whether the air/fuel ratio was correct.

This is very hard work, as malfunctions have to be responded to immediately when found.

Sometimes the members were so nervous that their stomachs hurt and they could not eat until the signal changed and the vehicle got off to a trouble-free start.

More than actual race results, the mission of the assembly crew was to make sure the vehicle finished without trouble: for example, during the race, members read data from the telemetry system and prepared for an emergency pit stop if any abnormalities occurred.

3.2. Kitching

On the job at the racetrack, it was necessary to make sure that all mechanics were correctly producing the same output. This is referred to as "kintaroame," which is a kind of candy formed with an image in it. The association implies that it was vital, no matter who cuts it open and where they cut it, that the result will be the same.

The two race engines provided for the two drivers were lined up and arranged in detail right down to the thickness, position, direction and binding force of the wire harness and the tie-wrap that anchors the auxiliary components (Fig. 3).

Deciding on assembly work procedures and specifications when there are no blueprint instructions is, in large part, dependent on the experience of the mechanics.

This is because the heat, vibration and stress on

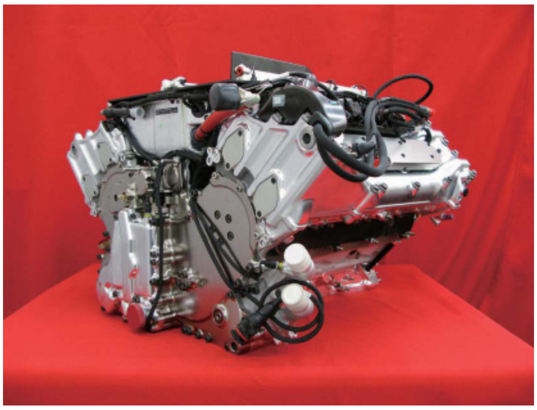


Fig. 3 Completed engine

components are so much more severe than is the case with mass-produced vehicles. Particularly during the summer races when it was so hot, the need to take countermeasures arose often, and when malfunctions occurred, there was a series of revisions of work procedures and specifications in keeping with changes in the environment so that crews on the site could determine the causes of the malfunctions and make the correct judgment on how to repair them.

Most particularly, with work procedures and specifications for engines built to new specifications, the crew, racking their brains, reflected procedures taken against past troubles, then verified these procedures in shake-downs and running tests.

Work procedures and specifications that were found to be reliable were then compiled in the form of a manual, including photos taken with a digital camera to illustrate things that are difficult to explain with text alone.

Surely, this was a case of “seeing is believing.”

The crew furthermore created a checklist so that nothing would escape the attention of all the personnel. As the years went by, the check items became subdivided, so that by 2008 there were 120 items.

This final check process before shipping equipment out is referred to as “kitching,” a word that is completely unique to the Honda Formula One world. This word goes all the way back to the era when the company participated in Formula Two racing in the 1980s. At the time, some veterans did not have enough time before a race to send an engine back to the R&D Center, so they did an emergency rebuild in the kitchen of the local company housing unit in England before sending it on to the race track, and the process was subsequently named after the rebuild site.

To be a mechanic, first of all one had to be able to do kitching correctly, and this became the gateway to success in becoming a mechanic.

During the third-era Honda Formula One activities, new mechanics were strongly advised by the veterans: “As a mechanic, you are carrying a big Honda sign on your back. As a representative of the assembly process, act in ways that will not make us ashamed.” Those words were bracing whenever I heard them. I felt as if I were an Olympic athlete wearing the rising sun of Japan on my back.

“Go for it! You can do it! We’ll be rooting for you on TV!”

As the vehicle set out from HRD to the circuit, all members came out to send it off. This was always a happy moment.

Formula One is very much a part of European culture and an event of its noble society. We took on this technical battle as representatives of Honda from Japan all the way in the Far East, with a different language and culture, and as representatives of Japan itself.

3.3. Onsite Engine Dismantling

During the engine startup check on preparation day during the 2005 Canadian Grand Prix, a lot of air erupted from the oil tank freezer. Both the P1 pressure (gas cylinder pressure) and the P2 pressure (engine supply pressure after regulator pressure adjustment) started falling and would not stop. When the P2 sub-line was removed to identify the location of the leak, the leaking stopped.

The leak was from the engine itself.

With no time to think, the place erupted into an uproar, with some people rushing to change out the engine to be in time for the vehicle inspection, and others acting decisively to hurry and open the cam cover to analyze the cause.

What the engine showed when it was dismantled was that a valve cotter had become embedded in a pneumatic valve return system (PVRS) piston and the seal ring was protruding very slightly beyond the chamber, causing the air leak. Results of analysis confirmed a malfunction in the taper angle of the cotter contact face of the PVRS piston, creating a looser taper angle than usual so that the cotter went deeper.

In this state, it was impossible to determine whether the trouble was caused by a quality defect.

As such, all the crew could do was check the interior of all five remaining engines. This would take as much as five hours per engine, and everyone was involved, including those who had come to hand over their work to substitute members.

Ordinarily, if one were to open the cover and remove everything down to the cam, the conventional wisdom would be to run the engine on a bench and check to see if there are any oil leaks or other abnormalities. But this kind of work cannot be done at the racetrack, and neither can the crew afford any errors. Nonetheless, they finished in time for the qualifying round, during which the team earned its first pole position in the season, giving the crew a taste of having accomplished something great. With the extreme tension lifted from their shoulders, the team mechanics shook our hands and congratulated us with a “Good job! Good job!”

Naturally, it would be better if there were no problems, but this experience in particular helped develop fearless race mechanics and was also reflected in HGT’s engine development blueprints, which enhanced durability and reliability. Responding to phenomena as they occur at the actual site, checking actual articles and getting to know the actual facts are essential to preventing the same trouble from happening again; this is the practice we call the “three-reality principles” in Honda. The crew relayed to Japan what they were thinking at the actual site and what was necessary, and passed that information on to Japan.

We understood the significance of mechanics because, as in this case, “if there is no word from the actual site, nothing will change.”

3.4. Adding an Air Pump

An air-pump system to supply air valves was added in 2003 as a way of reducing friction in the valve-train system, and this system was a source of many troubles for site mechanics.

The team requested that the volume of the air cylinder mounted on the vehicle be reduced. A mounted air cylinder would certainly seem to have an impact on aerodynamics, and making it any larger would be challenging.

An air pump was the way to solve the problem in order to continue supplying a limitless amount of air, which is consumed in such great volume.

It was a wonderful idea, as it would also free the mechanics from having to fill the cylinder with air onsite. However, at the start of development much more work was awaiting the crew than they had experienced up to that point. There was no way it was going to become easier.

The crew mounted a small compression pump on the engine and used engine oil to lubricate it. It looked just as if a radio-controlled car's engine were attached.

At first during development, it was one malfunction after another, and there was not a single day when it ran properly and without any trouble.

"The air pressure seems off. How's the regulator doing? Is the pump working? How about the hose?" The work consisted of a lot of urgent evaluations, because if the air supply were cut off, the engine would soon blow.

Simultaneously with air being supplied from the pump, the engine oil used as lubricant forms oil sludge, blocking the air channel. The filter intended to remove this became clogged from the great volume of oil sludge, so the air channel in the engine front cover also got blocked.

For the front cover, the crew attached a brush to an electric drill to clean out the air channel, but this failed to remove the blockage, so carburetor cleaner was blown in from the air-pump intake for 10 seconds and left to sit for five minutes before one minute of firing, a process that was then repeated a second time. This was done after both morning and afternoon runs. Components that had to be dismantled, moreover, were cleaned inside the trailer. For secrecy, this could not be done in the garage.

The process of cleaning out the acrid, black, slimy sludge lasted late into the night.

During initial development, filters had to be changed and channels cleaned out, and there was no way to avoid stopping the vehicle so much that it impacted not only the race distance but even vehicle testing.

We still cannot forget the team leaders gathering around and peering in, then giving us a cold British stare.

It reminded us that as mechanics, we are the front line and must bear the brunt of any troubles. As the engineers worked in the trailer each night under pressure to organize running data, they were close to that strong sludge odor, but nobody said, "Stop it, it smells terrible!" Everybody was aware of the situation and they put up with it together.

Naturally, once we all got the idea that "we've got to do something about this," it had a positive effect on later enhancements. Measures taken included reducing the temperature of the air blown in to a point where oil sludge was less likely to form, and also managing the clearance between pistons and cylinders as a measure against oil outflow.

3.5. Adding a Quick-Shift Gearbox

In 2003, the first thing the development supervisor said was, "Let's build the world's greatest gearbox," which was the start of a project to reduce lap times by developing an original new mechanism called a quick shift, which theoretically had no time lag during shifting. The technology of the system was later transferred to BAR, initiating the project to put it to use in races.

The development team and we were braced to do this, but at the time the basis for achieving such technology had not been established. Those of us who had specialized in developing and supplying Formula One engines were also dealing with gearbox development for the first time, and with no experience in these matters, we were starting from zero. It was also a turning point, after which the scope of our responsibilities expanded.

The British American Racing (BAR) team leadership expressed disapproval at the start of development, as they questioned what role Honda would be able to play in joint development of a gearbox and suggested they had nothing to learn from us. There was a reason for this attitude: in the year before the command applied, we received a chilly response when a malfunction developed on the driving site in a narrow-shift gear mechanism that had been suggested by HGT. Taking this experience into account, we responded this time with greater passion and speed.

A quick-shift system allows double engagement of gears: the gear being driven and the next gear to which the driver is changing. Because of this, during shifting there is no torque drop. The following year the crew proceeded to actual driving evaluation, when BAR, which got to know this mechanism on actual vehicles, had completely changed their attitude. Many engineers came to look at how the shift mechanism moved in the first equipment.

All of them said, "That's crazy," and left. At the initial running test, after a pit stop, we saw engineers and mechanics coming by, even climbing up the course-side fence behind the pit, to hear the sound of gears changing.

In reality, however, a series of severe development issues occurred. The crew tried all the countermeasures they could onsite on both the control soft side and hard side (making manual modifications to change the form), but each time the same phenomenon occurred at the same travel distance (35 km).

Although the system had passed the bench tests, there was now a baffling series of phenomena. From this situation, the crew proceeded with the priority of enhancing durability and reliability, with HRD rearranging components that HGT had sent out, but every time the logistics took until just before test time. Everyone worked late into the night to conduct stopgap solutions and change out the gearbox at the running test site as we endeavored to somehow stay within the schedule.

Gradually we could see the replacement parts taking effect, and with the start of long runs, drivers began making such positive comments as:

"Shifting was seamless, with no sense of reaching a plateau. I could even shift up while cornering and the rear did not show any turbulence, and each time I shifted up it shrank the distance between me and the car in front." The new system was proven to reduce lap times by 0.2 sec/lap compared to a standard gearbox. Unfortunately, during the 2004 season the system was unable to meet the 500 km durability requirement, so it was introduced the following year, in the opening race of 2005, when it became the first such technology used in a race. And with that, the world's greatest gearbox was complete. At the circuits, other teams refer to this as seamless shifting.

In any case, after accepting all the facts, the assembly team had accomplished a transfer of work responsibilities to BAR, from creating systems and environments for supplying test equipment to running race operations.

3.6. Adding a J Valve

The J valve system added in 2005 was developed to reduce valve-train system friction.

In the structure used up until that time, one-way valves were placed in the air chambers that work as valve springs at each of the various entries/exits, so that even if in the worst case the air supply died, the valve would not fall from its own weight. These one-way valves were eliminated, however, and each air chamber was given an orifice to create a continuous passage, so that if air leaked from one air chamber, it would interfere with the valves of all cylinders. The crew learned that the engine quality could not be assured with this new system as in the past.

The engine assembly method had also changed, but above all else the mechanics put their energy into transportation methods and methods of operation from onsite storage to engine startup, and most especially on a technique of lubricating the valve stem with oil.

During engine startup there was concern about the valve stem getting burned, and what caused the most hardship of all was how to control the amount of oil inside the engine, which the crew could not see.

During operation, the system banged against the wall and we were not at all sure it would be OK.

- What is the minimum pressure needed to keep valves from falling from their own weight during transportation?
- What is the minimum pressure for valves to track movement of the camshaft when the crank is being turned by hand?
- What are the temperature and pressure that would make air leaks unlikely based on characteristics of the PVRS piston seal?
- Can oil be purged from air lines if collected and retained in air chambers?
- What capacity is necessary in large, high-pressure air cylinders?
- Can the engine be shipped by airplane with a large, high-pressure air cylinder attached?
- How can one check the initial lubrication of valve stems?

These worries were dispelled by testing each item one-by-one.

In all cases, while engines were being assembled at HGT, sent to the circuit and returned to HGT again, we constantly attended to the engines and collected data on them and finally determined the requirements.

We proceeded with particular caution when the technology made its racing debut in 2005, and the J valve proved it had reached completion on the race track operations side, and it later became the standard.

We continued to collect data after that, to the point where, by 2008, we had successfully simplified the technology on the equipment and operations side in large part.

The technology could then be normally used on an engine without any of the anxiety as in the early days of development. The supervisor of J valve development at the time said something I never forgot:

"Whether the J valve will be successful or not all depends on how it's used on the race track. I expect a lot, especially of the mechanics."

3.7. Extending Engine Mileage

Previously it was possible to change engines on Friday, Saturday and Sunday, but regulations starting in 2004 required that engine mileage be extended and that only one engine be used per race. The mileage requirement was further extended in 2005, so that a single engine had to be used in two races, starting with practice. The distance each engine was used went from 300 km to 800 km to 1500 km. Instead of simply extending the mileage of an engine with the same specifications, the engine framework specifications should be altered in the course of steady development to achieve both durability and reliability on the one hand, and performance on the other.

The penalties for changing engines were severe: if the engine was changed because of trouble occurring within the required usage distance, the driver would be penalized 10 slots if it were before the qualifying, and if it were changed after the qualifying, the driver would have to start in last place.

Before the penalties were imposed, if trouble occurred, the team could weigh the time it would take to analyze and repair the defect against the time it would take to change the engine, and choose whichever was faster.

If done quickly, an engine can be changed in 30 minutes.

Of course, the pit was set up to deal with such situations. Nearly all the components concerned with the engine, including radiators, oil coolers, oil tanks, exhaust pipes, clutches and hydraulics systems, were installed in the pit.

At each circuit, there were always practice engines, qualifying engines and race engines for two drivers, as well as three spare engines and one engine for a training car. This meant that there were at least 10 engines in the garage, and of course not all of them could fit. On older circuits in particular, garage space is limited and it is a very challenging work environment just to load and unload engines from their special transportation boxes.

In a complete turnabout, the change of regulations eliminated the work of changing engines every day as well as reducing the total number of engines, so everyone thought there would be less work to do.

Everyone was wrong.

Now the crew had to check the inside of the cylinders each day after the engine's run, instead of only when an abnormality occurred. A fiberscope was used to detect any piston scuffing inside the cylinder, which was determined by the appearance of the sleeve (Fig. 4).

Another important check item was to detect any pitting in the spline of the input shaft that couples the crank tail-end to the clutch housing.

If any pitting were discovered on the spline on the crank side, it was polished away. If this is overlooked or ignored, it can start a process that results in the crank breaking and metal burning as the engine fails to maintain oil pressure.

Removing pitting marks with a compound and diamond rasp is a difficult job because the crank is so hard. The question of whether an engine can be saved is a heavy burden on the mechanic in charge.

Flushing with oil was also done after each event. The objective is to get rid of any degraded oil still clinging to the conrod pin metal and main journal metal. This job is essential after each race, because experience has shown that engines with stagnant oil in them for an extended time will occasionally suffer metal burning, either in bench tests or on the circuit.

It was also unavoidable that components would have to be changed that did not assure sufficient durability.

The crew had to use every possible means to meet the high hurdle of 1500 km.

The engine truly was a lot of work.

At the circuit, although ideally there should be no more of these jobs to do, they became a regular feature.

Maintenance of an engine to keep it in continued use depends in many ways on the mechanic's competence and experience, and if just one thing is overlooked, it dooms the vehicle to withdrawal from a race.

Fig. 4 Checking inside of engine

The assembly crew discussed this a lot with the design, research and electrical staff. Looking back now, it seems not so much a discussion as a pathetic plea from us to them to "Please fix this engine right away, and please get it out to the track so we can use it with confidence." A major part of this plea was the fact that each race was contested with a special kind of pressure: the driver's life was in our hands.

Even aside from the need for continuous maintenance, the requirement of two races for each engine put great pressure on mechanics in other ways.

It was no longer simple to decide whether or not to change the engine, and now the mechanics had to analyze malfunctions and do repair work with greater accuracy and speed than ever before.

Accomplishing this created a need for tools with greater functionality and a wider variety of components than before.

The amount of equipment and components brought onsite increased to the point where the crew could assemble an entire engine from them.

Of course, there were also enhancements in maintainability and work efficiency.

The veterans were fond of telling us, "The race does not wait; this is a battle with time," and in fact we were often forced into such situations.

In a sense, the changes in regulations created a challenge to the mechanics to provide quality assurance of the engines.

3.8. Launch of SAF

At the end of 2005 it was decided to start supplying engines to SAF. Five members of the first crew in charge of this area entered the UK on January 12 of the following year. At the SAF factory in which they settled, the rising sun flag of Japan was a big landmark in the rural area of Leafield west of Oxford, England.

This day was the start of a big challenge.

When the crew arrived, SAF had fewer than 20 employees and no vehicles, equipment or components. That was the first impression: they don't have anything. This was quite different from supplying engines to an existing, already organized team, so the crew had to do double duty by also launching a team.

Their first goal was to get two vehicles to the final round grid in the first race of the season. As the crew gave instructions, everything had to start from the very beginning: preparing tools and equipment spare parts; changing the engine auxiliary components (replacing BAR specifications with SAF's); integrating engine operating rules (post-arrival, return shipping methods, warm up, oil management, pre-mounting status, etc.); engine startup and running procedures; distance-controlled components; and so on.

Everything started with dialogue.

Having set foot here, the first crew continued working patiently, keeping its feelings under control.

On January 17, the initial vehicle (the base was an old Arrows A23 built to 2002 specifications, which had been out of the Formula One world for an unusual four years) was brought in, after which the team modified the monocoque to allow Honda's engine to be mounted and modified the chassis to enable it to pass the FIA's collision requirements. Then, equipment was introduced and physical production began.

Ten mechanics were also hired, so that on January 30, vehicle assembly began. Everyone worked on production, which was all done by hand. There were no differences among members, and even the chief designer and technical director were put to work. These people responded well, and we had no troubles in terms of communication.

Everybody was helping each other with their roles and responsibilities, without anyone getting in each other's way. It was all positive and forward-looking, and with everyone cooperating, there was a strong sense of teamwork.

Day 32; Shake-down of initial vehicle (SA05-05)

Day 39; Barcelona joint testing (SA05-05): time was not an issue here, but the vehicle ran 590 km.

Day 58; First race, practice stage, two vehicles began running

Day 59; Got through qualifying stage with 20th (SAT) and 21st place (IDE)

Day 60; Two vehicles line up in the final round grid

Having met the first target must have given the crew a sense of accomplishment, but they were actually quite cool about it. It seemed as if they took it for granted that we would be in time for the opening race. Those two months seemed to go by very slowly, but this was just the starting line of the season.

Although the team did not bring home any trophies, it attracted increasing attention. With each race, the look on team members' faces became sterner. There should be no failures and no delays, so their expressions changed.

As the races went by, we ran out of spare parts and team vehicles had to retire more often, emphasizing the fact that we were not making progress on the vehicle. Starting from Rd12, we added the Honda gearbox (quick-shift) and gave technical instruction.

Along with that, we changed the suspension and the vehicles advanced significantly, which enhanced vehicle performance and provided valuable racetrack experience. Prior to qualifying rounds, gear ratios were changed on three vehicles, including the training car. This work concerned a gearbox with a new mechanism, and the SAF mechanics did not have any onsite experience changing the new mechanism. Two hours before the qualifying, the three vehicles were lined up for presentation, and the team had acquired its first experience working in an actual race. We recognized again the importance of maintainability and reflecting enhancements. From there, we began special training to give the team mechanics more independence.

In 2006, SAF was born as a new team and competed in all Formula One races.

Everyone who set foot in Leafield that year had a part in this dream.

3.9. KERS Development

Even the Formula One world is expected to care about the environment, and it was decided to change the regulations greatly in 2009.

That is when the kinetic energy recovery system (KERS) was introduced. It became a matter of some urgency to develop a hybrid system, a Formula One version of the INSIGHT, as it were. The reason hybrid cars are considered so good for the environment is because of their good fuel economy. And in a race, of course, competitive fuel economy is a huge advantage strategically.

The regulations, furthermore, allowed output of up to 60 kW of electric power charged from the motor during deceleration, with engine assist of up to 400 kJ of energy released per lap; functionalities like increasing maximum speed and enhancing acceleration coming out of corners were expected to reduce lap times. This created a need to develop a lightweight, compact and efficient system, and Honda did all the development of the motor, PCU and battery in-house. Having this development done in-house was of particular importance to the mechanics as they headed out to the racetrack.

They would absolutely need a thorough familiarity with this new system in order to gain the knowledge and technology to be able to safely and flexibly deal with malfunctions on the circuit.

If one were to use a purchased item, there would be no way to know what is inside. In other words, a purchased item is like a black box, and there is nothing more to do with it.

In contrast, the crew had no excuse for not understanding something developed at HGT. Fortunately, the motor was built by the assembly department.

At first glance, no one looking at this motor measuring a mere 105 mm x ϕ100 mm would think that it creates even as much output as one FIT motor.

One can easily imagine that developing this motor took much more than the average effort.

Everyone on the crew started as a novice in terms of motor assembly.

It also required a very complex way of winding completely different from a mass-produced motor, and was only completed with a great deal of trial and error.

UN rules on handling batteries proved to be a great obstacle.

Using a battery lithium ion, individual cells were combined into modules, which were then combined into packs. The mechanics thought it would be the packs that they received at the site.

But what was sent from Japan to HRF in England was a large number of individual cells and a welding tool to join them together.

It takes at least three months to get permission under UN rules to ship cell modules and packs, and a new application is necessary each time specifications change. This would be fatal in the Formula One world, which has very short development spans, and we had never felt how far Japan was from England until that point.

Our goal, nonetheless, was to never give up, and to be the first to do this task successfully, and the only way to do that was to have the mechanics do the welding in England. Still, the welding effort had to start from zero.

A mechanic had to learn the special welding technique in a short time and then bring it to England, where the welding was done in a corner of the HRF warehouse. As we struggled with this we paid close attention to safety.

Each cell has a slight charge to stabilize it, and there was some concern that during assembly, an error could electrocute the person. Of course, the people doing the welding first took a course in electrical safety, and even went to Honda’s Suzuka Factory, where hybrid cars are mass-produced, to observe there. To be honest, we thought that electricity was so frightening because it is invisible to the eye, but as our knowledge grew, we mastered ways of protecting ourselves from danger.

The mechanics undertook to identify 187 malfunctions and take care of pending items that could occur by the time the motor, PCU and battery were mounted to the vehicle.

Not only were there assembly malfunctions from parts touching each other, but assembly work procedures and specifications were decided where there had been no blueprints or instructions relating to reliability and work characteristics, from putting on the harness to assembly procedures.

To ensure safe operation at the site, everyone discussed what was safe and what was dangerous, based on system diagrams. The team checked each work process, starting with the preparations prior to engine startup and continuing through post-cruise dismantling (Fig. 5).

As this was going on, the Internet started circulating news and photos of a mechanic with BMW Sauber getting a shock on a circuit during a test driving. It shook up everyone and showed that there was disaster lurking in unexpected places.

Ultimately, it reminded us that, to protect ourselves, each person had to be aware of his own safety.

The result was mechanics with a wide body of knowledge and skills and an awareness of safety in areas completely unlike the past.

Fig. 5　Checking KERS system

In April 2008, actual driving development began with a shake-down test at Santa Pod in the UK. It was hard going, since there was a series of issues we had never experienced before, but with this event the crew succeeded with a KERS road trial for the first time in Formula One.

There were subsequently two more shake-downs and a driving on the Jerez circuit in Spain, and in November the crew was back in Santa Pod for the fourth development shake-down, and during these events we overcame the various issues.

We faced phenomena such as being unable to pre-charge or unable to connect the main conductor, the cause of which turned out to be a broken contact point melting damage prevention circuit when connecting the main conductor incorporated into the high-voltage circuit in the battery. Because the battery of the third development lot was stored under the driver’s seat, we tried connecting the main conductor manually from the outside by having the driver step out of the monocoque and removing the seat each time the engine was started. This connection procedure was complicated and required much time.

In Formula One development, which constantly requires more speed, no matter how tiny the part may appear, this was indeed a serious trouble.

Moreover, the effect of electrical noise on the communications system prevented the technology from providing full output, and there was nothing more that onsite analysis could do in this state. To give priority to analysis of the electrical noise, the team canceled their participation in the next Paul Ricard Test, and the vehicle was hurriedly sent from England to HGT. Of course, the mechanics also went home with regret in their hearts.

At Spain’s Jerez circuit the vehicle made a new start, the driver’s evaluation stating, “I still have a lot to learn about KERS, but I was impressed by how amazingly fast I approached the hairpin when the assist kicked in. It makes non-assisted acceleration from the engine alone feel as if we are carrying a heavy weight.”

The fourth development lot system, which was more lightweight and compact, also suffered frequent issues with communications line noise, and the mechanics providing support at HGT took a number of countermeasures, like adding a condenser to the PCU, adding ferrite and enhancing locations with defective grounding.

It was the first time for assembly team members to touch the PCU. In the meantime, HRF actually performed more than 100 engine startups and put their energy into identifying the source of the electrical noise. The PCU on which measures had been taken reached the site just barely in time. The electrical noise had been eliminated.

Then came the December 5, 2008 surprise announcement that Honda was pulling out of Formula One.

That marked the end of a solitary 12-month battle the mechanics had waged in anticipation of a new era.

4. Afterword

Looking back on nine years of activities, Honda's third era of Formula One started with the development and supply of engines. The effort subsequently responded to diverse needs, from meeting development requirements to developing a gearbox, conducting vehicle running tests, developing KERS and, moreover, launching a private team. Although Honda's withdrawal from Formula One ended an era and we were never able to ascertain the competitive strength of the 2009 vehicle, we believe that the racing spirit cultivated here will one day shine again in a new age of racing.

While we did not unfortunately leave a great record of race success, the guidance given by our veterans helped to pass down their mindset. We are moreover thankful for the cooperation of all departments concerned with development that helped us to carry out our duties.

■ Author ■

Tsuyoshi ISHIHARA

Naoto SUNAKO

Atsushi NORIHARA

Motoaki ISHIZAKA

Activities of Honda Racing Development

Yusuke HASEGAWA* Shiro HISATSUNE** Tatsuya KODAMA**

ABSTRACT

In conjunction with the launch of our Formula One project in 1998, Honda Racing Development (HRD) was established as the British base for administering the Honda Formula One team. Subsequently, plans were modified to supply engines and joint development efforts with the BAR Team, and HRD came to assume the role of Honda's frontline European base for timely project and decision-making. This article describes the roles and accomplishments of HRD.

1. The Roles of HRD

All engine development was formerly carried out at Honda R&D Co., Ltd. Tochigi R&D Center (HGT), including design, assembly, testing, and shipping (including performance checks and packing). Since the Formula One team, which was a customer for Honda engines, was based in the UK, and since almost all the running tests were conducted in Europe, HRD has functioned as a logistical supply base for providing engines and parts for races and running tests in a timely manner. It was the site not only for receiving and managing parts sent from Japan and then sending them to the circuit, but also for reassembling engines, checking shipments, and procuring parts locally.

The technical staff that manages the testing site was initially made up of mostly of visiting personnel from Japan, but as ties with the team grew stronger, it became essential to assign regular personnel to the UK base in order to enhance communication between the two facilities. Employees from several sectors, including Engines, Electrical Components, Materials, Design, and Assembly, were dispatched to HRD on long-term assignments, and they performed the pivotal roles of onsite administration and maintaining ties with the team.

The Formula One Team and HGT were each assigned roles in chassis development, but in the interests of sharing technology and enhancing the abilities of the team, HGT staff members took part in each area of the team's concerns, including Aerodynamics, VDG, Composite Materials, and Mechanical Design, and worked on chassis development as members of the team.

In addition, HRD took advantage of its location to promote personal interaction with other Europe-based Formula One teams and suppliers, and gathered technical information.

2. Location

As shown in Fig. 1, HRD is located in Bracknell, almost in the center of Berkshire, about one hour west of London by car. This location is 80 km from the Formula One team's base in Northamptonshire and 20 km from Heathrow Airport, making it a convenient location for Formula One-related activities, which require a lot of travel.

The layout of HRD, comprising an office building and a factory, is shown in Fig. 2.

In addition to local British employees, half the staff was made up of Japanese employees on long-term or temporary assignments, which made HRD's employee-mix somewhat different from that of other overseas locations. Many Japanese employees are there on short-term assignments, and in addition to working on projects, HRD also played a major role in managing housing, hotels, and travel arrangements for the Japanese employees.

* Fundamental Technology Research Center

** Automobile R&D Center

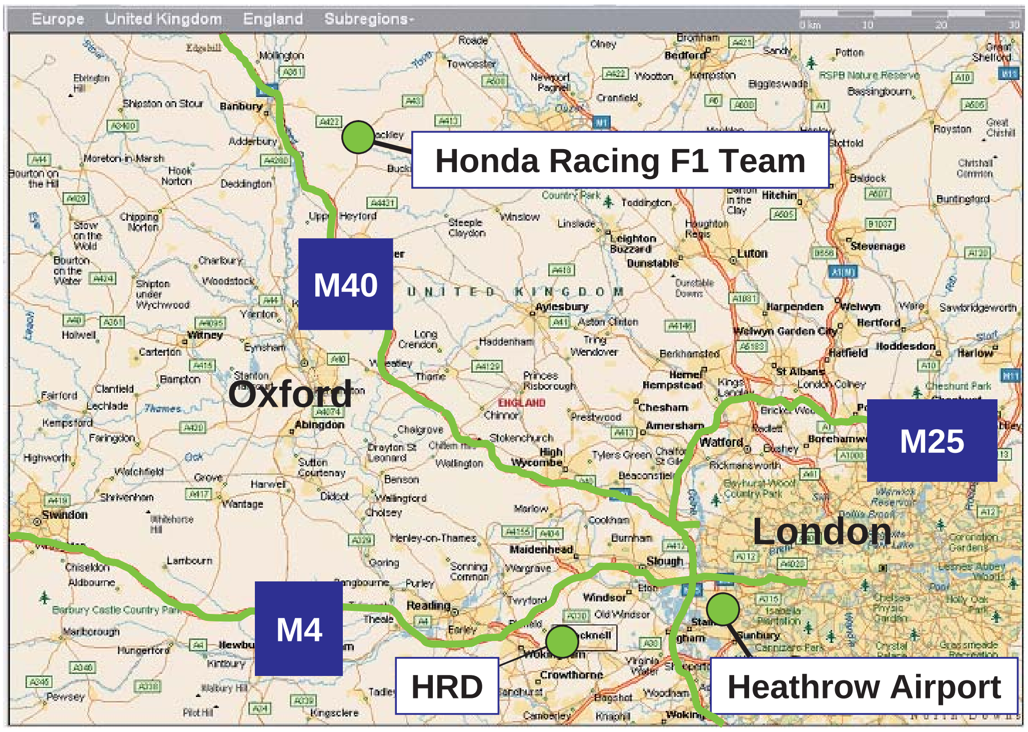


Fig. 1 HRD Location (Source: Microsoft AutoRout)

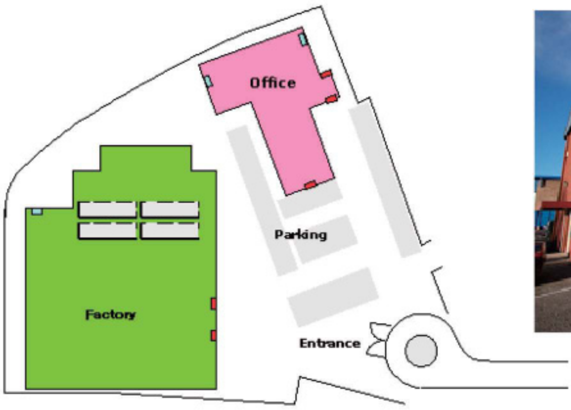


Fig. 2 HRD Factory and office layout

3. Factory Management

Figure 3 is a chronology of activities at HRD. The Formula One team was BAR at the beginnings, and then became BAR-Honda, with a 45% investment by Honda. During the transition to all-Honda-owned HRF1, HRD's functions also evolved one after another. The division of labor between HRF1 and HRD can be essentially described as vehicle body on one hand, and engines on the other.

Initially, HRD's function was to serve as a contact point for BAR, FIA and other Formula One associations, and as a logistics and supply base. However, as BAR HONDA shifted to Honda Racing Formula One (HRF1), contact point for BAR became no longer necessary and the contact point for FIA and other organizations was also shifted to HRF1.

Management of the factory, which began in 1999, started with a focus on engine maintenance and parts supply to testing sites.

Partly due to almost yearly modifications of regulations and changes in dealing with the second-string team, HRD expanded its functions beyond those of a logistics and supply base. From 2005, HRD functions have been shifted away from only engine and parts supply and replacement, making HRD an organization that takes advantage of its location and its mobility for primary analysis and solution of racing test issues, initial powertrain checks and settings before running tests, local parts procurement and other such functions. It also worked on gathering information by investigating European technology, which will be discussed later.

Due to modifications of regulations and other factors, the number of engines supplied decreased or increased as the occasion demanded, but the share of engines shipped from the UK was increased. The reason for the increase was a shift of the routine business of shipping engines to HRD, with the objective being to allow the Japanese side to strengthen its development efforts. In fact, an initial ratio of approximately 4:1 (HGT:HRD) had shifted to 1:3 by 2008, leading to an environment in which HGT could more easily serve as the main player in engine development.

Primary analysis and solution of racing test issues, and powertrain initial checks and settings before running tests, which were the core of HRD's expanded functions, took the greatest possible advantage of its location. HRD conducted bench tests, including initial settings, before shakedowns of new vehicles or functions, helping to prevent issues from occurring during initial runs.

As shown in Fig. 3, the increase in engines supplied, early analysis and solution of issues, and enhancement in engine quality led to changes in the HRD facility. HRD was equipped with dynamometers and materials inspection devices. At first there was only one stationary dynamometer, but as HRD's role changed, transient dynamometers were added, and powertrain settings performed. At the same time, the Materials Department upgraded its equipment to include such things as a scanning electron microscope, making it possible for the department to analyze factors that contributed to issues at the test site and to make the primary judgments. A five-axis milling machine and three-dimensional measuring machine were installed, and a clean room was introduced in connection with assembly machinery and inspection equipment. In this and other ways, efforts were made to enhance parts inspection and quality and efficiency of the assembly process.

These actions facilitated the HRD factory's basic role as a frontline base for managing races and running tests.

4. Races and Running Tests

Of the 18 races in which the team participated in 2008, there were eight Grand Prix races that required travel to distant countries such as Japan or China and 10 races in the European round. Some of the teams participated in running tests in Bahrain, but Honda conducted all of its tests in Europe, mainly in Spain.

Parts were transported overland by trailer or truck in Europe, while engines from Japan were shipped to the UK by air, after which they were transported to the circuit overland along with the HRD engines.

When there were overseas races, the components that were normally transported by trailer were packed into a crate for long-distance shipping commonly called a black box, and transported by air. Some components were used only for overseas races and were usually stored at HRD.

In addition to engineers and mechanics, the racing staff consisted of support personnel, such as truck drivers. There could be more than 30 persons from HRD alone, in addition to the team members, making a total group of nearly 100 persons comprising one team. On the circuit, this included staff members who prepared meals for team members and their guests, since each team had to prepare its own meals.

An important duty of the HRD onsite staff was to guarantee the reliability and performance of engines on the circuit, and in order to get the maximum performance out of an engine, they had to set the engine equipment and data according to the weather, the features of the course, and the race strategy, as well as the driver's own

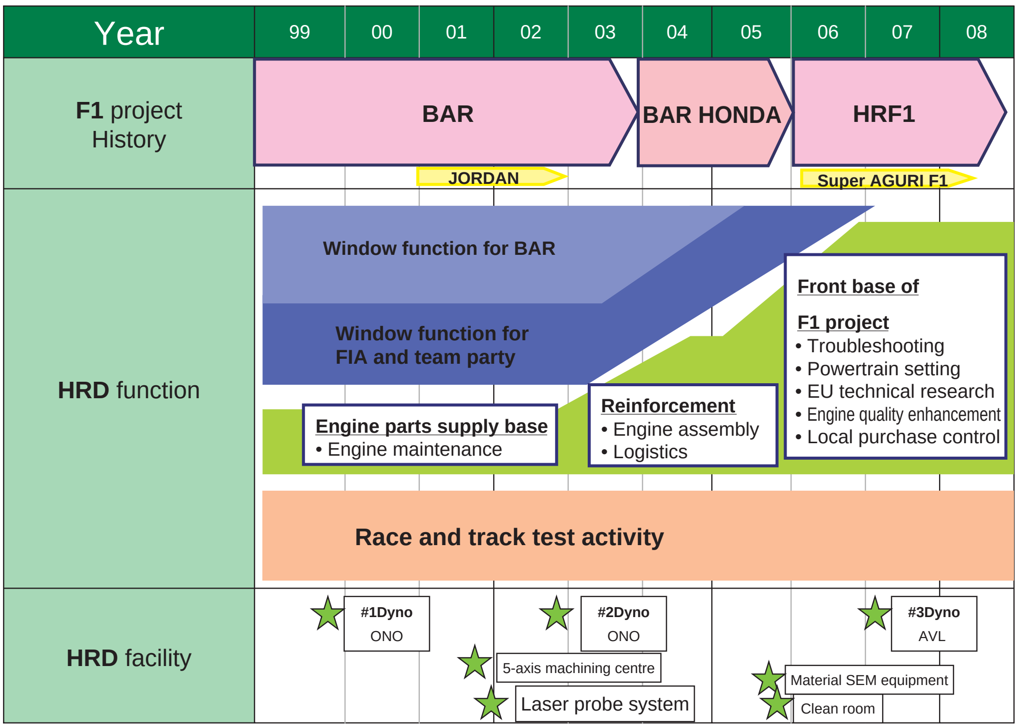


Fig. 3 HRD history

driving habits and preferences.

Running tests were thought of as the final checks for items that already had been checked for performance by unit test and bench test. Drivers' comments and data were used to check performance, while checks of race mileage reliability, temperature, and vibration were among the tests performed to check conformance to actual vehicles.

Ignition timing settings were changed in 0.5-degree increments based on the outside air temperature, humidity, and engine temperature. The whole process was painstakingly managed, so that, for example, during a pit stop in the middle of a race, drivers were told to change the engine mode if the engine temperature was too high.

Since shift-timing and straight-end speed affect lap time and frequency of maximum engine speed, gear ratios were selected with consideration for the features of the circuit and the direction of the wind, and in consultation with the vehicle engineer. Since 2008, gear ratio exchanges have been limited to one per race, so ratios have to be selected in advance in anticipation of the preliminary rounds and the final race. This led to serious discussions after the practice runs on Friday. Since cooling performance is directly linked with the reliability of the hardware, it is important to ascertain the cooling settings of each part at the assembly stage of new vehicles in particular. It was therefore adjusted for throughout the season, depending on the outside air temperature and use environment. In reality, however, it was difficult to determine everything beforehand in low temperature conditions during winter tests, so we often struggled with onsite settings due to unanticipated intense heat or unusual weather conditions.

It is usually difficult to determine whether or not an engine setting is appropriate through data alone, so settings were carried out in consultations with the driver and the vehicle engineer, as they determined what points could be enhanced and what the general settings should be. For example, when the safety car was introduced, the lean driving limits for saving fuel were determined by the driver in the midst of joint runs, and on the basis of whether engine response would become an issue. The pedal-to-torque characteristics in a drive-by-wire situation, as shown in Fig. 4, were tuned for each circuit based on the driver's comments.

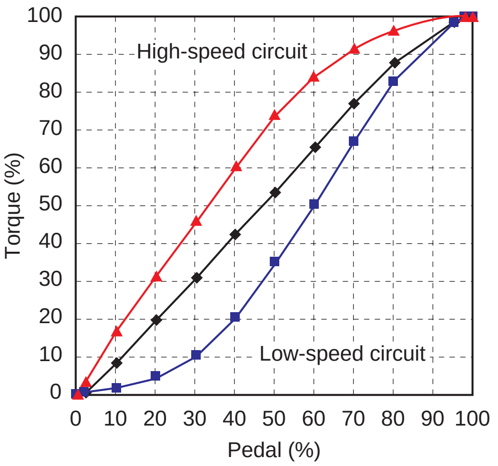


Fig. 4 Pedal-to-torque characteristics

When engine blow and other mishaps occurred, primary factor analysis was conducted on the circuit, and the secondary analysis was conducted at HRD, depending on the urgency of further investigations. HRD often had to take emergency measures when trouble arose during a race or during the test runs just before a race.

5. Investigating European Technology

Another responsibility of the onsite personnel was to gather information about trends in regulations, competitiveness, and technology, whether through official or unofficial channels. Since it was difficult for HGT to gain an overview of this information, it was considered to be one of the major roles of onsite personnel to feed back their perceptions of directions in technology to development.

Programs that took advantage of HRD's location as a Formula One base in Europe were the Formula One Engine European Technology Research and Manufacturers' Joint Development Project, which began in 2006 with HRD local staff playing the central role.

We first contacted Formula One engine builders, design consultants, parts manufacturers, and simulation development manufacturers who were based in Europe and surveyed them, comparing Formula One engine technology trends in Europe with Honda's engines.

As a result of comparative surveys, it was found that Formula One engine trends in Europe revolved around systems with high-lift, narrow-duration camshaft profiles, lightweight valve-train systems, and cam gear-train damper systems.

In particular, they led the way in the development of cam gear-train damper systems, with damper characteristic specifications being determined with gear-train simulations.

Figure 5 and Table 1 show the names and locations

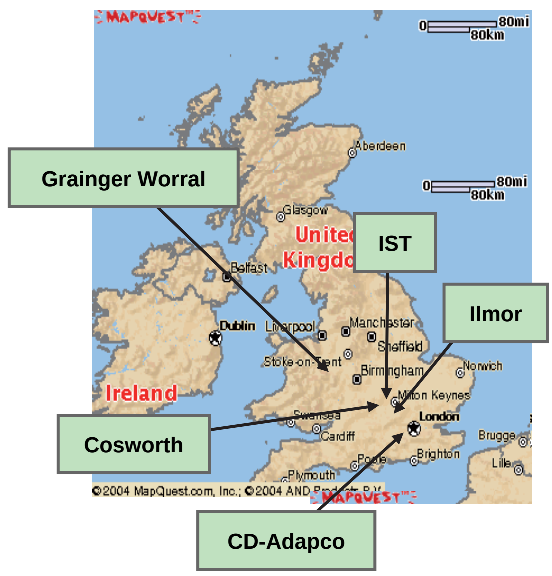


Fig. 5 Supplier location (Source: MapQuest)

Table 1 Supplier and development items

Supplier	Company profile	Development item
IST	Consulting company for F1 engine design and material	EU concept cylinder head assy
		Inlet valve (Ti3Al) (high stiffness / light weight)
		High-lift + narrow-duration cam profile
		EU concept cylinder block assy
		Light-weight piston (5-axis machine)
		Twin oil circuit system (piston jet - crank)
		EU concept air box
Ilmor	Ex. F1 engine design and manufacturer	8-cylinder LAF sensor +Real-time mapping
		EU style head-port design
		High-energy IG coil
Cosworth	Ex. F1 engine design and manufacturer	Tortional damper for cam gear-train
		Gear train (+damper) simulation
Grainger Worrall	F1 casting parts manufacturer	Forced-air chill-casting for cylinder head
CD-Adapco	CFD simulation development	Port / Expi / Airbox 3D CFD simulation

of the manufacturers with which HRD carried out its joint development projects. Most of the development projects were eventually carried out with manufacturers in the UK, and this is believed not unconnected to the fact that Mercedes High Performance Engine, Cosworth, Ilmor, and other top-ranked racing engine constructors are based in the UK.

These joint projects allowed HRD to propose the aforementioned high-lift, narrow-duration camshaft profile, cam gear-train damper system, and simulations, and their effectiveness was verified in preliminary tests conducted with HGT. Subsequently, in 2009, while they were preparing to manufacture parts with the aim of adapting this technology for the Formula One engine project, Honda decided to withdraw from Formula One racing.

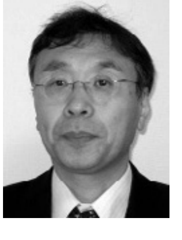
6. Conclusion

At HRD, which functioned as a frontline base far away from Japan, both short-term and long-term personnel encountered difficulties due to the overseas working environment and unique strictures of racing. As we struggled on the front lines in direct contact with races and teams, we learned a great deal and were able to grow.

We would like to take this opportunity to thank everyone inside and outside Honda who constantly gave priority to local programs and supported and cooperated with us.

■ Author ■

Yusuke HASEGAWA



Shiro HISATSUNE

Tatsuya KODAMA

Concluding Remarks for Honda's Third-Era Formula One Activities

Katsuhiko SUZUKI*

To all the members of the Formula One Project, I would like to say, "Thank you and well done."

To all our friends and colleagues inside and outside the company who generously gave their support, I must offer apologies. Our strength was not sufficient to keep the project going, and so it came to an end before you could enjoy it to the full. I am truly sorry.

My own involvement with this Formula One project started on August 31, 2007. That was when I received a telephone call from Vice President Ikeno of the R&D Center. "I want you to become Operating Officer of the MS Department (the Motor Sports Development Department) from September 1," he told me. "But September 1 is tomorrow," I said, and even as I spoke those words, I remembered that something very similar had happened to me before.

This was back when I was still an assistant chief engineer, and it was the afternoon of the day before I was supposed to leave on an overseas business trip when a telephone call came from the Manager. He said, "Everything has been called off! You're going to start on a special assignment tomorrow, instead!" At that point, all I could say was a very vague, "Huh? What?" This had taken me completely by surprise, and I had to wonder whether they could really do this. If you said this was a Honda-like way of doing things, then I would have to agree, so far as that goes.

In 2007, I was General Manager of the Mass Production Engine Development Division, and at that point we were involved in some very challenging issues related to a major mass-production start-up under extremely demanding conditions. When this latter request came, therefore, I insisted that I should wear two hats until March 2008, at least, and they let me simultaneously remain as Operating Officer of the Mass Production Engine Development Division and Operating Officer of the MS Department. As it turned out, it might have been better if I had concentrated on the MS Department. It seems to me, looking back at it now, that the MS Department actually was in greater need of that level of management and engineering.

When I took responsibility for the MS Department, I realized that our biggest sore point was communication:

- Communication among the Honda Formula One Racing Team (HRF1), the Honda R&D Co., Ltd. Automobile R&D Center in Tochigi (HGT), and Honda Racing Development (HRD);
- Communication between the Honda Motor Co. head office (HM) and HGT; and
- Communication among the HGT engine, chassis, gearbox, assembly, and operations divisions.

Unless communication of this kind is taking place smoothly, things do not get decided properly. And even if they do get decided, positive results cannot be expected unless a consensus is also reached.

What I did, therefore, was to have some long talks with Ross Brawn, the HRF1 team representative. As the individuals responsible for respective development in the UK and Japan, he and I set out to clearly define the roles and responsibilities of the various divisions concerned. The result was that the mobilization of our intentions and technologies regarding the 2009 car proceeded according to schedule, and is currently contributing to the activities of Brawn GP, which took over the project.

Although our effort ended before KERS could be used, we did manage to build up a system with the world's highest level of technology in this area, as well. This includes the super compact 60-kW motor, the PDU with ultra-fast processing capability, and the laminated lithium ion secondary batteries. I feel quite certain that these technologies are sure to be of use to developers in some form or other in the future.

Next, on the matter of communication with HM, both President Abe of HRD and General Manager Muramatsu of the head office Motor Sports Division shared a common perception of issues with us, and discussed what needed to be done on both hard and soft aspects of the project each time something came up. This included three conferences with HM, HRF1, HRD, and HGT that took place during the race season.

Regarding the matter of communication in the MS Department, which is an issue within HGT, I have the impression that in a time like this, when engine development is subject to such rigorous controls by the homologation system, the competition of technology has shifted over to the comprehensive enhancement of vehicle dynamics performance. However, the outstanding talents in the Vehicle Chassis Division at HGT were not in a position to demonstrate everything they were capable of doing. I think that what turned this situation

* Automobile R&D Center

around for them was the neutral thinking of Ross Brawn. Just one example is the rear diffuser for the 2009 car. In the development competition between HRF1 and HGT, the HGT proposal was adopted. Beyond that, they have also reinforced their contingent of HGT members stationed in the UK, and have started actively taking part in concept meetings for coming vehicle versions. The knowledge they have cultivated in this way is an irreplaceable resource that is in the keeping of the younger HGT members.

This is not just the case with the Vehicle Chassis Division: the Engine Division has also adopted a special framework for development. With the engine for the 2009 season having been partially released from homologation by the International Automobile Federation (FIA), the Engine Division went from development emphasizing peak output to development emphasizing mid- to high-range torque according to lap-time simulations. They accomplished this in the short time-frame of three months. The first model of their engine ran on the test bench and generally yielded the performance they desired. As I see it, realizing the ability to design engines from the conceptual approach of overall optimization (namely, lap-time simulation) as seen in this engine development effort contributed to the integration of communication by the MS Department and evolution by the designers. I was also impressed all over again by the pride and the potential we found in these engine men. It took them less than two months from starting development to having their first engine running.

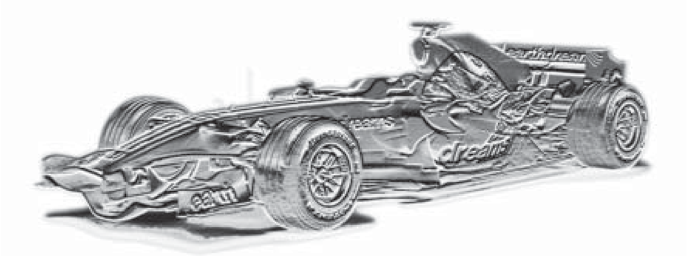
Ever since the seamless shift mechanism was developed, Honda has constantly held the lead with the most advanced Formula One gearbox technology. The lightweight, compact iMAX shift mechanism built into the shaft that had been scheduled for adoption in 2010 is the ultimate one of this kind. I hope that this kind of thinking, which pursuits the ideal goal and then achieves it, will go to work in mass-production, as well.

The Assembly, Procurement, and Operations Divisions have performed extremely well as unsung heroes in supporting roles. I feel that they served as my own two hands. They really wanted to help with winning, and I could feel the sincerity in our communication at that level. They have a full understanding of their respective functions, and they were developing frameworks for cooperation with HRF1, such as the dedicated system development project aimed at realizing smooth logistics.

I was able to spend a very enjoyable year and a half, thanks to the efforts of all these people, and I am truly grateful. Thank you very much. Finally, when conditions take a turn for the better, I fully expect that Honda will be making a comeback to Formula One racing.

After all, Formula One is the very pinnacle of racing. It is the ultimate discipline where victory can only be gained when the elements of technological capability, full range of speed, and management capability are all present.

Race Records



Chassis, Engine (1999)

■RA099 Prototype

Chassis specifications (RA099)	
Wheelbase (mm)	3020
Gravitational center hight (mm, above reference plane)	250*
Main monocoque weight (kg)	54
Tire supplier	Bridgestone
Tire size-front	265/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Engine specifications (RA099E)	
Engine Type	90°V10
Displacement (cm^3)	2995.6
Cylinder diameter (mm)	94.4
Stroke (mm)	42.8
Weight (kg)	122
Maximum power (kW)	563 Over
Maximum Ne (rpm)	16300 Over
Engine electronic control system	Honda

Honda	Chassis ; RA099 Engine ; RA099E	
Test Driver		Jos Verstappen (Netherland)

Chassis, Engine (2000)

■BAR002

Chassis specifications (BAR002)	
Wheelbase (mm)	3108
Gravitational center hight (mm, above reference plane)	259*
Main monocoque weight (kg)	61
Tire supplier	Bridgestone
Tire size-front	265/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR002)	
Gear box case	Aluminum Casting
Changing mechanism	Xtrac
Forward gear ratios	6 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■RA000E

Engine specifications (RA000E)	
Engine Type	88° V10
Displacement (cm^3)	2995.6
Cylinder diameter (mm)	94.4
Stroke (mm)	42.8
Compression ratio	13.4
Weight (kg)	112
Maximum power (kW)	567 Over
Maximum Ne (rpm)	17000 Over
Mileage (km)	400
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR002 Engine ; RA000E		5th
Driver	CARNo.22	Jacques Villeneuve (Canada)	7th
	CARNo.23	Ricardo Zonta (Brazil)	14th

Chassis, Engine (2001)

■BAR003

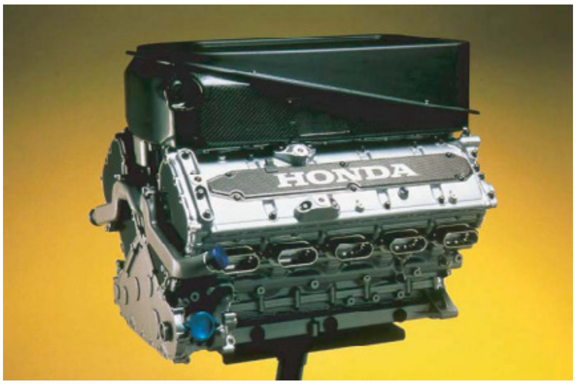
Chassis specifications (BAR003)	
Wheelbase (mm)	3183
Gravitational center hight (mm, above reference plane)	245*
Main monocoque weight (kg)	65
Tire supplier	Bridgestone
Tire size-front	265/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR003)	
Gear box case	Magnesium Casting
Changing mechanism	Xtrac
Forward gear ratios	6 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■EJ11

■RA001E



Engine specifications (RA001E)	
Engine Type	80° V10
Displacement (cm³)	2994.1
Cylinder diameter (mm)	95
Stroke (mm)	42.24
Compression ratio	13.4
Weight (kg)	108
Maximum power (kW)	588 Over
Maximum Ne (rpm)	17300 Over
Mileage (km)	400
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR003 Engine ; RA001E		6th
Driver	CAR No. 9	Olivier Panis (France)	14th
	CAR No. 10	Jacques Villeneuve (Canada)	7th
Test & Reserve Driver		Darren Manning (United Kingdom)	

			Ranking
B & H Jordan Honda	Chassis ; EJ11 Engine ; RA001E		5th
Driver	CAR No. 11	Heinz-Harald Frentzen (Germany)	13th
	CAR No. 12	Jarno Trulli (Italy)	9th
	CAR No. 12	Jean Alesi (France)	15th
Test & Reserve Driver	CAR No. 11	Ricardo Zonta (Brazil)	19th

Chassis, Engine (2002)

■BAR004

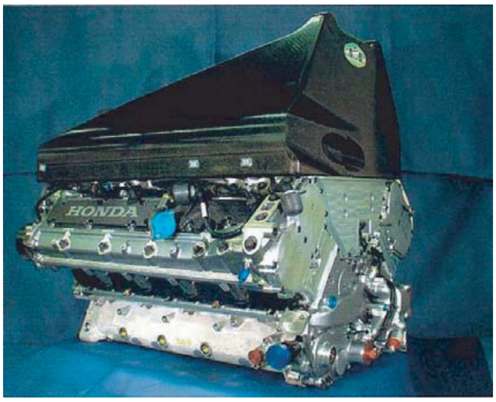
Chassis specifications (BAR004)	
Wheelbase (mm)	3133
Gravitational center hight (mm, above reference plane)	243*
Main monocoque weight (kg)	61
Tire supplier	Bridgestone
Tire size-front	265/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR004)	
Gear box case	Magnesium Casting
Changing mechanism	Xtrac
Forward gear ratios	6 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■EJ12

■RA002E



Engine specifications (RA002E)	
Engine Type	94° V10
Displacement (cm^3)	2994.4
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	12.2
Weight (kg)	111
Maximum power (kW)	640 Over
Maximum Ne (rpm)	18000 Over
Mileage (km)	400
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR004 Engine ; RA002E		8th
Driver	CAR No. 11	Jacques Villeneuve (Canada)	12th
	CAR No. 12	Olivier Panis (France)	14th
Test & Reserve Driver		Anthony Davidson (United Kingdom)	

			Ranking
DHL Jordan Honda	Chassis ; EJ12 Engine ; RA002E		6th
Driver	CAR No. 9	Giancarlo Fisichella (Italy)	11th
	CAR No. 10	Takuma Sato (Japan)	15th

Chassis, Engine (2003)

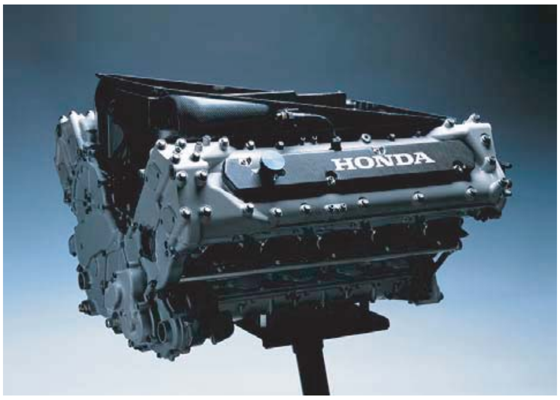
■BAR005

Chassis specifications (BAR005)	
Wheelbase (mm)	3194
Gravitational center hight (mm, above reference plane)	227*
Main monocoque weight (kg)	54
Tire supplier	Bridgestone
Tire size-front	265/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR005)	
Gear box case	Aluminum Casting
Changing mechanism	Xtrac
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■RA003E



Engine specifications (RA003E)	
Engine Type	90° V10
Displacement (cm³)	2994.4
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	12.4
Weight (kg)	99
Maximum power (kW)	662 Over
Maximum Ne (rpm)	18800 Over
Mileage (km)	400
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR005	Engine ; RA003E	5th
Driver	CAR No. 16	Jacques Villeneuve (Canada)	16th
	CAR No. 17	Jenson Button (United Kingdom)	9th
Reserve Driver	CAR No. 16	Takuma Sato (Japan)	18th
Test Driver		Anthony Davidson (United Kingdom)	

Chassis, Engine (2004)

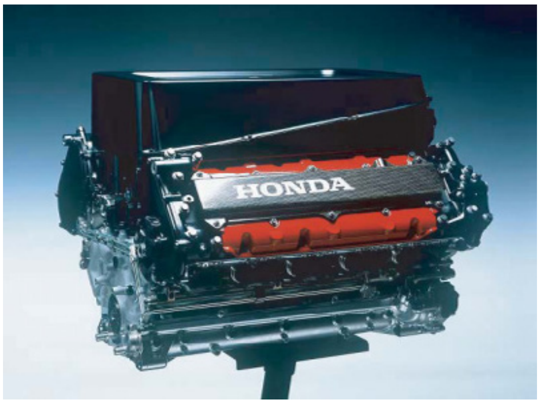
■BAR006

Chassis specifications (BAR006)	
Wheelbase (mm)	3131
Gravitational center hight (mm, above reference plane)	229*
Main monocoque weight (kg)	56
Tire supplier	Michelin
Tire size-front	270/66R13
Tire size-rear	320/66R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR006)	
Gear box case	CFRP
Changing mechanism	Honda
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■RA004E



Engine specifications (RA004E)	
Engine Type	90° V10
Displacement (cm^3)	2994.4
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	12.8
Weight (kg)	91
Maximum power (kW)	662 Over
Maximum Ne (rpm)	19000 Over
Mileage (km)	800
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR006 Engine ; RA004E		2nd
Driver	CAR No. 9	Jenson Button (United Kingdom)	3rd
	CAR No. 10	Takuma Sato (Japan)	8th
Test & Reserve Driver		Anthony Daivdson (United Kingdom)	

Chassis, Engine (2005)

■BAR007

Chassis specifications (BAR007)	
Wheelbase (mm)	3085
Gravitational center hight (mm, above reference plane)	230*
Main monocoque weight (kg)	56
Tire supplier	Michelin
Tire size-front	270/66R13
Tire size-rear	320/66R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Pi

*With driver, without fuel

Gear box specification (BAR007)	
Gear box case	CFRP
Changing mechanism	Honda (Seamledss shift)
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	Pull clutch

■RA005E

Engine specifications (RA005E)	
Engine Type	90° V10
Displacement (cm^3)	2994.4
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	12.8
Weight (kg)	89
Maximum power (kW)	662 Over
Maximum Ne (rpm)	19200 Over
Mileage (km)	1400
Engine electronic control system	Honda

			Ranking
Lucky Strike B.A.R Honda	Chassis ; BAR007 Engine ; RA005E		6th
Driver	CAR No. 3	Jenson Button (United Kingdom)	9th
	CAR No. 4	Takuma Sato (Japan)	23rd
Test & Reserve Driver	CAR No. 4	Anthony Davidson (United Kingdom)	

Chassis, Engine (2006)

■RA106

Chassis specifications (RA106)	
Wheelbase (mm)	until Rd.13 : 3085 as from Rd.14 : 3135
Gravitational center hight (mm, above reference plane)	231*
Main monocoque weight (kg)	57
Tire supplier	Michelin
Tire size-front	270/66R13
Tire size-rear	320/66R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Honda

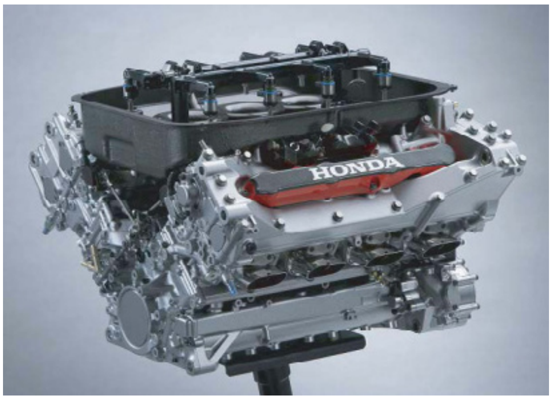
*With driver, without fuel

■SA05 (Rd.1-Rd.11)

Gear box specification (RA106)	
Gear box case	CFRP
Changing mechanism	Honda (Seamledss shift)
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	DPC

■SA06 (Rd.15-Rd.18)

■RA806E



Engine specifications (RA806E)	
Engine Type	90° V8
Displacement (cm^3)	2395.5
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	12.8
Weight (kg)	95
Maximum power (kW)	537 Over
Maximum Ne (rpm)	19600 Over
Mileage (km)	1400
Engine electronic control system	Honda

			Ranking
Lucky Strike Honda Racing F1 Team	Chassis ; RA106 Engine ; RA806E		4th
Driver	CAR No. 11	Jenson Button (United Kingdom)	6th
	CAR No. 12	Rubens Barrichello (Brazil)	7th
Test & Reserve Driver		Anthony Davidson (United Kingdom)	

			Ranking
Super Aguri Formula 1	Chassis ; SA06 Engine ; RA806E		11th
Driver	CAR No. 22	Takuma Sato (Japan)	23rd
	CAR No. 23	Yuji Ide (Japan)	25th
	CAR No. 23	Sakon Yamamoto (Japan)	26th
Test & Reserve Driver	CAR No. 23	Franck Montagny (France)	27th

Chassis, Engine (2007)

■RA107

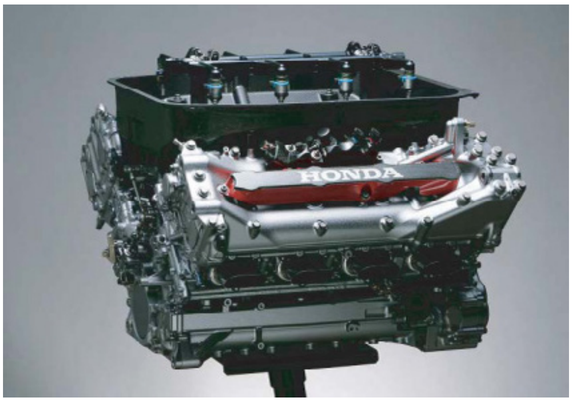
Chassis specifications (RA107)	
Wheelbase (mm)	3165
Gravitational center hight (mm, above reference plane)	235*
Main monocoque weight (kg)	56
Tire supplier	Bridgestone
Tire size-front	270/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Honda

*With driver, without fuel

■SA07

Gear box specification (RA107)	
Gear box case	CFRP
Changing mechanism	Honda (Seamledss shift)
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	DPC

■RA807E



Engine specifications (RA807E)	
Engine Type	90° V8
Displacement (cm^3)	2395.5
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	13
Weight (kg)	95
Maximum power (kW)	544 Over
Maximum Ne (rpm)	19000 Limit
Mileage (km)	1400
Engine electronic control system	Honda

			Ranking
Honda Racing F1 Team	Chassis ; RA107 Engine ; RA807E		8th
Driver	CAR No. 7	Jenson Button (United Kingdom)	15th
	CAR No. 8	Rubens Barrichello (Brazil)	20th
Test & Reserve Driver		Christian Klien (Austria)	

			Ranking
Super Aguri F1 Team	Chassis ; SA07 Engine ; RA807E		9th
Driver	CAR No. 12	Takuma Sato (Japan)	17th
	CAR No. 14	Anthony Davidson (United Kingdom)	23rd
Test & Reserve Driver		James Rossiter (United Kingdom)	

Chassis, Engine (2008)

■RA108

Chassis specifications (RA108)	
Wheelbase (mm)	3210
Gravitational center hight (mm, above reference plane)	243*
Main monocoque weight (kg)	55
Tire supplier	Bridgestone
Tire size-front	270/55R13
Tire size-rear	325/45R13
Brake caliper (front and rear)	6 pistons
Brake disk (front and rear)	Carbon, ventilated
Suspension-front	Double wishbone, push rod
Suspension-rear	Double wishbone, push rod
Chassis electronic control system	Microsoft MES

*With driver, without fuel

■SA08

Gear box specification (RA108)	
Gear box case	CFRP
Changing mechanism	Honda (Seamledss shift)
Forward gear ratios	7 speed, sequential
Clutch disk	Carbon
Clutch mechanism	DPC

■RA808E

Engine specifications (RA808E)	
Engine type	90° V8
Displacement (cm^3)	2395.5
Cylinder diameter (mm)	97
Stroke (mm)	40.52
Compression ratio	13
Weight (kg)	95
Maximum power (kW)	548 Over
Maximum Ne (rpm)	19000 Limit
Mileage (km)	1400
Engine electronic control system	Microsoft MES

			Ranking
Honda Racing F1 Team	Chassis ; RA108 Engine ; RA808E		9th
Driver	CAR No. 7	Jenson Button (United Kingdom)	18th
	CAR No. 8	Rubens Barrichello (Brazil)	14th
Test & Reserve Driver		Alex Wurz (Austria)	

			Ranking
Super Aguri F1 Team	Chassis ; SA08 Engine ; RA808E		11th
Driver	CAR No. 12	Takuma Sato (Japan)	21st
	CAR No. 14	Anthony Davidson (United Kingdom)	22nd
Test & Reserve Driver		James Rossiter (United Kingdom)	

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike BAR Honda		POLE	PODIUM
		#22 J. VILLENEUVE	#23 R. ZONTA	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	8th (1'31.968)	16th (1'33.117)	#1 M. HAKKINEN 1'30.556	#3 M. SCHUMACHER
Melbourne	RACE	4th	6th		#4 R. BARRICHELLO
5.302km×58Laps=307.574km	(DELAY)	(44.447)	(46.468)		#9 R. SCHUMACHER
ROUND 2 BRAZIL GP	QUALIFYING	10th (1'15.515)	8th (1'15.484)	#1 M. HAKKINEN 1'14.111	#3 M. SCHUMACHER
Interlagos	RACE	R (16LAPS)	9th		#11 G. FISICHELLA
4.325km×71Laps=305.909km	(DELAY)	-	(2LAPS)		#5 H. FRENTZEN
ROUND 3 SAN MARINO GP	QUALIFYING	9th (1'26.124)	14th (1'26.814)	#1 M. HAKKINEN 1'24.714	#3 M. SCHUMACHER
Imola	RACE	5th	12th		#1 M. HAKKINEN
4.693km×62Laps=305.609km	(DELAY)	(1LAP)	(1LAP)		#2 D. COULTHARD
ROUND 4 GREAT BRITAIN GP	QUALIFYING	10th (1'27.025)	16th (1'27.772)	#4 R. BARRICHELLO 1'25.703	#2 D. COULTHARD
Silverstone	RACE	16th	R (36LAPS)		#1 M. HAKKINEN
5.141km×60Laps=308.356km	(DELAY)	(DNF)	-		#3 M. SCHUMACHER
ROUND 5 SPAIN GP	QUALIFYING	6th (1'21.963)	16th (1'22.882)※1	#3 M. SCHUMACHER 1'20.974	#1 M. HAKKINEN
Barcelona	RACE	R (21LAPS)	8th		#2 D. COULTHARD
4.727km×65Laps=307.323km	(DELAY)	-	(1LAP)		#4 R. BARRICHELLO
ROUND 6 EUROPE GP	QUALIFYING	9th (1'18.742)	18th (1'19.766)※2	#2 D. COULTHARD 1'17.529	#3 M. SCHUMACHER
Nurburgring	RACE	R (46LAPS)	R (51LAPS)		#1 M. HAKKINEN
4.556km×67Laps=305.235km	(DELAY)	-	-		#2 D. COULTHARD
ROUND 7 MONACO GP	QUALIFYING	17th (1'21.848)	20th (1'22.324)	#3 M. SCHUMACHER 1'19.475	#2 D. COULTHARD
Monte Carlo	RACE	7th	R (48LAPS)		#4 R. BARRICHELLO
3.367km×78Laps=262.860km	(DELAY)	(1LAP)	-		#11 G. FISICHELLA
ROUND 8 CANADA GP	QUALIFYING	6th (1'19.544)	8th (1'19.742)	#3 M. SCHUMACHER 1'18.439	#3 M. SCHUMACHER
Montreal	RACE	15th	8th		#4 R. BARRICHELLO
4.421km×69Laps=305.049km	(DELAY)	(DNF)	-		#11 G. FISICHELLA
ROUND 9 FRANCE GP	QUALIFYING	7th (1'16.653)	19th (1'17.668)	#3 M. SCHUMACHER 1'15.632	#2 D. COULTHARD
Magny Cours	RACE	4th	R (16LAPS)		#1 M. HAKKINEN
4.250km×72Laps=305.886km	(DELAY)	(61.322)	-		#4 R. BARRICHELLO
ROUND 10 AUSTRIA GP	QUALIFYING	7th (1'11.649)	6th (1'11.647)	#1 M. HAKKINEN 1'10.410	#1 M. HAKKINEN
Spielberg	RACE	4th	R (58LAPS)		#2 D. COULTHARD
4.319km×71Laps=307.146km	(DELAY)	(1LAP)	-		#4 R. BARRICHELLO
ROUND 11 GERMANY GP	QUALIFYING	9th (1'48.121)	12th (1'48.665)	#2 D. COULTHARD 1'45.697	#4 R. BARRICHELLO
Hockenheim	RACE	8th	R (37LAPS)		#1 M. HAKKINEN
6.823km×45Laps=307.125km	(DELAY)	(47.537)	-		#2 D. COULTHARD
ROUND 12 HUNGARY GP	QUALIFYING	16th (1'19.937)	18th (1'20.272)	#3 M. SCHUMACHER 1'17.514	#1 M. HAKKINEN
Budapest	RACE	12th	14th		#3 M. SCHUMACHER
3.971km×77Laps=306.075km	(DELAY)	(2LAPS)	(2LAPS)		#2 D. COULTHARD
ROUND 13 BELGIUM GP	QUALIFYING	7th (1'51.799)	13th (1'53.002)	#1 M. HAKKINEN 1'50.646	#1 M. HAKKINEN
Spa Francorchamps	RACE	7th	12th		#3 M. SCHUMACHER
6.968km×44Laps=306.592km	(DELAY)	(72.380)	(1LAP)		#9 R. SCHUMACHER
ROUND 14 ITALY GP	QUALIFYING	4th (1'24.238)	17th (1'25.337)	#3 M. SCHUMACHER 1'23.770	#3 M. SCHUMACHER
Monza	RACE	R (14LAPS)	6th		#1 M. HAKKINEN
5.770km×53Laps=306.764km	(DELAY)	-	(69.292)		#9 R. SCHUMACHER
ROUND 15 UNITED STATES GP	QUALIFYING	8th (1'15.317)	12th (1'15.784)	#3 M. SCHUMACHER 1'14.266	#3 M. SCHUMACHER
Indianapolis	RACE	4th	6th		#4 R. BARRICHELLO
4.192km×73Laps=305.999km	(DELAY)	(17.935)	(51.694)		#5 H. FRENTZEN
ROUND 16 JAPAN GP	QUALIFYING	9th (1'37.267)	18th (1'38.269)	#3 M. SCHUMACHER 1'35.825	#3 M. SCHUMACHER
Suzuka	RACE	6th	9th		#1 M. HAKKINEN
5.864km×53Laps=310.596km	(DELAY)	(1LAP)	(1LAP)		#2 D. COULTHARD
ROUND 17 MALAYSIA GP	QUALIFYING	6th (1'38.653)	11th (1'39.158)	#3 M. SCHUMACHER 1'37.397	#3 M. SCHUMACHER
Kuala Lumpur	RACE	5th	R (46LAPS)		#2 D. COULTHARD
5.542km×56Laps=310.408km	(DELAY)	(70.692)	-		#4 R. BARRICHELLO

※1 Raised from 17th to 16th position after a fuel infringement by P. de la Rosa in 9th position caused his qualifying times to be deleted.
※2 Raised from 19th to 18th position after a minimum weight infringement by N. Heidfeld in 13th position resulted in his disqualification.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	BRA	SMR	GBR	SPA	EUR	MON	CAN	FRA	AUT	GER	HUN	BEL	ITA	USA	JPN	MAL
1	Scuderia Ferrari Marlboro	170	16	10	13	4	6	13	6	16	4	4	10	9	6	10	16	13	14
2	West McLaren Mercedes	152			10	16	16	10	11	3	16	6	10	14	13	6	2	10	9
3	BMW Williams F1 Team	36	4	3		5	3				2	2	3	2	6	4		2	
4	Mild Seven Benetton Playlife	20	2	6				2	4	4						2			
5	Lucky Strike BAR Honda	20	4		2						3	3				1	4	1	2
6	Benson & Hedges Jordan	17		7		1	1			1	1			1	1		4		
7	Arrows	7						1		2			1			3			
8	Red Bull Sauber Petronas	6			1				2			1	2						
9	Jaguar Racing	4							3										1
10	Telefonica Minardi Fondmetal	0																	
11	Gauloises Prost Peugeot	0																	

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	BRA	SMR	GBR	SPA	EUR	MON	CAN	FRA	AUT	GER	HUN	BEL	ITA	USA	JPN	MAL
1	3	M. SCHUMACHER (GER)	108	10	10	10	4	2	10		10				6	6	10	10	10	10
2	1	M. HAKKINEN (FIN)	89			6	6	10	6	1	3	6	10	6	10	10	6		6	3
3	2	D. COULTHARD (GBR)	73			4	10	6	4	10		10	6	4	4	3		2	4	6
4	4	R. BARRICHELLO (BRA)	62	6		3		4	3	6	6	4	4	10	3			6	3	4
5	9	R. SCHUMACHER (GER)	24	4	2		3	3				2			2	4	4			
6	11	G. FISICHELLA (ITA)	18	2	6				2	4	4									
7	22	J. VILLENEUVE (CAN)	17	3		2						3	3					3	1	2
8	10	J. BUTTON (GBR)	12		1		2						2	3		2			2	
9	5	H. FRENTZEN (GER)	11		4			1							1	1		4		
10	6	J. TRULLI (ITA)	6		3		1				1	1								
11	17	M. SALO (FIN)	6			1				2			1	2						
12	19	J. VERSTAPPEN (NED)	5								2						3			
13	7	E. IRVINE (GBR)	4							3										1
14	23	R. ZONTA (BRA)	3	1													1	1		
15	12	A. WURZ (AUT)	2														2			
16	18	P. DE LA ROSA (ESP)	2						1					1						
17	8	J. HERBERT (GBR)	0																	
18	16	P. DINIZ (BRA)	0																	
19	20	M. GENE (ESP)	0																	
20	15	N. HEIDFELD (GER)	0																	
21	21	G. MAZZACANE (ARG)	0																	
22	14	J. ALESI (FRA)	0																	
23	7	L. BURTI (BRA)	0																	

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike BAR Honda		B&H Jordan Honda		POLE	PODIUM
		#9 O. PANIS	#10 J. VILLENEUVE	#11 H. FRENTZEN	#12 J. TRULLI	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	9th (1'28.518)	8th (1'28.435)	4th (1'27.658)	7th (1'28.377)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Melbourne	RACE	7th※1	R (4LAPS)	5th※2	R (38LAPS)	1'26.892	#4 D. COULTHARD
5.303km×58Laps=307.574km	(DELAY)	(62.050)	-	(72.807)	-		#2 R. BARRICHELLO
ROUND 2 MALAYSIA GP	QUALIFYING	10th (1'36.681)	7th (1'36.397)	9th (1'36.578)	5th (1'36.180)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Kuala Lumpur	RACE	R (1LAP)	R (3LAPS)	4th	8th	1'35.220	#2 R. BARRICHELLO
5.543km×55Laps=304.865km	(DELAY)	-	-	(46.543)	(1LAP)		#4 D. COULTHARD
ROUND 3 BRAZIL GP	QUALIFYING	11th (1'15.046)	12th (1'15.182)	8th (1'14.633)	7th (1'14.630)	#1 M. SCHUMACHER	#4 D. COULTHARD
Interlagos	RACE	4th	7th	11th	5th	1'13.780	#1 M. SCHUMACHER
4.309km×71Laps=305.909km	(DELAY)	(1LAP)	(1LAP)	(DNF)	(1LAP)		#16 N. HEIDFELD
ROUND 4 SAN MARINO GP	QUALIFYING	8th (1'24.213)	11th (1'24.769)	9th (1'24.436)	5th (1'23.658)	#4 D. COULTHARD	#5 R. SCHUMACHER
Imola	RACE	8th	R (30LAPS)	6th	5th	1'23.054	#4 D. COULTHARD
4.933km×62Laps=305.609km	(DELAY)	(1LAP)	-	(1LAP)	(85.558)		#2 R. BARRICHELLO
ROUND 5 SPAIN GP	QUALIFYING	11th (1'19.479)	7th (1'19.122)	8th (1'19.150)	6th (1'19.093)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Barcelona	RACE	7th	3th	R (5LAPS)	4th	1'18.201	#6 J. MONTOYA
4.730km×65Laps=307.323km	(DELAY)	(64.977)	(49.626)	-	(51.253)		#10 J. VILLENEUVE
ROUND 6 AUSTRIA GP	QUALIFYING	10th (1'10.435)	12th (1'11.058)	11th (1'10.923)	5th (1'10.202)	#1 M. SCHUMACHER	#4 D. COULTHARD
Spielberg	RACE	5th	8th	R (0LAP)	D	1'09.562	#1 M. SCHUMACHER
4.326km×71Laps=307.146km	(DELAY)	(53.776)	(1LAP)	-	-		#2 R. BARRICHELLO
ROUND 7 MONACO GP	QUALIFYING	12th (1'19.294)	9th (1'19.086)	13th (1'19.316)	8th (1'18.921)	#4 D. COULTHARD	#1 M. SCHUMACHER
Monte Carlo	RACE	R (13LAPS)	4th	R (49LAPS)	R (30LAPS)	1'17.430	#2 R. BARRICHELLO
3.370km×78Laps=262.860km	(DELAY)	-	(32.454)	-	-		#18 E. IRVINE
				#11 R. ZONTA			
ROUND 8 CANADA GP	QUALIFYING	6th (1'16.771)	9th (1'17.035)	12th (1'17.328)	4th (1'16.459)	#1 M. SCHUMACHER	#5 R. SCHUMACHER
Montreal	RACE	R (38LAPS)	R (34LAPS)	7th	11th	1'15.782	#1 M. SCHUMACHER
4.421km×69Laps=305.049km	(DELAY)	-	-	(1LAP)	(DNF)		#3 M. HAKKINEN
				#11 H. FRENTZEN			
ROUND 9 EUROPE GP	QUALIFYING	13th (1'16.872)	11th (1'16.439)	8th (1'16.376)	7th (1'16.138)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Nurburgring	RACE	R (23LAPS)	9th	R (48LAPS)	R (44LAPS)	1'14.960	#6 J. MONTOYA
4.556km×67Laps=305.235km	(DELAY)	-	(1LAP)	-	-		#4 D. COULTHARD
ROUND 10 FRANCE GP	QUALIFYING	11th (1'14.181)	10th (1'14.096)	7th (1'13.815)	5th (1'13.310)	#5 R. SCHUMACHER	#1 M. SCHUMACHER
Magny Cours	RACE	9th	R (5LAPS)	8th	5th	1'12.989	#5 R. SCHUMACHER
4.251km×72Laps=305.886km	(DELAY)	(1LAP)	-	(1LAP)	(68.285)		#2 R. BARRICHELLO
ROUND 11 GREAT BRITAIN GP	QUALIFYING	11th (1'22.316)	12th (1'22.916)	5th (1'21.217)	4th (1'20.930)	#1 M. SCHUMACHER	#3 M. HAKKINEN
Silverstone	RACE	R (0LAP)	8th	7th	R (0LAP)	1'20.447	#1 M. SCHUMACHER
5.141km×60Laps=308.356km	(DELAY)	-	(1LAP)	(1LAP)	-		#2 R. BARRICHELLO
				#11 R. ZONTA			
ROUND 12 GERMANY GP	QUALIFYING	13th (1'40.610)	12th (1'40.437)	15th (1'41.174)	10th (1'40.322)	#6 J. MONTOYA	#5 R. SCHUMACHER
Hockenheim	RACE	7th	3th	R (7LAPS)	R (34LAPS)	1'38.117	#2 R. BARRICHELLO
6.825km×45Laps=307.125km	(DELAY)	(77.527)	(62.806)	-	-		#10 J. VILLENEUVE
				#11 J. TRULLI	#12 J. ALESI		
ROUND 13 HUNGARY GP	QUALIFYING	11th (1'16.382)	10th (1'16.212)	5th (1'15.394)	12th (1'16.471)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Budapest	RACE	R (58LAPS)	9th	R (63LAPS)	10th	1'14.059	#2 R. BARRICHELLO
3.975km×77Laps=306.075km	(DELAY)	-	(2LAPS)	-	(2LAPS)		#4 D. COULTHARD
ROUND 14 BELGIUM GP	QUALIFYING	11th (1'58.838)	6th (1'57.038)	16th (1'59.647)	13th (1'59.128)	#6 J. MONTOYA	#1 M. SCHUMACHER
Spa Francorchamps	RACE	11th	8th	R (31LAPS)	6th	1'52.072	#4 D. COULTHARD
6.968km×36Laps=250.831km	(DELAY)	(1LAP)	(64.970)	-	(59.684)		#7 G. FISICHELLA
ROUND 15 ITALY GP	QUALIFYING	17th (1'24.677)	15th (1'24.164)	5th (1'23.126)	16th (1'24.198)	#6 J. MONTOYA	#6 J. MONTOYA
Monza	RACE	9th	6th	R (0LAP)	8th	1'22.216	#2 R. BARRICHELLO
5.793km×53Laps=306.749km	(DELAY)	(1LAP)	(82.469)	-	(1LAP)		#5 R. SCHUMACHER
ROUND 16 UNITED STATES GP	QUALIFYING	13th (1'13.122)	18th (1'14.012)	8th (1'12.605)	9th (1'12.607)	#1 M. SCHUMACHER	#3 M. HAKKINEN
Indianapolis	RACE	11th	R (45LAPS)	4th	7th	1'11.708	#1 M. SCHUMACHER
4.192km×73Laps=306.016km	(DELAY)	(1LAP)	-	(57.423)	(1LAP)		#4 D. COULTHARD
ROUND 17 JAPAN GP	QUALIFYING	17th (1'35.766)	14th (1'35.109)	8th (1'34.002)	11th (1'34.420)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Suzuka	RACE	13th	10th	8th	R (5LAPS)	1'32.484	#6 J. MONTOYA
5.859km×53Laps=310.331km	(DELAY)	(2LAPS)	(1LAP)	(1LAP)	-		#4 D. COULTHARD

※1 O. Panis finished in 4th position, but 25 seconds was added to his racing time for overtaking under a yellow flag, and he was demoted to 7th position.

※2 H. Frentzen finished in 6th position, but was raised to 5th position after 25 seconds was added to O. Panis' racing time for overtaking under a yellow flag.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRA	SMR	ESP	AUT	MON	CAN	EUR	FRA	GBR	GER	HUN	BEL	ITA	USA	JPN
1	Scuderia Ferrari Marlboro	179	14	16	6	4	10	10	16	6	12	14	10	6	16	12	9	6	12
2	West McLaren Mercedes	102	6	5	10	9	2	10	2	4	5	3	10		6	9		14	7
3	BMW Williams F1 Team	80		2		10	6			10	9	6	3	10	3		14		7
4	Red Bull Sauber Petronas	21	4		4		1	3		3		1	3		1			1	
5	B&H Jordan Honda	19	2	3	2	3	3					2				1		3	
6	Lucky Strike BAR Honda	17			3		4	2	3					4			1		
7	Mild Seven Benetton Renault	10			1									5		4			
8	Jaguar Racing	9							4	1							2	2	
9	Prost Acer	4							1	2				1					
10	Orange Arrows Asiatech	1						1											
11	European Minardi F1	0																	

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRA	SMR	ESP	AUT	MON	CAN	EUR	FRA	GBR	GER	HUN	BEL	ITA	USA	JPN
1	1	M. SCHUMACHER (GER)	123	10	10	6		10	6	10	6	10	10	6		10	10	3	6	10
2	4	D. COULTHARD (GBR)	65	6	4	10	6	2	10	2		4	3			4	6		4	4
3	2	R. BARRICHELLO (BRA)	56	4	6		4		4	6		2	4	4	6	6	2	6		2
4	5	R. SCHUMACHER (GER)	49		2		10				10	3	6		10	3		4		1
5	3	M. HAKKINEN (FIN)	37		1		3				4	1		10		2	3		10	3
6	6	J. MONTOYA (COL)	31					6				6		3				10		6
7	10	J. VILLENEUVE (CAN)	12					4		3					4			1		
8	16	N. HEIDFELD (GER)	12	3		4		1					1	1		1			1	
9	11	J. TRULLI (ITA)	12			2	2	3					2						3	
10	17	K. RAIKKONEN (FIN)	9	1					3		3			2						
11	7	G. FISICHELLA (ITA)	8			1									3		4			
12	18	E. IRVINE (GBR)	6							4									2	
13	22	H. FRENTZEN (GER)	6	2	3		1													
14	9	O. PANIS (FRA)	5			3			2											
15	12	J. ALESI (FRA)	5							1	2				1		1			
16	19	P. DELA ROSA (ESP)	3								1							2		
17	8	J. BUTTON (GBR)	2												2					
18	14	J. VERSTAPPEN (NED)	1						1											
19	11	R. ZONTA (BRA)	0																	
20	19	L. BURTI (BRA)	0																	
21	15	E. BERNOLDI (BRA)	0																	
22	20	T. MARQUES (BRA)	0																	
23	21	F. ALONSO (ESP)	0																	
24	23	T. ENGE (CZE)	0																	
25	23	G. MAZZACANE (ARG)	0																	
26	20	A. YOONG (MAS)	0																	

FIA FORMULA ONE WORLD CHAMPIONSHIP

		DHL Jordan Honda		Lucky Strike BAR Honda		POLE	PODIUM
		#9 G. FISICHELLA	#10 T. SATO	#11 J. VILLENEUVE	#12 O. PANIS	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	8th (1'27.869)	(1'53.351)	13th (1'28.657)	12th (1'28.381)	#2 R. BARRICHELLO	#1 M. SCHUMACHER
Melbourne	RACE	R (0LAP)	R (12LAPS)	R (27LAPS)	R (0LAP)	1'25.843	#6 J. MONTOYA
5.303km×58Laps=307.574km	(DELAY)	-	-	-	-		#4 K. RAIKKONEN
ROUND 2 MALAYSIA GP	QUALIFYING	9th (1'37.536)	15th (1'38.141)	13th (1'38.039)	18th (1'38.390)	#1 M. SCHUMACHER	#5 R. SCHUMACHER
Kuala Lumpur	RACE	13th	9th	8th	R (9LAPS)	1'35.266	#6 J. MONTOYA
5.543km×56Laps=310.408km	(DELAY)	(3LAPS)	(2LAPS)	(1LAP)	-		#1 M. SCHUMACHER
ROUND 3 BRAZIL GP	QUALIFYING	14th (1'14.748)	19th (1'15.296)	15th (1'14.760)	17th (1'14.996)	#6 J. MONTOYA	#1 M. SCHUMACHER
Interlagos	RACE	R (6LAPS)	9th	10th	R (25LAPS)	1'13.114	#5 R. SCHUMACHER
4.309km×71Laps=305.909km	(DELAY)	-	(2LAPS)	(DNF)	-		#3 D. COULTHARD
ROUND 4 SAN MARINO GP	QUALIFYING	15th (1'24.253)	14th (1'24.050)	10th (1'23.116)	12th (1'23.821)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Imola	RACE	R (19LAPS)	R (5LAPS)	7th	R (44LAPS)	1'21.091	#2 R. BARRICHELLO
4.933km×62Laps=305.609km	(DELAY)	-	-	(1LAP)	-		#5 R. SCHUMACHER
ROUND 5 SPAIN GP	QUALIFYING	12th (1'18.291)	18th (1'19.002)※1	15th (1'18.847)	13th (1'18.472)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Barcelona	RACE	R (5LAPS)	R (10LAPS)	7th	R (43LAPS)	1'16.364	#6 J. MONTOYA
4.730km×65Laps=307.327km	(DELAY)	-	-	(1LAP)	-		#3 D. COULTHARD
ROUND 6 AUSTRIA GP	QUALIFYING	15th (1'09.901)	18th (1'10.058)	17th (1'10.051)	9th (1'09.561)	#2 R. BARRICHELLO	#1 M. SCHUMACHER
Spielberg	RACE	5th	R (26LAPS)	10th	R (22LAPS)	1'08.082	#2 R. BARRICHELLO
4.326km×71Laps=307.146km	(DELAY)	(49.965)	-	(1LAP)	-		#6 J. MONTOYA
ROUND 7 MONACO GP	QUALIFYING	11th (1'18.342)	16th (1'19.461)	14th (1'19.252)	18th (1'19.569)	#6 J. MONTOYA	#3 D. COULTHARD
Monte Carlo	RACE	5th	R (22LAPS)	R (44LAPS)	R (51LAPS)	1'16.676	#1 M. SCHUMACHER
3.370km×78Laps=262.860km	(DELAY)	(1LAP)	-	-	-		#5 R. SCHUMACHER
ROUND 8 CANADA GP	QUALIFYING	6th (1'14.132)	15th (1'14.940)	9th (1'14.564)	11th (1'14.713)	#6 J. MONTOYA	#1 M. SCHUMACHER
Montreal	RACE	5th	10th	R (8LAPS)	8th	1'12.836	#3 D. COULTHARD
4.361km×70Laps=305.270km	(DELAY)	(42.812)	(1LAP)	-	(1LAP)		#2 R. BARRICHELLO
ROUND 9 EUROPE GP	QUALIFYING	18th (1'32.591)	14th (1'31.999)	19th (1'32.968)	12th (1'31.906)	#6 J. MONTOYA	#2 R. BARRICHELLO
Nurburgring	RACE	R (26LAPS)	16th	12th	9th	1'29.906	#1 M. SCHUMACHER
5.146km×60Laps=308.743km	(DELAY)	-	(2LAPS)	(1LAP)	(1LAP)		#4 K. RAIKKONEN
ROUND 10 GREAT BRITAIN GP	QUALIFYING	17th (1'21.636)	14th (1'21.337)	9th (1'21.130)	13th (1'21.274)	#6 J. MONTOYA	#1 M. SCHUMACHER
Silverstone	RACE	7th	R (50LAPS)	4th	5th	1'18.998	#2 R. BARRICHELLO
5.141km×60Laps=308.356km	(DELAY)	(1LAP)	-	(1LAP)	(1LAP)		#6 J. MONTOYA
ROUND 11 FRANCE GP	QUALIFYING	DNQ	14th (1'13.542)	13th (1'13.506)	11th (1'13.457)	#6 J. MONTOYA	#1 M. SCHUMACHER
Magny Cours	RACE	-	R (23LAPS)	R (35LAPS)	R (29LAPS)	1'11.985	#4 K. RAIKKONEN
4.251km×72Laps=305.886km	(DELAY)	-	-	-	-		#3 D. COULTHARD
ROUND 12 GERMANY GP	QUALIFYING	6th (1'15.690)	12th (1'16.072)	11th (1'16.070)	7th (1'15.851)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Hockenheim	RACE	R (59LAPS)	8th	R (27LAPS)	R (39LAPS)	1'14.389	#6 J. MONTOYA
4.574km×67Laps=306.458km	(DELAY)	-	(1LAP)	-	-		#5 R. SCHUMACHER
ROUND 13 HUNGARY GP	QUALIFYING	5th (1'14.880)	14th (1'15.804)	13th (1'15.583)	12th (1'15.556)	#2 R. BARRICHELLO	#2 R. BARRICHELLO
Budapest	RACE	6th	10th	R (20LAPS)	12th	1'13.333	#1 M. SCHUMACHER
3.975km×77Laps=306.069km	(DELAY)	(68.804)	(1LAP)	-	(1LAP)		#5 R. SCHUMACHER
ROUND 14 BELGIUM GP	QUALIFYING	14th (1'46.508)	16th (1'46.875)	12th (1'46.403)	15th (1'46.553)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Spa Francorchamps	RACE	R (38LAPS)	11th	8th	12th	1'43.726	#2 R. BARRICHELLO
6.963km×44Laps=306.355km	(DELAY)	-	(1LAP)	(79.855)	(DNF)		#6 J. MONTOYA
ROUND 15 ITALY GP	QUALIFYING	12th (1'22.515)	18th (1'23.166)	9th (1'22.126)	16th (1'22.645)	#6 J. MONTOYA	#2 R. BARRICHELLO
Monza	RACE	8th	12th	9th	6th	1'20.264	#1 M. SCHUMACHER
5.793km×53Laps=306.719km	(DELAY)	(70.891)	(1LAP)	(81.068)	(68.491)		#16 E. IRVINE
ROUND 16 UNITED STATES GP	QUALIFYING	9th (1'11.902)	15th (1'12.647)	7th (1'11.738)	12th (1'12.161)	#1 M. SCHUMACHER	#2 R. BARRICHELLO
Indianapolis	RACE	7th	11th	6th	12th	1'10.790	#1 M. SCHUMACHER
4.192km×73Laps=306.016km	(DELAY)	(1LAP)	(1LAP)	(58.211)	(1LAP)		#3 D. COULTHARD
ROUND 17 JAPAN GP	QUALIFYING	8th (1'33.276)	7th (1'33.090)	9th (1'33.349)	16th (1'34.192)	#1 M. SCHUMACHER	#1 M. SCHUMACHER
Suzuka	RACE	R (37LAPS)	5th	R (27LAPS)	R (8LAPS)	1'31.317	#2 R. BARRICHELLO
5.821km×53Laps=308.317km	(DELAY)	-	(82.694)	-	-		#4 K. RAIKKONEN

※1 Raised from 19^{th} to 18^{th} position after a fuel infringement by E. Irvine resulted in his disqualification.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRA	SMR	ESP	AUT	MON	CAN	EUR	GBR	FRA	GER	HUN	BEL	ITA	USA	JPN
1	Scuderia Ferrari Marlboro	221	10	4	10	16	10	16	6	14	16	16	10	13	16	16	16	16	16
2	BMW Williams F1 Team	92	6	16	8	7	6	7	4		3	4	5	10	4	6		3	3
3	West McLaren Mercedes	65	4		4	1	4	1	10	9	4		10	2	5	3		4	4
4	Mild Seven Renault F1 Team	23		3	3	2			3	1	2		1				5	2	1
5	Sauber Petronas	11		3			5				1	1		1					
6	DHL Jordan Honda	9						2	2	2					1				2
7	Jaguar Racing	8	3													1	4		
8	Lucky Strike BAR Honda	7										5					1	1	
9	KL Minardi Asiatech	2	2																
10	Panasonic Toyota Racing	2	1		1														
11	Orange Arrows	2					1		1										

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRA	SMR	ESP	AUT	MON	CAN	EUR	GBR	FRA	GER	HUN	BEL	ITA	USA	JPN
1	1	M. SCHUMACHER (GER)	144	10	4	10	10	10	10	6	10	6	10	10	10	6	10	6	6	10
2	2	R. BARRICHELLO (BRA)	77				6		6		4	10	6		3	10	6	10	10	6
3	6	J. MONTOYA (COL)	50	6	6	2	3	6	4				4	3	6		4		3	3
4	5	R. SCHUMACHER (GER)	42		10	6	4		3	4		3		2	4	4	2			
5	3	D. COULTHARD (GBR)	41			4	1	4	1	10	6			4	2	2	3		4	
6	4	K. RAIKKONEN (FIN)	24	4							3	4		6		3				4
7	15	J. BUTTON (GBR)	14		3	3	2					2		1				2		1
8	14	J. TRULLI (ITA)	9							3	1							3	2	
9	16	E. IRVINE (GBR)	8	3													1	4		
10	7	N. HEIDFELD (GER)	7		2			3					1		1					
11	9	G. FISICHELLA (ITA)	7						2	2	2					1				
12	11	J. VILLENEUVE (CAN)	4										3						1	
13	8	F. MASSA (BRA)	4		1			2				1								
14	12	O. PANIS (FRA)	3										2					1		
15	10	T. SATO (JPN)	2																	2
16	23	M. WEBBER (AUS)	2	2																
17	24	M. SALO (FIN)	2	1		1														
18	20	H. FRENTZEN (GER)	2					1		1										
19	25	A. McNISH (GBR)	0																	
20	22	A. YOONG (MAS)	0																	
21	17	P. DE LA ROSA (ESP)	0																	
22	21	E. BERNOLDI (BRA)	0																	
23	22	A. DAVIDSON (GBR)	0																	

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike BAR Honda		POLE	PODIUM
		#16 J. VILLENEUVE	#17 J. BUTTON	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	6th (1'28.420)	8th (1'28.682)	#1 M. SCHUMACHER 1'27.173	#5 D. COULTHARD
Melbourne	RACE	9th	10th		#3 J. MONTOYA
5.303km×58Laps=307.574km	(DELAY)	(65.536)	(65.974)		#6 K. RAIKKONEN
ROUND 2 MALAYSIA GP	QUALIFYING	12th (1'38.289)	9th (1'38.073)	#8 F. ALONSO 1'37.044	#6 K. RAIKKONEN
Kuala Lumpur	RACE	R (0LAP)	7th		#2 R. BARRICHELLO
5.543km×56Laps=310.408km	(DELAY)	-	(1LAP)		#8 F. ALONSO
ROUND 3 BRAZIL GP	QUALIFYING	13th (1'14.668)	11th (1'14.504)	#2 R. BARRICHELLO 1'13.807	#11 G. FISICHELLA※1
Interlagos	RACE※1	6th	R (32LAPS)		#6 K. RAIKKONEN
4.309km×54Laps=232.656km	(DELAY)	(17.910)	-		#8 F. ALONSO
ROUND 4 SAN MARINO GP	QUALIFYING	7th (1'23.160)	9th (1'23.381)	#1 M. SCHUMACHER 1'22.327	#1 M. SCHUMACHER
Imola	RACE	R (19LAPS)	8th		#6 K. RAIKKONEN
4.993km×62Laps=305.609km	(DELAY)	-	(1LAP)		#2 R. BARRICHELLO
ROUND 5 SPAIN GP	QUALIFYING	11th (1'19.563)	5th (1'18.704)	#1 M. SCHUMACHER 1'17.762	#1 M. SCHUMACHER
Barcelona	RACE	R (12LAPS)	9th		#8 F. ALONSO
4.730km×65Laps=307.324km	(DELAY)	-	(2LAPS)		#2 R. BARRICHELLO
ROUND 6 AUSTRIA GP	QUALIFYING	12th (1'10.618)	7th (1'09.935)	#1 M. SCHUMACHER 1'09.150	#1 M. SCHUMACHER
Spielberg	RACE	12th	4th		#6 K. RAIKKONEN
4.326km×69Laps=298.494km	(DELAY)	(1LAP)	(42.243)		#2 R. BARRICHELLO
ROUND 7 MONACO GP	QUALIFYING	11th (1'16.755)	- ※2	#4 R. SCHUMACHER 1'15.259	#3.J. MONTOYA
Monte Carlo	RACE	R (63LAPS)	-		#6 K. RAIKKONEN
3.340km×78Laps=260.520km	(DELAY)	-	-		#1 M. SCHUMACHER
ROUND 8 CANADA GP	QUALIFYING	14th (1'17.347)	17th (1'18.205)	#4 R. SCHUMACHER 1'15.529	#1 M. SCHUMACHER
Montreal	RACE	R (14LAPS)	R (51LAPS)		#4 R. SCHUMACHER
4.361km×70Laps=305.270km	(DELAY)	-	-		#3. J. MONTOYA
ROUND 9 EUROPE GP	QUALIFYING	17th (1'34.596)	12th (1'33.395)	#6 K. RAIKKONEN 1'31.523	#4 R. SCHUMACHER
Nurburgring	RACE	R (51LAPS)	7th		#3 J. MONTOYA
5.148km×60Laps=308.863km	(DELAY)	-	(1LAP)		#2 R. BARRICHELLO
ROUND 10 FRANCE GP	QUALIFYING	12th (1'16.990)	14th (1'17.077)	#4 R. SCHUMACHER 1'15.019	#4 R. SCHUMACHER
Magny Cours	RACE	9th	R (21LAPS)		#3. J. MONTOYA
4.411km×70Laps=308.586km	(DELAY)	(1LAP)	-		#1 M. SCHUMACHER
ROUND 11 GREAT BRITAIN GP	QUALIFYING	9th (1'22.591)	20th (-)	#2 R. BARRICHELLO 1'21.209	#2 R. BARRICHELLO
Silverstone	RACE	10th	8th		#3 J. MONTOYA
5.141km×60Laps=308.355km	(DELAY)	(63.569)	(45.478)		#6 K. RAIKKONEN
ROUND 12 GERMANY GP	QUALIFYING	13th (1'17.090)	17th (1'18.085)	#3 J. MONTOYA 1'15.167	#3 J. MONTOYA
Hockenheim	RACE	9th	8th		#5 D. COULTHARD
4.574km×67Laps=306.458km	(DELAY)	(2LAPS)	(1LAP)		#7 J. TRULLI
ROUND 13 HUNGARY GP	QUALIFYING	16th (1'24.100)	14th (1'23.847)	#8 F. ALONSO 1'21.688	#8 F. ALONSO
Budapest	RACE	R (14LAPS)	10th		#6 K. RAIKKONEN
4.381km×70Laps=306.663km	(DELAY)	-	(1LAPS)		#3 J. MONTOYA
ROUND 14 ITALY GP	QUALIFYING	10th (1'22.717)	7th (1'22.301)	#1 M. SCHUMACHER 1'20.963	#1 M. SCHUMACHER
Monza	RACE	6th	R (24LAPS)		#3 J. MONTOYA
5.793km×53Laps=306.720km	(DELAY)	(1LAP)	-		#2 R. BARRICHELLO
ROUND 15 UNITED STATES GP	QUALIFYING	12th (1'13.050)	11th (1'12.695)	#6 K. RAIKKONEN 1'11.670	#1 M. SCHUMACHER
Indianapolis	RACE	R (63LAPS)	R (41LAPS)		#6 K. RAIKKONEN
4.192km×73Laps=306.016km	(DELAY)	-	-		#10 H. FRENTZEN
		#16 T. SATO			
ROUND 16 JAPAN GP	QUALIFYING	13th (1'33.924)	9th (1'33.474)	#2 R. BARRICHELLO 1'31.713	#2 R. BARRICHELLO
Suzuka	RACE	6th	4th		#6 K. RAIKKONEN
5.807km×53Laps=307.573km	(DELAY)	(51.692)	(33.106)		#5 D. COULTHARD

※ For 2003, QF2 times are recorded

※1 The red flag was displayed during lap 56, and the race was ended. Because more than 75% of the race had been completed, times at lap 54, two laps before the red flag was displayed, were used as the official times.

※2 J. Button was involved in an accident during the Saturday morning free practice session, and was ordered not to participate in the race by a doctor.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRA	RSM	ESP	AUT	MON	CAN	EUR	FRA	GBR	GER	HUN	ITA	USA	JPN
1	Scuderia Ferrari Marlboro	158	5	11		16	16	16	7	14	10	8	15	2	1	16	10	11
2	BMW Williams F1 Team	144	9	5	2	7	9	3	15	14	18	18	8	10	11	12	3	
3	West McLaren Mercedes	142	16	10	13	12		12	10	3		9	10	8	12	5	8	14
4	Mild Seven Renault F1 Team	88	6	10	7	3	8	1	7	5	5		3	11	12	1	5	4
5	Lucky Strike BAR Honda	26		2	3	1		5			2		1	1		3		8
6	Sauber Petronas	19	3	1	4						1						10	
7	Jaguar Racing	18					2	2		2	3	3			3	2	1	
8	Panasonic Toyota Racing	16					3			1		1	2	7				2
9	Jordan Ford	13			10		1										2	
10	European Minardi Cosworth	0																

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRA	RSM	ESP	AUT	MON	CAN	EUR	FRA	GBR	GER	HUN	ITA	USA	JPN
1	1	M. SCHUMACHER (GER)	93	5	3		10	10	10	6	10	4	6	5	2	1	10	10	1
2	6	K. RAIKKONEN (FIN)	91	6	10	8	8		8	8	3		5	6		8	5	8	8
3	3	J. MONTOYA (COL)	82	8			2	5		10	6	8	8	8	10	6	8	3	
4	2	R. BARRICHELLO (BRA)	65		8		6	6	6	1	4	6	2	10			6		10
5	4	R. SCHUMACHER (GER)	58	1	5	2	5	4	3	5	8	10	10			5			
6	8	F. ALONSO (ESP)	55	2	6	6	3	8		4	5	5			5	10	1		
7	5	D. COULTHARD (GBR)	51	10		5	4		4	2			4	4	8	4			6
8	7	J. TRULLI (ITA)	33	4	4	1			1	3				3	6	2		5	4
9	17	J. BUTTON (GBR)	17		2		1		5			2		1	1				5
10	14	M. WEBBER (AUS)	17					2	2		2	3	3			3	2		
11	10	H. FRENTZEN (GER)	13	3		4												6	
12	11	G. FISICHELLA (ITA)	12			10												2	
13	21	C. DA MATTA (BRA)	10					3						2	3				2
14	9	N. HEIDFELD (GER)	6		1							1						4	
15	20	O. PANIS (FRA)	6								1		1		4				
16	16	J. VILLENEUVE (CAN)	6			3											3		
17	4	M. GENE (ESP)	4														4		
18	16	T.SATO (JPN)	3																3
19	12	R. FIRMAN (IRL)	1					1											
20	15	J. WILSON (GBR)	1															1	
21	15	A. PIZZONIA (BRA)	0																
22	19	J. VERSTAPPEN (NED)	0																
23	18	N. KIESA (DEN)	0																
24	12	Z. BAUMGARTNER (HUN)	0																

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike BAR Honda		POLE	PODIUM
		#9 J. BUTTON	#10 T. SATO	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	4th (1'24.998)	7th (1'25.851)	#1 M. SCHUMACHER 1'24.408	#1 M. SCHUMACHER
Melbourne	RACE	6th	9th		#2 R. BARRICHELLO
5.303km×58Laps=307.574km	(DELAY)	(70.598)	(1LAP)		#8 F. ALONSO
ROUND 2 MALAYSIA GP	QUALIFYING	6th (1'34.221)	20th (-)	#1 M. SCHUMACHER 1'33.074	#1 M. SCHUMACHER
Kuala Lumpur	RACE	3th	15th		#3 J. MONTOYA
5.543km×56Laps=310.408km	(DELAY)	(11.568)	(DNF)		#9 J. BUTTON
ROUND 3 BAHRAIN GP	QUALIFYING	6th (1'30.856)	5th (1'30.827)	#1 M. SCHUMACHER 1'30.139	#1 M. SCHUMACHER
Bahrain	RACE	3th	5th		#2 R. BARRICHELLO
5.417km×57Laps=308.523km	(DELAY)	(26.687)	(52.460)		#9 J. BUTTON
ROUND 4 SAN MARINO GP	QUALIFYING	PP (1'19.753)	7th (1'20.913)	#9 J. BUTTON 1'19.753	#1 M. SCHUMACHER
Imola	RACE	2th	16th		#9 J. BUTTON
4.933km×62Laps=305.609km	(DELAY)	(9.702)	(DNF)		#3 J. MONTOYA
ROUND 5 SPAIN GP	QUALIFYING	14th (1'17.575)	3th (1'15.809)	#1 M. SCHUMACHER 1'15.022	#1 M. SCHUMACHER
Barcelona	RACE	8th	5th		#2 R. BARRICHELLO
4.627km×66Laps=305.256km	(DELAY)	(1LAP)	(42.327)		#7 J. TRULLI
ROUND 6 MONACO GP	QUALIFYING	3th (1'14.396)※1	8th (1'14.827)※1	#7 J. TRULLI 1'13.985	#7 J. TRULLI
Monte Carlo	RACE	2th	R (2LAPS)		#9 J. BUTTON
3.340km×77Laps=257.180km	(DELAY)	(0.497)	-		#2 R. BARRICHELLO
ROUND 7 EUROPE GP	QUALIFYING	5th (1'29.245)	2th (1'28.986)	#1 M. SCHUMACHER 1'28.351	#1 M. SCHUMACHER
Nurburgring	RACE	3th	R (47LAPS)		#2 R. BARRICHELLO
5.148km×60Laps=308.863km	(DELAY)	(22.533)	-		#9 J. BUTTON
ROUND 8 CANADA GP	QUALIFYING	2th (1'12.341)	17th (1'17.004)	#4 R. SCHUMACHER 1'12.275	#1 M. SCHUMACHER
Montreal	RACE	3th※2	R (48LAPS)		#2 R. BARRICHELLO
4.361km×70Laps=305.270km	(DELAY)	(20.409)	-		#9 J. BUTTON※2
ROUND 9 UNITED STATES GP	QUALIFYING	4th (1'10.820)	3th (1'10.601)	#2 R. BARRICHELLO 1'10.223	#1 M. SCHUMACHER
Indianapolis	RACE	R (26LAPS)	3th		#2 R. BARRICHELLO
4.192km×73Laps=306.016km	(DELAY)	-	(22.036)		#10 T. SATO
ROUND 10 FRANCE GP	QUALIFYING	4th (1'13.995)	7th (1'14.240)	#8. F. ALONSO 1'13.698	#1 M. SCHUMACHER
Magny Cours	RACE	5th	R (15LAPS)		#8. F. ALONSO
4.411km×70Laps=308.586km	(DELAY)	(32.484)	-		#2 R. BARRICHELLO
ROUND 11 GREAT BRITAIN GP	QUALIFYING	3th (1'18.580)	9th (1'19.688)※3	#6 K.RAIKKONEN 1'18.233	#1 M. SCHUMACHER
Silverstone	RACE	4th	11th		#6 K.RAIKKONEN
5.141km×60Laps=308.355km	(DELAY)	(10.683)	(33.736)		#2 R. BARRICHELLO
ROUND 12 GERMANY GP	QUALIFYING	3th (1'13.674)※4	9th (1'14.287)※4	#1 M. SCHUMACHER 1'13.306	#1 M. SCHUMACHER
Hockenheim	RACE	2th	8th		#9 J. BUTTON
4.574km×66Laps=301.884km	(DELAY)	(8.388)	(46.842)		#8 F. ALONSO
ROUND 13 HUNGARY GP	QUALIFYING	4th (1'19.700)	3th (1'19.693)	#1 M. SCHUMACHER 1'19.146	#1 M. SCHUMACHER
Budapest	RACE	5th	6th		#2 R. BARRICHELLO
4.381km×70Laps=306.663km	(DELAY)	(67.439)	(1LAP)		#8 F. ALONSO
ROUND 14 BELGIUM GP	QUALIFYING	12th (2'00.237)	15th (2'01.813)	#7 J. TRULLI 1'56.232	#6 K.RAIKKONEN
Spa Francorchamps	RACE	R (29LAPS)	R (0LAP)		#1 M. SCHUMACHER
6.976km×44Laps=306.927km	(DELAY)	-	-		#2 R. BARRICHELLO
ROUND 15 ITALY GP	QUALIFYING	6th (1'20.786)	5th (1'20.715)	#2 R. BARRICHELLO 1'20.089	#2 R. BARRICHELLO
Monza	RACE	3th	4th		#1 M. SCHUMACHER
5.793km×53Laps=306.720km	(DELAY)	(10.197)	(15.370)		#9 J. BUTTON
ROUND 16 CHINA GP	QUALIFYING	3th (1'34.295)	9th (1'34.993)	#2 R. BARRICHELLO 1'34.012	#2 R. BARRICHELLO
Shanghai	RACE	2th	6th		#9 J. BUTTON
5.451km×56Laps=305.066km	(DELAY)	(1.035)	(54.791)		#6 K.RAIKKONEN
ROUND 17 JAPAN GP	QUALIFYING	5th (1'35.157)	4th (1'34.897)	#1 M. SCHUMACHER 1'33.542	#1 M. SCHUMACHER
Suzuka	RACE	3th	4th		#4 R. SCHUMACHER
5.807km×53Laps=307.573km	(DELAY)	(19.662)	(31.781)		#9 J. BUTTON
ROUND 18 BRAZIL GP	QUALIFYING	5th (1'11'092)	6th (1'11.120)	#2 R. BARRICHELLO 1'10.646	#3. J. MONTOYA
Interlagos	RACE	R (3LAPS)	6th		#6 K. RAIKKONEN
4.309km×71Laps=305.909km	(DELAY)	-	(50.248)		#2 R. BARRICHELLO

※1 As a result of an engine replacement penalty, R. Schumacher, who qualified in 2nd position, was demoted 10 grid positions, and J. Button and T. Sato took the 2nd and 7th positions on the grid, respectively.

※2 J. Button was raised from 4th to 3rd position as a result of R. Schumacher's disqualification due to a brake duct size infringement.

※3 Due to an engine replacement, F. Alonso, who qualified in 6th position, was demoted 10 grid positions; as a result T. Sato moved from 9th to 8th position on the grid.

※4 J. Button qualified 3rd, but was demoted 10 grid positions due to an engine replacement. He started the race in 13th position on the grid, while T. Sato was raised from 9th to 8th position on the grid.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRN	SMR	ESP	MON	EUR	CAN	USA	FRA	GBR	GER	HUN	BEL	ITA	CHN	JPN	BRA
1	Scuderia Ferrari Marlboro	262	18	15	18	13	18	6	18	18	18	16	16	10	18	14	18	10	10	8
2	Lucky Strike BAR Honda	119	3	6	10	8	5	8	6	6	6	4	5	9	7		11	11	11	3
3	Mild Seven Renault F1 Team	105	8	6	8	9	11	10	9		5	13		6	6			5	4	5
4	BMW Williams F1 Team	88	9	8	2	8	3	5	1			1	4	6	7		6	4	10	14
5	West McLaren Mercedes	69	1	3		1				7	5	5	10	5		12	3	6	3	8
6	Sauber Petronas	34		1			2	4	3	5			3		1	9	1	3	1	1
7	Jaguar Racing	10			1				2				1	3		3				
8	Panasonic Toyota Racing	9						4			4					1				
9	Jordan Ford	5						2		3										
10	Minardi Cosworth	1									1									

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRN	SMR	ESP	MON	EUR	CAN	USA	FRA	GBR	GER	HUN	BEL	ITA	CHN	JPN	BRA
1	1	M. SCHUMACHER (GER)	148	10	10	10	10	10		10	10	10	10	10	10	10	8	8		10	2
2	2	R. BARRICHELLO (BRA)	114	8	5	8	3	8	6	8	8	8	6	6		8	6	10	10		6
3	9	J. BUTTON (GBR)	85	3	6	6	8	1	8	6	6		4	5	8	4		6	8	6	
4	8	F. ALONSO (ESP)	59	6	2	3	5	5		4			8		6	6			5	4	5
5	3	J. MONTOYA (COL)	58	4	8		6		5	1			1	4	4	5		4	4	2	10
6	7	J. TRULLI (ITA)	46	2	4	5	4	6	10	5		5	5								
7	6	K. RAIKKONEN (FIN)	45				1				4	3	2	8			10		6	3	8
8	10	T. SATO (JPN)	34			4		4				6			1	3		5	3	5	3
9	4	R. SCHUMACHER (GER)	24	5		2	2	3												8	4
10	5	D. COULTHARD (GBR)	24	1	3						3	2	3	2	5		2	3			
11	11	G. FISICHELLA (ITA)	22					2		3	5			3		1	4	1	2	1	
12	12	F. MASSA (BRA)	12		1				4								5		1		1
13	14	M. WEBBER (AUS)	7			1				2				1	3						
14	17	O. PANIS (FRA)	6						1			4					1				
15	4	A. PIZZONIA (BRA)	6												2	2		2			
16	15	C. KLIEN (AUT)	3														3				
17	16	C. DA MATTA (BRA)	3						3												
18	18	N. HEIDFELD (GER)	3						2		1										
19	19	T. GLOCK (GER)	2								2										
20	21	Z. BAUMGARTNER (HUN)	1									1									
21	7	J. VILLENEUVE (CAN)	0																		
22	17	R. ZONTA (BRA)	0																		
23	4	M. GENE (ESP)	0																		
24	19	G. PANTANO (ITA)	0																		
25	20	G. BRUNI (ITA)	0																		

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike BAR Honda		POLE	PODIUM
		#3 J. BUTTON	#4 T. SATO	POSITION	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	8th (3'12.128)※1	- ※1	#6 G. FISICHELLA 3'01.460※1	#6 G. FISICHELLA
Melbourne	RACE	11th	14th		#2 R. BARRICHELLO
5.303km×57Laps=302.271km	(DELAY)	(DNF)	(DNF)		#5 F. ALONSO
			#4 A. DAVIDSON		
ROUND 2 MALAYSIA GP	QUALIFYING	9th (3'09.832)※1	15th (3'11.890)※1	#5 F. ALONSO 3'07.672※1	#5 F. ALONSO
Kuala Lumpur	RACE	R (2LAPS)	R (2LAPS)		#16 J. TRULLI
5.543km×56Laps=310.408km	(DELAY)	-	-		#8 N. HEIDFELD
			#4 T. SATO		
ROUND 3 BAHRAIN GP	QUALIFYING	11th (3'04.348)※1	13th (3'05.563)※1	#5 F. ALONSO 3'01.902※1	#5 F. ALONSO
Bahrain	RACE	R (46LAPS)	R (27LAPS)		#16 J. TRULLI
5.412km×57Laps=308.238km	(DELAY)	-	-		#9 K. RAIKKONEN
ROUND 4 SAN MARINO GP	QUALIFYING	3th (2'44.105)※1	6th (2'44.658)※1	#9 K. RAIKKONEN 2'42.880※1	#5 F. ALONSO
Imola	RACE	D ※2	D ※2		#1 M. SCHUMACHER
4.933km×62Laps=305.609km	(DELAY)	-	-		#10 A. WURZ
ROUND 5 SPAIN GP	QUALIFYING	- ※3	- ※3	#9 K. RAIKKONEN 2'31.421※1	#9 K. RAIKKONEN
Barcelona	RACE	-	-		#5 F. ALONSO
4.627km×66Laps=305.256km	(DELAY)				#16 J. TRULLI
ROUND 6 MONACO GP	QUALIFYING	- ※3	- ※3	#9 K. RAIKKONEN 2'30.323※1	#9 K. RAIKKONEN
Monte Carlo	RACE	-	-		#8 N. HEIDFELD
3.340km×78Laps=260.520km	(DELAY)				#7 M. WEBBER
		予選方式の変更※4			
ROUND 7 EUROPE GP	QUALIFYING	13th (1'32.594)	16th (1'32.926)	#8 N. HEIDFELD 1'30.081	#5 F. ALONSO
Nurburgring	RACE	10th	12th		#8 N. HEIDFELD
5.148km×59Laps=303.715km	(DELAY)	(95.786)	(1LAPS)		#2 R. BARRICHELLO
ROUND 8 CANADA GP	QUALIFYING	PP (1'15.217)	6th (1'15.729)	#3 J. BUTTON 1'15.217	#9 K. RAIKKONEN
Montreal	RACE	R (46LAPS)	R (40LAPS)		#1 M. SCHUMACHER
4.361km×70Laps=305.270km	(DELAY)	-	-		#2 R. BARRICHELLO
ROUND 9 UNITED STATES GP	QUALIFYING	3th (1'11.277)	8th (1'11.497)	#16 J. TRULLI 1'10.625	#1 M. SCHUMACHER
Indianapolis	RACE	R (0LAPS)	R (0LAPS)		#2 R. BARRICHELLO
4.192km×73Laps=306.016km	(DELAY)	-	-		#18 T. MONTEIRO
ROUND 10 FRANCE GP	QUALIFYING	8th (1'15.051)	5th (1'14.655)	#5 F. ALONSO 1'14.412	#5 F. ALONSO
Magny Cours	RACE	4th	11th		#9 K. RAIKKONEN
4.411km×70Laps=308.586km	(DELAY)	(1LAP)	(1LAP)		#1 M. SCHUMACHER
ROUND 11 GREAT BRITAIN GP	QUALIFYING	3th (1'20.207)※5	8th (1'21.114)※5	#5 F. ALONSO 1'19.905	#10 J. MONTOYA
Silverstone	RACE	5th	16th		#5 F. ALONSO
5.141km×60Laps=308.355km	(DELAY)	(40.264)	(2LAPS)		#9 K. RAIKKONEN
ROUND 12 GERMANY GP	QUALIFYING	2th (1'14.759)	8th (1'15.501)	#9 K. RAIKKONEN 1'14.320	#5 F. ALONSO
Hockenheim	RACE	3th	12th		#10 J. MONTOYA
4.574km×67Laps=306.458km	(DELAY)	(24.422)	(1LAP)		#3 J. BUTTON
ROUND 13 HUNGARY GP	QUALIFYING	8th (1'21.302)	10th (1'21.787)	#1 M. SCHUMACHER 1'19.882	#9 K. RAIKKONEN
Budapest	RACE	5th	8th		#1 M. SCHUMACHER
4.381km×70Laps=306.663km	(DELAY)	(58.832)	(1LAP)		#17 R. SCHUMACHER
ROUND 14 TURKEY GP	QUALIFYING	13th (1'30.063)	- ※6	#9 K. RAIKKONEN 1'26.797	#9 K. RAIKKONEN
Istanbul	RACE	5th	9th		#5 F. ALONSO
5.338km×58Laps=309.396km	(DELAY)	(39.304)	(109.987)		#10 J. MONTOYA
ROUND 15 ITALY GP	QUALIFYING	4th (1'21.369)	5th (1'21.477)	#9 K. RAIKKONEN 1'20.878	#10 J. MONTOYA
Monza	RACE	8th	16th		#5 F. ALONSO
5.793km×53Laps=306.720km	(DELAY)	(63.635)	(1LAP)		#6 G. FISICHELLA
ROUND 16 BELGIUM GP	QUALIFYING	9th (1'47.978)	11th (1'48.353)	#10 J. MONTOYA 1'46.391	#9 K. RAIKKONEN
Spa Francorchamps	RACE	3th	R (13LAPS)		#5 F. ALONSO
6.976km×44Laps=306.927km	(DELAY)	(32.077)	-		#3 J. BUTTON
ROUND 17 BRAZIL GP	QUALIFYING	4th (1'12.696)	-	#5 F. ALONSO 1'11.988	#10 J. MONTOYA
Interlagos	RACE	7th	10th		#9 K. RAIKKONEN
4.309km×71Laps=305.909km	(DELAY)	(1LAP)	(1LAP)		#5 F. ALONSO
ROUND 18 JAPAN GP	QUALIFYING	2th (1'46.141)	5th (1'46.841)	#17 R. SCHUMACHER 1'46.106	#9 K. RAIKKONEN
Suzuka	RACE	5th	- ※7		#6 G. FISICHELLA
5.807km×53Laps=307.573km	(DELAY)	(29.507)	-		#5 F. ALONSO
ROUND 19 CHINA GP	QUALIFYING	4th (1'34.801)	17th (1'37.083)	#5 F. ALONSO 1'34.080	#5 F. ALONSO
Shanghai	RACE	8th	R (34LAPS)		#9 K. RAIKKONEN
5.451km×56Laps=305.066km	(DELAY)	(41.249)	-		#17 R. SCHUMACHER

※1 From the opening round to Round 6 (Monaco), the aggregate of QF1 and QF2 times was used as the qualifying time.

※2 Disqualified due to vehicle weight infringement.

※3 Suspended from racing due to vehicle weight infringement at San Marino GP.

※4 From Round 7, rule changes specified a single Saturday qualifying run.

※5 K. Räikkönen, who was qualified in 2nd position, was demoted 10 grid positions due to an engine replacement penalty, raising J. Button from 3rd to 2nd grid position and T. Sato from 8th to 7th grid position.

※6 Qualifying times were deleted as a result of impeding another vehicle.

※7 Eliminated from results due to contact with another vehicle.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRN	SMR	ESP	MON	EUR	CAN	USA	FRA	GBR	GER	HUN	TUR	ITA	BEL	BRA	JPN	CHN
1	Mild Seven Renault F1 Team	191	16	10	10	10	12	5	13			13	13	15		13	14	8	10	14	15
2	Team McLaren Mercedes	182	4	5	10	6	12	14	2	10		8	16	8	10	16	15	10	18	10	8
3	Scuderia Ferrari Marlboro	100	8	2		8		3	10	14	18	6	5	4	8			4	8	2	
4	Panasonic Toyota Racing	88		12	13	4	11	3	1	3		6	1	3	11	3	7	2	1	1	6
5	BMW Williams F1 Team	66	4	6	3	5	3	14	8	4					5		2	5		5	2
6	Lucky Strike BAR Honda	38										5	4	6	5	4	1	6	2	4	1
7	Red Bull Racing	34	7	4	1	1	1		5	3				2		3				3	4
8	Sauber Petronas	20			2	5				5		1		1				3			3
9	Jordan Grand Prix	12									11							1			
10	Minardi F1 Team	7									7										

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRN	SMR	ESP	MON	EUR	CAN	USA	FRA	GBR	GER	HUN	TUR	ITA	BEL	BRA	JPN	CHN
1	5	F. ALONSO (ESP)	133	6	10	10	10	8	5	10			10	8	10		8	8	8	6	6	10
2	9	K. RAIKKONEN (FIN)	112	1		6		10	10		10		8	6		10	10	5	10	8	10	8
3	1	M. SCHUMACHER (GER)	62		2		8		2	4	8	10	6	3	4	8				5	2	
4	10	J. MONTOYA (COL)	60	3	5			2	4	2				10	8		6	10		10		
5	6	G. FISICHELLA (ITA)	58	10				4		3			3	5	5		5	6		4	8	5
6	17	R. SCHUMACHER (GER)	45		4	5		5	3		3		2	1	3	6		3	2	1	1	6
7	16	J. TRULLI (ITA)	43		8	8	4	6		1			4			5	3	4				
8	2	R. BARRICHELLO (BRA)	38	8					1	6	6	8		2					4	3		
9	3	J. BUTTON (GBR)	37										5	4	6	4	4	1	6	2	4	1
10	7	M. WEBBER (AUS)	36	4		3	2	3	6		4					2			5		5	2
11	8	N. HEIDFELD (GER)	28		6		3		8	8						3						
12	14	D. COULTHARD (GBR)	24	5	3	1		1		5	2				2		2				3	
13	12	F. MASSA (BRA)	11			2					5				1							3
14	11	J. VILLENEUVE (CAN)	9				5						1						3			
15	15/37	C. KLIEN (AUT)	9	2	1						1						1					4
16	18	T. MONTEIRO (POR)	7									6							1			
17	10/35	A. WURZ (AUT)	6				6															
18	19	N. KARTHIKEYAN (IND)	5									5										
19	21	C. ALBERS (NED)	4									4										
20	10/35	P. DE LA ROSA (ESP)	4			4																
21	20	P. FRIESACHER (AUT)	3									3										
22	8	A. PIZZONIA (BRA)	2															2				
23	4	T. SATO (JPN)	1													1						
24	15/37	V. LIUZZI (ITA)	1				1															
25	20	R. DOORNBOS (NED)	0																			

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Lucky Strike Honda Racing F1 Team		Super Aguri F1		POLE	QF FASTEST	PODIUM
		#11 R. BARRICHELLO	#12 J. BUTTON	#22 T. SATO	#23 Y. IDE	POSITION	LAP TIME	(FIRST 3)
ROUND 1 BAHRAIN GP	QUALIFYING	6th (1'32.579)	3th (1'31.549)	20th (1'37.411)	21th (1'40.270)	#5 M. SCHUMACHER 1'31.431	#5 M. SCHUMACHER 1'31.431	#1 F. ALONSO
Bahrain	RACE	15th	4th	18th	R (35LAPS)			#5 M. SCHUMACHER
5.412km×57Laps=308.238km	(DELAY)	(1LAP)	(19.992)	(4LAPS)	-			#3 K. RAIKKONEN
ROUND 2 MALAYSIA GP	QUALIFYING	12th (1'34.683)	2th (1'33.986)	21th (1'39.011)	22th (1'40.720)	#2 G. FISICHELLA 1'33.840	#12 J. BUTTON 1'33.527	#2 G. FISICHELLA
Kuala Lumpur	RACE	10th	3th	14th	R (33LAPS)			#1 F. ALONSO
5.543km×56Laps=310.408km	(DELAY)	(1LAP)	(9.631)	(3LAPS)	-			#12 J. BUTTON
ROUND 3 AUSTRALIA GP	QUALIFYING	17th (1'29.943)	PP (1'25.229)	21th (1'32.279)	22th (1'36.164)	#12 J. BUTTON 1'25.229	#12 J. BUTTON 1'25.229	#1 F. ALONSO
Melbourne	RACE	7th	10th	12th	13th			#3 K. RAIKKONEN
5.303km×57Laps=302.271km	(DELAY)	(51.904)	DNF	(2LAPS)	(3LAPS)			#7 R. SCHUMACHER
ROUND 4 SAN MARINO GP	QUALIFYING	3th (1'23.242)	2th (1'22.988)	21th (1'27.609)	22th (1'29.282)	#5 M. SCHUMACHER 1'22.795	#5 M. SCHUMACHER 1'22.579	#5 M. SCHUMACHER
Imola	RACE	10th	7th	R (44LAPS)	R (23LAPS)			#1 F. ALONSO
4.959km×62Laps=307.221km	(DELAY)	(77.851)	(39.635)	-	-			#4 J. MONTOYA
					#23 F. MONTAGNY			
ROUND 5 EUROPE GP	QUALIFYING	4th (1'30.754)	6th (1'30.940)	21th (1'35.239)	22th (1'46.505)	#1 F. ALONSO 1'29.819	#1 F. ALONSO 1'29.819	#5 M. SCHUMACHER
Nurburgring	RACE	5th	R (28LAPS)	R (45LAPS)	R (29LAPS)			#1 F. ALONSO
5.148km×60Laps=308.863km	(DELAY)	(72.586)	-	-	-			#6 F. MASSA
ROUND 6 SPAIN GP	QUALIFYING	5th (1'15.885)	8th (1'16.008)	20th (1'18.920)	21th (1'20.763)	#1 F. ALONSO 1'14.648	#5 M. SCHUMACHER	#1 F. ALONSO
Barcelona	RACE	7th	6th	17th	R (10LAPS)		1'14.637	#5 M. SCHUMACHER
4.627km×66Laps=305.256km	(DELAY)	(1LAP)	(58.347)	(4LAPS)	-			#2 G. FISICHELLA
ROUND 7 MONACO GP	QUALIFYING	- ※1	13th (1'14.982)※2	19th (1'17.276)※2	20th (1'17.502)※2	#1 F. ALONSO※2 1'13.962※2	#3 K. RAIKKONEN	#1 F. ALONSO
Monte Carlo	RACE	4th	11th	R (46LAPS)	16th		1'13.532	#4 J. MONTOYA
3.340km×78Laps=260.520km	(DELAY)	(53.337)	(1LAP)	-	(3LAPS)			#14 D. COULTHARD
ROUND 8 GREAT BRITAIN GP	QUALIFYING	6th (1'20.943)	19th (1'23.247)	20th (1'26.158)	21th (1'26.316)	#1 F. ALONSO 1'20.253	#1 F. ALONSO	#1 F. ALONSO
Silverstone	RACE	10th	R (8LAPS)	17th	18th		1'20.253	#5 M. SCHUMACHER
5.141km×60Laps=308.355km	(DELAY)	(1LAP)	-	(3LAPS)	(3LAPS)			#3 K. RAIKKONEN
ROUND 9 CANADA GP	QUALIFYING	9th (1'16.912)	8th (1'16.608)	21th (1'19.088)	22th (1'19.152)	#1 F. ALONSO 1'14.942	#1 F. ALONSO	#1 F. ALONSO
Montreal	RACE	R (11LAPS)	9th	15th	R (2LAPS)		1'14.726	#5 M. SCHUMACHER
4.361km×70Laps=305.270km	(DELAY)	-	(1LAP)	(DNF)	-			#3 K. RAIKKONEN
ROUND 10 UNITED STATES GP	QUALIFYING	4th (1'12.109)	7th (1'12.523)	18th (1'13.496)	21th (1'16.036)	#5 M. SCHUMACHER 1'10.832	#5 M. SCHUMACHER	#5 M. SCHUMACHER
Indianapolis	RACE	6th	R (3LAPS)	R (6LAPS)	R (0LAP)		1'10.636	#6 F. MASSA
4.192km×73Laps=306.016km	(DELAY)	(36.516)	-					#2 G. FISICHELLA
ROUND 11 FRANCE GP	QUALIFYING	14th (1'17.027)	19th (1'17.495)	22th (1'18.845)	21th (1'18.637)	#5 M. SCHUMACHER 1'15.493	#5 M. SCHUMACHER	#5 M. SCHUMACHER
Magny Cours	RACE	R (18LAPS)	R (61LAPS)	R (0LAP)	16th		1'15.111	#1 F. ALONSO
4.411km×70Laps=308.586km	(DELAY)	-	-	-	(3LAPS)			#6 F. MASSA
					#23 S. YAMAMOTO			
ROUND 12 GERMANY GP	QUALIFYING	6th (1'14.934)	4th (1'14.862)	19th (1'17.185)	21th (1'20.444)	#3 K. RAIKKONEN 1'14.070	#5 M. SCHUMACHER	#5 M. SCHUMACHER
Hockenheim	RACE	R (18LAPS)	4th	R (38LAPS)	R (1LAP)		1'13.778	#6 F. MASSA
4.574km×67Laps=306.458km	(DELAY)	-	(18.898)	-	-			#3 K. RAIKKONEN
ROUND 13 HUNGARY GP	QUALIFYING	3th (1'20.085)	4th (1'20.092)※3	19th (1'22.967)	22th (1'24.016)	#3 K. RAIKKONEN 1'19.599	#6 F. MASSA	#12 J. BUTTON
Budapest	RACE	4th	WIN	13th	R (0LAP)		1'19.504	#4 P. DE LA ROSA
4.381km×70Laps=306.663km	(DELAY)	(45.205)	-	(5LAPS)				#16 N. HEIDFELD
ROUND 14 TURKEY GP	QUALIFYING	14th (1'28.257)	7th (1'27.790)	22th (1'30.850)	21th (1'30.607)	#6 F. MASSA 1'26.907	#5 M. SCHUMACHER	#6 F. MASSA
Istanbul	RACE	8th	4th	R (41LAPS)	R (23LAPS)		1'25.850	#1 F. ALONSO
5.338km×58Laps=309.396km	(DELAY)	(60.034)	(12.334)	-	-			#5 M. SCHUMACHER
ROUND 15 ITALY GP	QUALIFYING	8th (1'22.787)※4	5th (1'22.011)※4	21th (1'24.289)	22th (1'26.001)	#3 K. RAIKKONEN 1'21.484	#6 F. MASSA	#5 M. SCHUMACHER
Monza	RACE	6th	5th	16th	R (18LAPS)		1'21.225	#3 K. RAIKKONEN
5.793km×53Laps=306.720km	(DELAY)	(42.409)	(32.685)	(2LAPS)	-			#17 R. KUBICA
ROUND 16 CHINA GP	QUALIFYING	3th (1'45.503)	4th (1'45.503)	20th (1'50.326)※5	21th (1'55.560)※5	#1 F. ALONSO 1'44.360	#1 F. ALONSO	#5 M. SCHUMACHER
Shanghai	RACE	6th	4th	- ※6	16th※7		1'43.951	#1 F. ALONSO
5.451km×56Laps=305.066km	(DELAY)	(79.131)	(72.056)	-	(4LAPS)			#2 G. FISICHELLA
ROUND 17 JAPAN GP	QUALIFYING	8th (1'31.478)	7th (1'30.992)	20th (1'33.666)	-	#6 F. MASSA 1'29.599	#5 M. SCHUMACHER	#1 F. ALONSO
Suzuka	RACE	12th	4th	15th	17th		1'28.954	#6 F. MASSA
5.807km×53Laps=307.573km	(DELAY)	(1LAP)	(34.101)	(1LAP)	(3LAPS)			#2 G. FISICHELLA
ROUND 18 BRAZIL GP	QUALIFYING	5th (1'11.619)	14th (1'11.742)	20th (1'13.269)	21th (1'13.357)	#6 F. MASSA 1'10.680	#5 M. SCHUMACHER	#6 F. MASSA
Interlagos	RACE	7th	3th	10th	16th		1'10.313	#1 F. ALONSO
4.309km×71Laps=305.909km	(DELAY)	(40.294)	(19.394)	(1LAP)	(2LAPS)			#12 J. BUTTON

※1 Qualifying times deleted for a judgment of intentionally impeding another vehicle.

※2 M. Schumacher's qualifying times were deleted for intentionally impeding another vehicle in Q3, raising J. Button one position.

※3 Due to an engine replacement penalty, J. Button was demoted 10 grid positions, from 4th to 14th position on the grid.

※4 F. Alonso was qualified in 5th position, but his best three times in Q3 were deleted for impeding F. Massa. This raised R. Barrichello from 9th to 8th position and J. Button from 6th to 5th position.

※5 R. C. Albers' qualifying times were deleted for failing to respond to a request for vehicle scrutinizing. As a result, T. Sato was raised from 21st to 20th position, and S. Yamamoto from 22nd to 21st position.

※6 Eliminated from race results for ignoring a blue flag.

※7 As a result of **※6**, S. Yamamoto was raised from 17th to 16th position.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	BRN	MAL	AUS	SMR	EUR	ESP	MON	GBR	CAN	USA	FRA	GER	HUN	TUR	ITA	CHN	JPN	BRA
1	Mild Seven Renault F1 Team	206	10	18	14	9	11	16	13	15	15	10	11	7		11	5	14	16	11
2	Scuderia Ferrari Marlboro	201	8	7		15	16	13	4	12	12	18	16	18	3	16	10	10	8	15
3	Team McLaren Mercedes	110	10	5	8	10	5	4	8	9	6		6	6	8	4	8	4	4	5
4	Lucky Strike Honda Racing F1 Team	86	5	6	2	2	4	5	5			3		5	15	6	7	8	5	8
5	BMW Sauber F1 Team	36		2	8		1	1	2	3	2		1		6		7	2	1	
6	Panasonic Toyota Racing	35		1	6				1		3	5	5	2	3	2	2		5	
7	Red Bull Racing	16	1		1				6		1	2		1	4					
8	Williams F1 Team	11	5			3	2											1		
9	Scuderia Toro Rosso	1										1								
10	Spyker M F1 Team	0																		
11	Super Aguri Formula 1	0																		

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	BRN	MAL	AUS	SMR	EUR	ESP	MON	GBR	CAN	USA	FRA	GER	HUN	TUR	ITA	CHN	JPN	BRA
1	1	F. ALONSO (ESP)	134	10	8	10	8	8	10	10	10	10	4	8	4		8		8	10	8
2	5	M.SCHUMACHER (GER)	121	8	3		10	10	8	4	8	8	10	10	10	1	6	10	10		5
3	6	F. MASSA (BRA)	80		4		5	6	5		4	4	8	6	8	2	10			8	10
4	2	G. FISICHELLA (ITA)	72		10	4	1	3	6	3	5	5	6	3	3		3	5	6	6	3
5	3	K. RAIKKONEN (FIN)	65	6		8	4	5	4		6	6		4	6			8		4	4
6	12	J. BUTTON (GBR)	56	5	6		2		3						5	10	5	4	5	5	6
7	11	R. BARRICHELLO (BRA)	30			2		4	2	5			3			5	1	3	3		2
8	4	J. MONTOYA (COL)	26	4	5		6			8	3										
9	16	N. HEIDFELD (GER)	23			5			1	2	2	2		1		6		1	2	1	
10	7	R. SCHUMACHER (GER)	20		1	6				1				5		3	2			2	
11	4	P. DE LA ROSA (ESP)	19											2		8	4		4		1
12	8	J. TRULLI (ITA)	15									3	5		2			2		3	
13	14	D. COULTHARD (GBR)	14			1				6		1	2			4					
14	9	M. WEBBER (AUS)	7	3			3												1		
15	17	J. VILLENEUVE (CAN)	7		2	3		1			1										
16	17/38	R. KUBICA (POR)	6															6			
17	10	N. ROSBERG (GER)	4	2				2													
18	15	C. KLIEN (AUT)	2	1																	
19	20	V. LIUZZI (ITA)	1									1									
20	21	S. SPEED (USA)																			
21	18	T. MONTEIRO (POL)																			
22	19	C. ALBERS (NED)																			
23	22	T. SATO (JPN)																			
24	15/37	R. DOORNBOS (NED)																			
25	23	Y. IDE (JPN)																			
26	23/41	S. YAMAMOTO (JPN)																			
27	23/41	F. MONTAGNY																			

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Honda Racing F1 Team		Super Aguri F1 Team		POLE POSITION	QF FASTEST LAP TIME	PODIUM (FIRST 3)
		#7 J. BUTTON	#8 R. BARRICHELLO	#12 T. SATO	#14 A. DAVIDSON			
ROUND 1 AUSTRALIA GP	QUALIFYING	14th (1'27.264)	17th (1'27.679)	10th (1'28.871)	11th (1'26.909)	#6 K. RAIKKONEN 1'26.072	#1 F. ALONSO 1'25.326	#6 K. RAIKKONEN
Melbourne	RACE	15th	11th	12th	16th			#1 F. ALONSO
5.303km×58Laps=307.574km	(DELAY)	(1LAP)	(1LAP)	(1LAP)	(2LAPS)			#2 L. HAMILTON
ROUND 2 MALAYSIA GP	QUALIFYING	15th (1'36.088)	19th (1'36.827)	14th (1'35.945)	18th (1'36.816)	#5 F. MASSA 1'35.043	#1 F. ALONSO 1'34.057	#1 F. ALONSO
Kuala Lumpur	RACE	12th	11th	13th	16th			#2 L. HAMILTON
5.543km×56Laps=310.408km	(DELAY)	(1LAP)	(1LAP)	(1LAP)	(1LAP)			#6 K. RAIKKONEN
ROUND 3 BAHRAIN GP	QUALIFYING	16th (1'33.731)	15th (1'33.624)	17th (1'33.984)	13th (1'33.082)	#5 F. MASSA 1'32.652	#5 F. MASSA 1'31.359	#5 F. MASSA
Bahrain	RACE	R (0LAP)	13th	R (34LAPS)	16th			#2 L. HAMILTON
5.412km×57Laps=308.238km	(DELAY)	-	(1LAP)	-	(DNF)			#6 K. RAIKKONEN
ROUND 4 SPAIN GP	QUALIFYING	14th (1'22.120)	12th (1'22.097)	13th (1'22.115)	15th (DNF)	#5 F. MASSA 1'21.421	#5 F. MASSA 1'20.597	#5 F. MASSA
Barcelona	RACE	12th	10th	8th	11th			#2 L. HAMILTON
4.655km×65Laps=302.449km	(DELAY)	(1LAP)	(1LAP)	(1LAP)	(1LAP)			#1 F. ALONSO
ROUND 5 MONACO GP	QUALIFYING	10th (1'17.939)※1	9th (1'17.498)※1	21th (1'18.554)	17th (1'18.250)	#1 F. ALONSO 1'15.726	#1 F. ALONSO 1'15.431	#1 F. ALONSO
Monte Carlo	RACE	11th	10th	17th	18th			#2 L. HAMILTON
3.340km×78Laps=260.520km	(DELAY)	(1LAP)	(1LAP)	(2LAPS)	(2LAPS)			#5 F. MASSA
ROUND 6 CANADA GP	QUALIFYING	15th (1'17.541)	13th (1'17.116)	11th (1'16.743)	17th (1'17.542)	#2 L. HAMILTON 1'15.707	#2 L. HAMILTON 1'15.486	#2 L. HAMILTON
Montreal	RACE	R (0LAP)	12th	6th	11th			#9 N. HEIDFELD
4.361km×70Laps=305.270km	(DELAY)	-	30.439	16.698	24.318			#17 A. WURZ
ROUND 7 UNITED STATES GP	QUALIFYING	13th (1'12.998)	15th (1'13.201)	18th (1'13.477)	16th (1'13.259)	#2 L. HAMILTON 1'12.331	#1 F. ALONSO 1'11.926	#2 L. HAMILTON
Indianapolis	RACE	12th	R (0LAP)	R (13LAPS)	11th			#1 F. ALONSO
4.192km×73Laps=306.016km	(DELAY)	(1LAP)	-	-	(1LAP)			#5 F. MASSA
ROUND 8 FRANCE GP	QUALIFYING	12th (1'15.584)	13th (1'15.761)	19th (1'16.244)※2	20th (1'16.366)	#5 F. MASSA 1'15.034	#2 L. HAMILTON 1'14.795	#6 K. RAIKKONEN
Magny Cours	RACE	8th	11th	16th	R (1LAP)			#5 F. MASSA
4.411km×70Laps=308.586km	(DELAY)	58.885	(1LAP)	(2LAPS)	-			#2 L. HAMILTON
ROUND 9 GREAT BRITAIN GP	QUALIFYING	18th (1'21.335)	14th (1'20.364)	21th (1'22.045)	19th (1'21.448)	#2 L. HAMILTON 1'19.997	#1 F. ALONSO 1'19.152	#6 K. RAIKKONEN
Silverstone	RACE	10th	9th	14th	R (35LAPS)			#1 F. ALONSO
5.141km×59Laps=303.214km	(DELAY)	(1LAP)	(1LAP)	(2LAP)	-			#2 L. HAMILTON
ROUND 10 EUROPE GP	QUALIFYING	17th (1'32.983)	14th (1'32.221)	16th (1'32.838)	15th (1'32.451)	#6 K. RAIKKONEN 1'31.450	#5 F. MASSA 1'30.912	#1 F. ALONSO
Nurburgring	RACE	R (2LAPS)	11th	R (19LAPS)	12th			#5 F. MASSA
5.148km×60Laps=308.863km	(DELAY)	-	(1LAP)	-	(1LAP)			#15 M. WEBBER
ROUND 11 HUNGARY GP	QUALIFYING	17th (1'21.737)	18th (1'21.877)	19th (1'22.143)	15th (1'21.127)	#1 F. ALONSO※3 1'19.674	#2 L. HAMILTON 1'19.301	#2 L. HAMILTON
Budapest	RACE	R (35LAPS)	18th	15th	R (41LAPS)			#6 K. RAIKKONEN
4.381km×70Laps=306.663km	(DELAY)	-	(2LAPS)	(1LAP)				#9 N. HEIDFELD
ROUND 12 TURKEY GP	QUALIFYING	15th (1'28.220)	14th (1'28.188)	19th (1'28.953)	11th (1'28.002)	#5 F. MASSA 1'27.329	#1 F. ALONSO 1'26.841	#5 F. MASSA
Istanbul	RACE	13th	17th	18th	14th			#6 K. RAIKKONEN
5.338km×58Laps=309.396km	(DELAY)	(1LAP)	(1LAP)	(1LAP)	(1LAP)			#1 F. ALONSO
ROUND 13 ITALY GP	QUALIFYING	10th (1'25.165)	12th (1'23.176)	17th (1'23.749)	14th (1'23.274)	#1 F. ALONSO 1'21.997	#1 F. ALONSO 1'21.356	#1 F. ALONSO
Monza	RACE	8th	10th	16th	14th			#2 L. HAMILTON
5.793km×53Laps=306.720km	(DELAY)	72.168	76.958	(1LAP)	(1LAP)			#6 K. RAIKKONEN
ROUND 14 BELGIUM GP	QUALIFYING	14th (1'46.955)	18th (1'47.954)	19th (1'47.980)	21th (1'48.199)	#6 K. RAIKKONEN 1'45.994	#6 K. RAIKKONEN 1'45.070	#6 K. RAIKKONEN
Spa Francorchamps	RACE	R (36LAPS)	13th	15th	16th			#5 F. MASSA
7.004km×44Laps=308.053km	(DELAY)	-	(1LAP)	(1LAP)	(1LAP)			#1 F. ALONSO
ROUND 15 JAPAN GP	QUALIFYING	7th (1'26.913)	17th (1'27.323)	21th (1'28.792)	19th (1'27.564)	#2 L. HAMILTON 1'25.368	#2 L. HAMILTON 1'24.753	#2 L. HAMILTON
Fuji	RACE	11th	10th	15th	R (54LAPS)			#4 H. KOVALAINEN
4.563km×67Laps=305.416km	(DELAY)	(DNF)	88.342	(DNF)	-			#6 K. RAIKKONEN
ROUND 16 CHINA GP	QUALIFYING	10th (1'39.285)	17th (1'37.251)	20th (1'38.218)	15th (1'37.247)	#2 L. HAMILTON 1'35.908	#6 K. RAIKKONEN 1'35.381	#6 K. RAIKKONEN
Shanghai	RACE	5th	15th	14th	R (11LAPS)			#1 F. ALONSO
5.451km×56Laps=305.066km	(DELAY)	68.666	(1LAP)	(1LAP)				#5 F. MASSA
ROUND 17 BRAZIL GP	QUALIFYING	16th (1'13.469)	11th (1'12.932)	18th (1'14.098)	20th (1'14.596)	#5 F. MASSA 1'11.931	#5 F. MASSA 1'11.931	#6 K. RAIKKONEN
Interlagos	RACE	R (20LAPS)	R (40LAPS)	12th	14th			#5 F. MASSA
4.309km×71Laps=305.909km	(DELAY)	-	-	(2LAPS)	(3LAPS)			#1 F. ALONSO

※1 D. Coulthard was not allowed to start in Q3 and was penalized two positions for impeding another vehicle in Q2. As a result, J. Button was raised from 11th to 10th position, and R. Barrichello from 10th to 9th position.

※2 T. Sato was penalized 10 grid positions for overtaking under a yellow flag during the previous US Grand Prix, and started the race at the back of the grid.

※3 F. Alonso was demoted five grid positions as a result of impeding another vehicle in the qualifying session.

CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRN	ESP	MON	CAN	USA	FRA	GBR	EUR	HUN	TUR	ITA	BEL	JPN	CHN	BRA
1	Scuderia Ferrari Marlboro	204	13	10	16	10	7	4	11	18	14	8	8	18	6	18	9	16	18
2	BMW Sauber F1 Team	101	5	5	8	5	7	8	1	9	8	5	10	6	9	4	2	2	7
3	ING Renault F1 Team	51	4	4	1	2	5	5	4	3	3	1	1	3	2	1	12		
4	AT&T Williams	33	2			3	2	6				5	2	2	3	3			5
5	Red Bull Racing	24				4			2			10				2	5	1	
6	Panasonic Toyota Racing	13	1	2	2			1	3				3						1
7	Scuderia Toro Rosso	8																8	
8	Honda Racing F1 Team	6								1					1			4	
9	Super Aguri F1 Team	4				1		3											
10	Etihad Aldar Spyker F1 Team	1															1		
11	Vodafone McLaren Mercedes	0																	

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRN	ESP	MON	CAN	USA	FRA	GBR	EUR	HUN	TUR	ITA	BEL	JPN	CHN	BRA
1	6	K. RAIKKONEN (FIN)	110	10	6	6		1	4	5	10	10		8	8	6	1 0	6	10	10
2	2	L. HAMILTON (GBR)	109	6	8	8	8	8	10	10	6	6		10	4	8	5	10		2
3	1	F. ALONSO (ESP)	109	8	10	4	6	10	2	8	2	8	10	5	6	10	6		8	6
4	5	F. MASSA (BRA)	94	3	4	10	10	6		6	8	4	8		10		8	3	6	8
5	9	N. HEIDFELD (GER)	61	5	5	5		3	8		4	3	3	6	5	5	4		2	3
6	10	R. KUBICA (POL)	39			3	5	4			5	5	2	4	1	4		2		4
7	4	H. KOVALAINEN (FIN)	30		1		2		5	4		2	1	1	3	2	1	8		
8	3	G. FISICHELLA (ITA)	21	4	3	1		5			3	1						4		
9	16	N. ROSBERG (GER)	20	2			3							2	2	3	3			5
10	14	D. COULTHARD (GBR)	14				4						4					5	1	
11	17	A. WURZ (AUT)	13					2	6				5							
12	15	M. WEBBER (AUS)	10							2			6				2			
13	12	J. TRULLI (ITA)	8		2	2				3										1
14	19	S. VETTEL (GER)	6							1									5	
15	7	J. BUTTON (GBR)	6								1					1			4	
16	11	R. SCHUMACHER (GER)	5	1					1					3						
17	22	T. SATO (JPN)	4				1		3											
18	18	V. LIUZZI (ITA)	3																3	
19	20	A. SUTIL (GER)	1															1		
20	8	R. BARRICHELLO (BRA)																		
21	19	S. SPEED (USA)																		
22	17/38	K. NAKAJIMA (JPN)																		
23	23	A. DAVIDSON (GBR)																		
24	21	S. YAMAMOTO (JPN)																		
25	21	C. ALBERS (NED)																		

FIA FORMULA ONE WORLD CHAMPIONSHIP

		Honda Racing F1 Team		Super Aguri F1 Team		POLE	QF FASTEST	PODIUM
		#7 J. BUTTON	#8 R. BARRICHELLO	#12 T. SATO	#14 A. DAVIDSON	POSITION	LAP TIME	(FIRST 3)
ROUND 1 AUSTRALIA GP	QUALIFYING	13th (1'26.259)	11th (1'26.173)	20th (1'28.208)	22th (1'29.059)	#22 L. HAMILTON 1'26.714	#22 L. HAMILTON 1'25.187	#22 L. HAMILTON
Melbourne	RACE	R (0LAP)	- ※1	R (32LAPS)	R (0LAP)			#3 N. HEIDFELD
5.303km×58Laps=307.574km	(DELAY)	-	-	-	-			#7 N. ROSBERG
ROUND 2 MALAYSIA GP	QUALIFYING	11th (1'35.208)	14th (1'35.622)	20th (1'37.087)	22th (1'37.481)	#2 F. MASSA 1'35.748	#1 K. RAIKKONEN 1'34.188	#1 K. RAIKKONEN
Kuala Lumpur	RACE	10th	13th	16th	15th			#4 R. KUBICA
5.543km×56Laps=310.408km	(DELAY)	86.214	(1LAP)	(2LAPS)	(1LAP)			#23 H. KOVALAINEN
ROUND 3 BAHRAIN GP	QUALIFYING	9th (1'35.057)	12th (1'32.508)	22th (1'35.725)	21th (1'34.140)	#4 R. KUBICA 1'33.096	#2 F. MASSA 1'31.188	#2 F. MASSA
Bahrain	RACE	R (19LAPS)	11th	17th	16th			#1 K. RAIKKONEN
5.412km×57Laps=308.238km	(DELAY)	-	77.862	(1LAP)	(1LAP)			#4 R. KUBICA
ROUND 4 SPAIN GP	QUALIFYING	13th (1'21.211)	11th (1'21.049)	22th (1'23.496)	21th (1'23.318)	#1 K. RAIKKONEN 1'21.813	#2 F. MASSA 1'20.584	#1 K. RAIKKONEN
Barcelona	RACE	6th	R (34LAPS)	13th	R (8LAPS)			#2 F. MASSA
4.655km×66Laps=307.104km	(DELAY)	53.010	-	(1LAP)	-			#22 L. HAMILTON
ROUND 5 TURKEY GP	QUALIFYING	13th (1'27.298)	12th (1'27.219)			#2 F. MASSA 1'27.617	#2 F. MASSA 1'25.994	#2 F. MASSA
Istanbul	RACE	11th	14th					#22 L. HAMILTON
5.338km×58Laps=309.396km	(DELAY)	(1LAP)	(1LAP)					#1 K. RAIKKONEN
ROUND 6 MONACO GP	QUALIFYING	12th (1'16.101) ※2	15th (1'16.537) ※2			#2 F. MASSA 1'15.787	#2 F. MASSA 1'15.110	#22 L. HAMILTON
Monte Carlo	RACE	11th	6th					#4 R. KUBICA
3.340km×76Laps=253.840km	(DELAY)	(1LAP)	(28.408)					#2 F. MASSA
ROUND 7 CANADA GP	QUALIFYING	19th (1'23.565)	9th (1'20.848)			#22 L. HAMILTON 1'17.886	#22 L. HAMILTON 1'16.909	#4 R. KUBICA
Montreal	RACE	11th	7th					#3 N. HEIDFELD
4.361km×70Laps=305.270km	(DELAY)	(67.540)	(53.597)					#9 D. COULTHARD
ROUND 8 FRANCE GP	QUALIFYING	17th (1'16.306)	18th (1'16.330)			#1 K. RAIKKONEN 1'16.449	#2 F. MASSA 1'15.024	#2 F. MASSA
Magny Cours	RACE	R (16LAPS)	14th					#1 K. RAIKKONEN
4.411km×70Laps=308.586km	(DELAY)	-	(1LAP)					#11 J. TRULLI
ROUND 9 GREAT BRITAIN GP	QUALIFYING	17th (1'21.631)	16th (1'21.512)			#23 H. KOVALAINEN 1'21.049	#22 L. HAMILTON 1'19.537	#22 L. HAMILTON
Silverstone	RACE	R (38LAPS)	3th					#3 N. HEIDFELD
5.141km×60Laps=308.355km	(DELAY)	-	(82.273)					#17 R. BARRICHELLO
ROUND 10 GERMANY GP	QUALIFYING	14th (1'15.701)	18th (1'16.246)			#22 L. HAMILTON 1'15.666	#22 L. HAMILTON 1'14.603	#22 L. HAMILTON
Hockenheim	RACE	17th	R (50LAPS)					#6 N. PIQUET
4.574km×67Laps=306.458km	(DELAY)	(1LAP)	-					#2 F. MASSA
ROUND 11 HUNGARY GP	QUALIFYING	12th (1'20.332)	18th (1'21.332) ※3			#22 L. HAMILTON 1'20.899	#2 F. MASSA 1'19.068	#23 H. KOVALAINEN
Budapest	RACE	12th	16th					#12 T. GLOCK
4.381km×70Laps=306.630km	(DELAY)	(1LAP)	(2LAPS)					#1 K. RAIKKONEN
ROUND 12 EUROPE GP	QUALIFYING	16th (1'38.880)	19th (1'39.811) ※4			#2 F. MASSA 1'38.989	#15 S. VETTEL 1'37.842	#2 F. MASSA
Valencia	RACE	13th	16th					#22 L. HAMILTON
5.419km×57Laps=308.883km	(DELAY)	(1LAP)	(1LAP)					#4 R. KUBICA
ROUND 13 BELGIUM GP	QUALIFYING	17th (1'48.211)	16th (1'48.153)			#22 L. HAMILTON 1'47.338	#23 H. KOVALAINEN 1'46.037	#2 F. MASSA
Spa Francorchamps	RACE	15th	R (19LAPS)					#3 N. HEIDFELD
7.004km×44Laps=308.052km	(DELAY)	(1LAP)	-					#22 L. HAMILTON ※5
ROUND 14 ITALY GP	QUALIFYING	19th (1'37.006)	16th (1'36.510)			#15 S. VETTEL 1'37.555	#23 H. KOVALAINEN 1'35.214	#15 S. VETTEL
Monza	RACE	15th	17th					#23 H. KOVALAINEN
5.793km×53Laps=306.720km	(DELAY)	(73.370)	(1LAP)					#4 R. KUBICA
ROUND 15 SINGAPORE GP	QUALIFYING	12th (1'45.133)	18th (1'46.583)			#2 F. MASSA 1'44.801	#2 F. MASSA 1'44.014	#5 F. ALONSO
Singapore	RACE	9th	R (14LAPS)					#7 N. ROSBERG
5.067km×61Laps=308.950km	(DELAY)	(19.885)	-					#22 L. HAMILTON
ROUND 16 JAPAN GP	QUALIFYING	18th (1'19.100)	17th (1'18.882)			#22 L. HAMILTON 1'18.404	#2 F. MASSA 1'17.287	#5 F. ALONSO
Fuji	RACE	14th	13th					#4 R. KUBICA
4.563km×67Laps=305.416km	(DELAY)	(1LAP)	(1LAP)					#1 K. RAIKKONEN
ROUND 17 CHINA GP	QUALIFYING	18th (1'37.053)	14th (1'36.079) ※6			#22 L. HAMILTON 1'36.303	#22 L. HAMILTON 1'34.947	#22 L. HAMILTON
Shanghai	RACE	16th	11th					#2 F. MASSA
5.451km×56Laps=305.066km	(DELAY)	(1LAP)	(85.061)					#1 K. RAIKKONEN
ROUND 18 BRAZIL GP	QUALIFYING	17th (1'12.810)	15th (1'13.139)			#2 F. MASSA 1'12.368	#23 H. KOVALAINEN 1'11.768	#2 F. MASSA
Interlagos	RACE	13th	15th					#5 F. ALONSO
4.309km×71Laps=305.909km	(DELAY)	(1LAP)	(1LAP)					#1 K. RAIKKONEN

※1 Eliminated from the race for ignoring a red signal at the pit lane exit.

※2 D. Coulthard was demoted five grid positions as a result of a gearbox change. As a result, J. Button was raised from 12th to 11th position, and R. Barrichello from 15th to 14th position on the grid.

※3 S. Bourdais was demoted five grid positions for impeding another vehicle in the qualifying session. As a result, R. Barrichello was raised from 18th to 17th position on the grid.

※4 Because a change was made to the vehicle in parc ferme, R. Barrichello was required to make a pit start in the race.

※5 25 seconds were added to L. Hamilton's racing time for cutting a chicane, demoting him from 1st to 3rd position.

※6 M. Webber was demoted 10 grid positions for an engine change, raising R. Barrichello from 14th to 13th position on the grid.

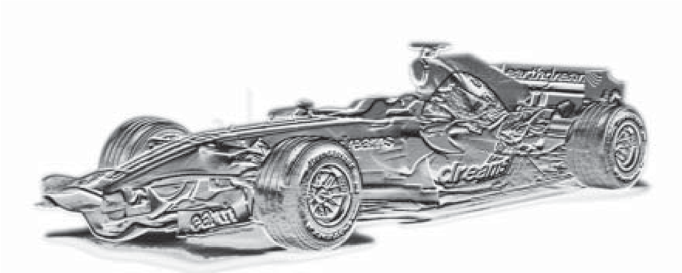
CONSTRUCTORS WORLD CHAMPIONSHIP

P.	TEAM (CAR NO.)	TOTAL	AUS	MAL	BRN	ESP	TUR	MON	CAN	FRA	GBR	GER	HUN	EUR	BEL	ITA	SIN	JPN	CHN	BRA
1	Scuderia Ferrari Marlboro	172	1	10	18	18	16	6	4	18	5	9	6	10	10	3		8	14	16
2	Vodafone McLaren Mercedes	151	14	10	4	6	8	11		5	14	14	14	13	6	10	6		10	6
3	BMW Sauber F1 Team	135	8	11	11	5	9	8	18	4	8	7	1	6	11	10	3	8	7	
4	ING Renault F1 Team	80	5	1			3			3	3	8	8		5	5	10	15	6	8
5	Panasonic Toyota Racing	56		5	3	1			8	6	2		10	6			5	4	2	4
6	Scuderia Toro Rosso	39	2					4	1			1		3	6	10	4	3		5
7	Red Bull Racing	29		2	2	4	2	5	6	3					1	1	2	1		
8	AT&T Williams	26	9		1	2	1	2			1			1			9			
9	Honda Racing F1 Team	14				3		3	2		6									
10	Force India F1 Team	0																		
11	Super Aguri F1 Team	0																		

DRIVERS WORLD CHAMPIONSHIP

P.	No.	DRIVER (NAT)	TOTAL	AUS	MAL	BRN	ESP	TUR	MON	CAN	FRA	GBR	GER	HUN	EUR	BEL	ITA	SIN	JPN	CHN	BRA
1	22	L. HAMILTON (GBR)	98	10	4		6	8	10			10	10	4	8	6	2	6		10	4
2	2	F. MASSA (BRA)	97			10	8	10	6	4	10		6		10	10	3		2	8	10
3	1	K. RAIKKONEN (FIN)	75	1	10	8	10	6			8	5	3	6					6	6	6
4	4	R. KUBICA (POL)	75		8	6	5	5	8	10	4		2	1	6	3	6		8	3	
5	5	F. ALONSO (ESP)	61	5	1			3			1	3		5		5	5	10	10	5	8
6	3	N. HEIDFELD (GER)	60	8	3	5		4		8		8	5			8	4	3		4	
7	23	H. KOVALAINEN (FIN)	53	4	6	4			1		5	4	4	10	5		8				2
8	15	S. VETTEL (GER)	35						4	1			1		3	4	10	4	3		5
9	11	J. TRULLI (ITA)	31		5	3	1			3	6	2		2	4				4		1
10	12	T. GLOCK (GER)	25							5				8	2			5		2	3
11	10	M. WEBBER (AUS)	21		2	2	4	2	5		3					1	1		1		
12	6	N. PIQUET (BRA)	19								2		8	3					5	1	
13	7	N. ROSBERG (GER)	17	6		1		1							1			8			
14	17	R. BARRICHELLO (BRA)	11						3	2		6									
15	8	K. NAKAJIMA (JPN)	9	3			2		2			1						1			
16	9	D. COULTHARD (GBR)	8							6								2			
17	14	S. BOURDAIS (FRA)	4	2												2					
18	16	J. BUTTON (GBR)	3				3														
19	21	G. FISICHELLA (ITA)																			
20	20	A. SUTIL (GER)																			
21	18	T. SATO (JPN)																			
22	19	A. DAVIDSON (GBR)																			

List of Companies Participating in Honda’s Third-Era Formula One Activities



List of Companies Participating in Honda's Third-Era Formula One Activities

Honda commenced its third era Formula One activities in 2000, competing in Formula One Grand Prix in collaboration with BAR. From 2006, we fielded our own team, and we were able to taste our long-anticipated first victory. We owe our ability to continue racing for a nine-year period during our third Formula One era to the generous assistance provided by the companies listed here, for which we offer them our heartfelt thanks.

We wish also to express our gratitude to all the staff members of these companies for their many tireless efforts on our behalf, to everyone at Honda and other companies who provided advice and assistance, and, finally, to everyone who offered us their warm support. We thank you.

A.L.M.T. Corp.

AEROSPACE METAL COMPOSITES Ltd

AJACOM CO.,LTD.

Akashi Communication CO.,LTD

ALCOA

ALTIMA Corp.

AN motorsport LTD

Arai Seisakusho Co.,Ltd.

Arisawa Manufacturing Co., Ltd.

ARTFAST

ASANO Co.,Ltd.

ASIA NETSUSYORI (in Japanese)

ASPECT Inc.

ATIA, Inc.

automax co.,ltd

AutoTechnicJapan co.,LTD

AVL Japan K.K.

BLANC AERO Technologies

Bodycote

Bridgestone Corporation

Busak+Shamban K.K.

Carpenter Specialty Alloys

CEROBEAR GmbH

CHIRIKA.Co.,ltd.

Chuo Spring Co.,Ltd.

CITIZEN MICRO CO.,LTD.

CMK CORPORATION

College Master Hands Inc.

COMATSU

CorporationYamazaki Setsubi.

Cosworth Electronics Ltd.

Cranfield University

CYAN Co.,Ltd.

Daido Castings Co.,Ltd.

DAIDO METAL CO.,LTD

Daido Steel Co., Ltd.

Daiei Barrel Co.,Ltd.

DALLARA AUTOMOBILI

DASSAULT SYSTEMES

DEL WEST EUROPE S.A.

DOME CO.,LTD.

DOWA HOLDINGS CO., LTD.

dSPACE Japan K.K.

Ebara Corporation

EMZLAB Inc.

ENAX,INC.

ENKEI CORPORATION

ERAMET INTERNATIONAL TOKYO BRANCH

FLOW SOLUTIONS LTD.

FUJI FURUKAWA ENGINEERING & CONSTRUCTION CO.,Ltd

FUJI KEISOKU SYSTEM Co. Ltd.

Fuji Manufacturing Co.,Ltd.

FujiStaff, Inc.

FUJITSU LIMITED

FUJIWARA CO.,LTD.

FUNAMIGUMI (in Japanese)

FURUNO ELECTRIC CO., LTD

GEO TECHNOLOGY S.A

GIKENSEIKI CO.,LTD

Global Active Technology Co.,Ltd.

GS Yuasa Corporation

HAGA KIGATA KANAGATA SEISAKUSHO CO.,LTD.

HEIWA SANGYO CO.,LTD.

HEPHAIST SEIKO CO.,LTD.

Hewlett-Packard Japan, Ltd.

HIDAKA SEIKI CO.,LTD.

HINODE-SEISAKUSHO CO,LTD

HIRAI CO.,LTD.

Hitachi Advanced Digital, Inc

Hitachi, Ltd.

Hitachi Metals,Ltd.

Hitachi Zosen Corporation

Hokuriku Light Metals Industry

HONDA ENGINEERING CO.,LTD

Honda Foundry Co.,Ltd.

Honda Kaihatsu Co.,Ltd.

HONDA PERFORMANCE DEVELOPMENT INC

Honda Trading Corporation

Hoshino Corporation

IBM Japan, Ltd.

Ichikawa.co.,ltd.

ICS Corporation

IHI Corporation

Illinois Tool Works Inc.

IMV CORPORATION

INCS INC.

INOMATA KOGYO (in Japanese)

IPEC co.ltd

Ishihara Precision,Inc.

ISHIKAWA DIE & MOLD CO.,LTD

IHI INSPECTION & INSTRUMENTATION CO.,LTD

Ishimoto Architectural & Engineering Firm, Inc.

ISHIZU EG co.,ltd

IWAHARA Sangyou Co.Ltd.

JEFTEC CO.,LTD.

JFE Steel Corporation

JTEKT Corporation

Kakamigahara Aero Equipment Co.,Ltd

Kandenko Company, Limited

KAWAGUCHI YUKI Co.,Ltd

KDDI CORPORATION

Keihin Corporation

Kenko Factory CO.,LTD

Kinzoku Giken Co,.Ltd.

Kistler Japan Co., Ltd.

Kitahara Shoji Co.Ltd

Kobe Material Testing Laboratory Co.,Ltd.

KOBE STEEL,. LTD

Kobelco Research Institute, Inc.

KOGANEI SEIKI CO.,LTD.

KOUKEN CO.,LTD

Kowa ss CO.,LTD.

ktel.co

KURISAKI GEAR MFG.CO.,LTD

KUWANA.,co.ltd.

KYB Corporation

KYOWA ELECTRONIC INSTRUMENTS CO., LTD.

KYOWA INDUSTRIAL CO.,LTD.

LMS INTERNATIONAL N.V.

M.T.S

MagCanica,Inc.

Magnesium Elektron Laboratory

MAGNETI MARELLI S.P.A.

MAHLE (Vandervell)

MAHLE ENGINE SYSTEMS UK LTD.

Märkisches Werk Japan K.K.

MARUBENI INFORMATION SYSTEMS CO., LTD.

MARUI-KEIKI CO.,LTD.

MEIDENSHA CORPORATION

MEIRA Corporation

METALORE, INC.

MICHELIN

MITEC CO.,LTD.

MITSUBISHI PLASTICS,INC

MITSUBISHI RESEARCH INSTITUTE, INC.

MITSUBISHI STEEL MFG. Co.,Ltd.

MITSUMORI SEISAKUSYO (in Japanese)

MIYAMA CORPORATION

Miyamoto Industry Co.,Ltd.

MIYASAKA RUBBER.,CO.LTD.

MORIKAWA INDUSTRIES CORPORATION

Morimura Bros.,Inc.

MORIYA CORPORATION.

M-TEC Co.Ltd

MTS Systems Corporation.

MW RACING GMBH

NAGANO KEIKI CO.LTD.

Nagase chemtex corporation

NDK,Incorporated

NEC TOHOKU.LTD.

NGK SPARK PLUG CO.LTD.

NHK SPRING CO.,LTD.

Nichiei Co.Ltd.

NIHON DENKEI CO.,LTD.

nippon ITF inc.

Nihon Mekki Industry Co., Ltd.

Nihon Unisys,Ltd.

Nippon Oil Corporation

NIPPON SEIKI CO.,LTD

NISHIKAWA KEISOKU .,CO.LTD.

NISSIN KOGYO Co.,Ltd.

NISSIN MANUFACTURING CO.,LTD.

Nissin Travel Service Co., Ltd.

NITTAN VALVE CO.,LTD.

NITTO DENKO CORPORATION

Nittobo Acoustic Engineering Co., Ltd.

NOK CORPORATION

NSK Ltd.

NTN CORPORATION

OHMIYANISSIN CO.,LTD

OHNISHI NETSUGAKU Co.,Ltd.

ONO SOKKI CO.,LTD.

OSHIDA SEIKI CO.,LTD.

Pankl Racing Systems AG

PARKER NETSUSHORI KOGYO CO.,LTD

PEARL GIKEN CO.,LTD.

PHIARO CORPORATION

PLANSEE Japan Ltd.

PLANSEE SE

PSG

QUALICA Inc.

RDS CO.,LTD.

RESS CONSULTING GMBH

Retrac Composite Ltd

ROHM CO., LTD.

ROKI Co., Ltd.

Saint-Gobain K.K.

SAITAMA SHATAI CO.,LTD.

SAKAMOTO KYORITSU SEIKI Co.,Ltd

SANKYO DENNKI SYOKAI (in Japanese)

Sankyou International Corp

Sansho Shoji Co., Ltd

SATO RASHI CO.,INC

SEKI DENNSETSU KOGYO (in Japanese)

SEKIDAI KOGYO CO.,LTD

SHIBAURA ELECTRONICS CO.,LTD.

SHINKO ENGINEERING CO., LTD.

Shin-Etsu Chemical Co.,Ltd

SHINRYO CORPORATION

Shinto Industrial Co.,Ltd.

Shizuki Electric Co.,Inc.

Shonan Design Co., Ltd.

SHOWA CORPORATION

Showa Denko K.K.

Showa Electric Laborator Co.,Ltd.

Showa Seisakusyo Co.,Ltd.

SINFONIA TECHNOLOGY CO., LTD.

SINTOKOGIO,LTD. SINTO BLASTEC CO.

Sony Manufacturing Systems Corporation

SOURIAU Japan K.K.

SOYO GOMU Co.,Ltd.

STANLEY ELECTRIC CO.,LTD.

Sugihara Software & Electron Industry

SUM Electro Mechanics Co.,LTD.

Sumico Lubricant Co.,Ltd.

Sumitomo 3M Limited

Sumitomo Corporation

Sumitomo Electric Industries, Ltd

SUMITOMO LIGHT METAL INDUSTRIES, LTD

Sumitomo Metal Technology Inc.

Sumitomo Wiring Systems, Ltd.

Super Resin, Inc.

SURFCOAT CO.,LTD

Suzuki Kinzoku Kougyou Co.,Ltd.

SUZUKI PRECION Co,Ltd

T.RAD Co.,Ltd.

Taikisha Ltd.

TAISEI CORPORATION

TAKASAGO LTD.

Takasago Thermal Engineering Co., Ltd.

TAKATA CORPORATION

TAMAGAWA SEIKI CO.,LTD.

TANAKA SEIMITSU KOGYO Co.,Ltd

TDK Corporation

Techno-Core Corporation

TECHNOPRO Engineering, Inc.

TEIKOKU PISTON RING Co.,Ltd

Tigers Polymer Corporation

TIMET

TNO-Automotive Japan K.K.

TOAMEC Inc.

TODA CORPORATION

Tohoku Steel Co., Ltd.

TOKAI SEIKI CO,Ltd.

TOKIWA CO.,LTD.

TOKOROZAWA ALLOY FOUNDRY CO.,LTD.

TOKYO GASKET CO.,LTD.

Tokyo Institute of Technology Graduate School of Science and Engineering

Tokyo Metal-Pack Co.,Ltd.

Tokyo R&D Co.,LTD.

Tokyo Sokki Kenkyujo Co.,Ltd.

TOKYO.FERROMITE.INDUSTRY.CO.LTD

TONETS CORPORATION

TOSEI ELECTROBEAM COMPANY LIMITED

TOSHIBA CORPORATION

TOYAMA PREFECTURAL UNIVERSITY Department of Intelligent Systems Design Engineering

TOYO Corporation

TOYO DENSO CO.,LTD.

TOYO iTEC Co., Ltd.

TOYO KOKU DENSHI Co.,Ltd.

Trelleborg Sealing Solutions Japan K.K.

Tsukasa Sokken Co., Ltd.

UBE Scientific Analysis .Inc

UCHIDA Co.,LTD

UCHINO SEISAKUSHO Co.,LTD.

UESAWA WORKS,INC.

Vector & Scalar Products Limited (VSPL)

Wave Front Co.,Ltd

Westmoreland Mechanical Testing & Research, Ltd

Watanabe Architects & Engineers co., Ltd

YACHIYO-SEIKI Ltd.

YAGISHITA GIKEN CO.,LTD.

Yamada Manufacturing Co., Ltd.

Yamaguchi University Graduate School of Science and Engineering

Yamaguchi University Faculty of Agriculture

YAMASAN CO., LTD

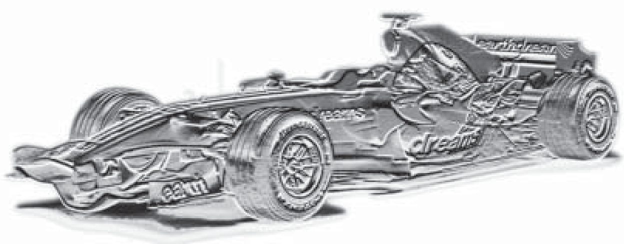
YAMASHITA RUBBER CO.,LTD.

YAMATO GOKIN

Yokohama National University Graduate School of Engineering/ Faculty of Engineering

Yuei Co., Ltd.

Comments from Project Members of Honda's Third-Era Formula One Activities



I entered the company (MSD at the time) in ’99 and could witness almost all of Honda Formula One Third era activities. It is good experience and memory for me that I participated in challenging duties within the limited time and shared emotions with many people in the ten years.

Daisuke Aoki

Test in Barcelona from November 17-19, 2008.

For me, it was the first running test as a data engineer.

It was also the last running test for the Honda Formula One Third era activities.

To be honest, I wanted to continue participating in the work since I’ve become long-awaited data engineer.

Kenji Akimoto

I’ve been taking charge of general management of technologies as ALPL of KERS. I feel proud that we could complete a competitive unit from the first year of the regulation owing to strenuous efforts of all project members and local staffs. I really enjoyed the job. I’d like to express my deep gratitude to the management members that awarded me such a chance, project members who pushed themselves to the max to achieve target performance and schedule and comrades who supported us in life in the UK and Germany. Thank you very much.

Hirofumi Atarashi

HRD Machining Div. members are local staffs and have been conducting training on site until they become able to perform the same level machining as HGT. HRD is now able to manufacture original manufacturing and I realize that we have contributed to making HRD independent. Associates on the spot/actual parts are the starting point to win sympathy of people who are different in cultural background and the way of thinking. I intend to make use of the experience in teaching junior fellows.

Toshikazu Arai

The Honda Formula One Third era activities developed me since when I entered the company. I met excellent people and gained wonderful experience in an environment which seemed to have embodied the challenging spirit. I don’t think I was happy; I couldn’t be happier. Let me say thank you and looking forward to come back some day to the stage of Formula One.

Yoshimi Ishii

My dream is to win the championship with an F1 racing car I constructed. I entered the company in 2008 and was assigned to the MS in October of the year but, immediately after that, Honda decided to withdraw from Formula One. I felt frustrated losing the opportunity to directly participate in the Honda Formula One Third era activities. I’d like to realize my dream in the fourth era activities never forgetting the vexation.

Yasuharu Ichikohara

I participated in the development of F1 engine valve drive system materials as well as DLC and Ti materials for three years and the state-of-the-art technologies including the surface form control technology. I cannot forget the emotion even now I experienced at the victory of Button in Hungary GP in ’06. I believe that Honda will come back to the stage of Formula One in the near future.

Naoaki Ito

I entered the company in April 2008 and assigned to the MS in October to participate in the development of F1 engine but, to my regret, Honda decided to withdraw from Formula One just when I was highly motivated becoming an F1 engineer which is my dream. Nevertheless, I’m proud of the experience of participating in Honda Formula One Third era activities if only for a short period of time.

Yuki Ito

I devoted myself to development making positive progress every day. I cannot forget, however, the days I passed in vexation unable to achieve results. I had to make efforts more and yet more. I believe that Honda will come back to the stage of Formula One mainly supported by junior staffs if we make use of the frustration as leverage for the next development.

Nobuyuki Imai

Our objective is simple and clear: to “win the race”.

In retrospect, I found myself always pursuing higher power. We held conviction to carry through all by our own unaided efforts and the spirit of engine developer never to give up whether beaten or have broken up.

However, the Honda Formula One Third era race events were not as simple as CART races.

I’d like to spring back stronger from the vexation of finishing without driving to victory.

Yoshiyuki Ugomori

It was the moment that my dream from the days of my earliest recollection was realized when I became a member of the Honda Formula One Third era activities. Feeding on the experience and the memory, I intend to vigorously push forward to realize a new dream. Lastly, I’d like to say a word to me in the past, “Yes, you did it! One of your BIG DREAM from early COME TRUE !!!”

Tsubasa Uchida

The experience of the challenging work in the development of F1 racing car I desired to participate in from the beginning when I entered Honda has become a precious knowledge of my life. I thank everyone who supported me in the chance and would like to engage in everything with the spirit of “Never give up!” also in future.

Mamoru Uraki

Dream in my boyhood days; ambitions; engine designer; chance; joy; BAR HONDA; energetic work; expectation; excitement; eagerness; accumulation; hardships; ALL HONDA; race track experience; world; culture; victory; delight; setback; challenging; struggle; chagrin; growth; bro; thanks for all.

Hajime Endo

I participated in Honda Formula One Third era project for about six years in the area of aerodynamics development. I acquired precious experiences beyond count and it was the greatest moment in my life when Honda won a victory with “FOX EAR” of my personal contrivance installed on the racing car.

Tsuyoshi Okubo

I participated in the development of F1 engine for eight years from when I entered the company. I remember well that I found it very tough physically at the beginning unable to catch up with the pace of work. The experience in Formula One is very precious having not only built up the base of work for me but also taught me importance of collaborating with comrades.

Yasuhiro Okura

We introduced high-strength material Mg casting/thin-wall Al vacuum casting and all-out high-quality casting utilizing X-ray CT scanning into actual race cars for ENG weight-saving. I’d like to continue to vigorously practice those matters I experienced in the high-technology manufacturing in the future work.

Yuuji Ooshima

I committed for three years from 2000 to the job of supporting in those F1 staffs that are sent overseas on business.

I was given a job in the race building in the middle of the years. I remember that I became glad and sad by turns in front of TV feeling Formula One closer to me and realizing that I’m a member of the team. Though I only had a little might, I’m proud that we fought together and felt sympathy.

Ryosuke Otsuka

With a simple objective to win the race, I could participate in the development of the unit giving priority to the performance.

As the result, we could get through to application of the heat spreader having excellent heat conductivity to racing cars.

Masatoshi Okumura

About five years and three months from 2003 when I entered the company and assignment until withdrawal from Formula One have passed in just about not time like a dream. I met and cooperated with various people through development, Test, Race events acquiring a lot of knowledge, techniques and experiences. I’d like to thank fans who gave me a lot of deep emotion.

Toru Ozawa

We achieved “zero” trouble in all race events overcoming difficulties in introducing Honda’s first original GBX into actual racing by organizational strength and technological competency and speed. We are proud of pursuit of high-precision finishing technology which is underlying the seamless shift and the luster of light-weight final gear. I’d like to leverage in the next job the sense of fulfillment/accomplishment I got in Formula One activities.

Shin Odagiri

Not to let defective products we manufactured outflow, that was the duty of us participating in F1 activities and was worth doing. Not to mention the processing precision, we were to solve problems one by one with respect to the surface scratches and dents to achieve high precision assurance. While it was difficult to achieve both quality and schedule, I intend to make use of the experience for supplying manufactured products in the future.

Tokio Kasai

I participated in the development of electrical components from 2001 to 2008.

I believe that the experience of being engaged upon the work toward clear and inexcusable objective to complete the development and let the racing car get in the grid by the first racing event in March next year hold clues also for developing mass-produced cars at present.

Toshikazu Karube

While it was a work of the greatest difficulty to complete development within such a short period of time as a year and a half, we succeeded to put the world’s first HEV F1 racing car on the racing track as the result of all persons involved in KERS including team members and HRF1 engineers who became single-minded in the efforts. I’d like to express my respect in particular to Mr. Dunkan Elliott of HRF1 and Miss Ohno who took charge of translation into English. Thank you very much everyone.

Masato Kita

I acquired precious experiences actually feeling the severity of racing and meeting excellent technicians and persons. I’m determined to catch up with competitors and overtake them in ten years dreaming a day when Honda makes a strong impression on people all over the world and win the championship with overwhelming technology.

Hiroshi Kimura

While it was a short period of time, the experience of learning many things among capable technicians is my wealth. In addition, I’m proud of having participated in F1 race events with excellent comrades.

Tatsuya Kyomitsu

Personally, I regret that the part I was developing would be introduced into F1 racing cars if Honda's withdrawal was decided a year later. Including this, I feel afresh that it was a racing.

That is, race against time, fight for performance and competition against ourselves.

I'm very much grateful that I could have precious experience on the stage which was my childhood ambition.

Yoeru Kono

I participated in F1 activities for about 10 years in the G/Box and ENG areas. I could have a precious experience to directly feel Honda DNA through my skin. I'd like to express my gratitude to comrades with whom I worked together and business partners who supported us despite the peculiar field of racing. I have my heart on becoming No. 1 in the Formula One Fourth era activities.

Daiki Konagaya

I watched F1 race event on TV in 2000 for the first time to love and yarn for Formula One and entered HONDA. The ten years of Honda Formula One Third era activities is precious for me encouraging me with stuff of dreams and is the period of time when my dream to be engaged in Formula One was realized. But I have another dream yet. That is to become a Honda Formula One Fourth era activities engineer!

Megumi Kobayashi

In the several years during which I worked near the F1 activities staffs, I didn't like Formula One but actually felt the biggest pleasure in working to make arrangements for those staffs who are striving for the objective in F1 with the utmost energy. I was much encouraged by them.

Thank you very much.

Yuichi Koyanagi

I have been engaged in the development of fuel injection system for about three years from 2005 when I entered the company. Though it is regrettable that Honda made the decision to withdraw from Formula One when we were working toward the '09 season, I remember well that I was deeply moved when we won a precious victory in Hungary.

Yohei Sakai

While it was such a short period time as two months from when I was assigned to the MS Materials, I gained very dense experience. What I felt in then is that it is important to interact with the staffs of other areas not shutting ourselves up in the work area we are commissioned. I'm determined to apply the experience in the future work.

Katsutoshi Sakurai

The sudden withdrawal from Formula One was really shocking for me.

I applied for the recruitment of Formula One activity staffs and transferred from Sayama Plant in April 2006 to participate in the parts management job for about three years.

I learned many things while making many fluffs through F1 activities. Thank you for all your trouble. The experience of participating in the F1 activities, very pinnacle of the motor sport, is lifelong treasure of me.

Satoshi Sasaki

While it was only a year and three months from when I was transferred to come to the MS as a KERS staff until Honda's withdrawal from Formula One, I learned many things from bench test to circuit running test was being supported by many people. I intend to devote myself so that I can make use of the experience for the next work.

Kinji Sato

We've been pursuing wear resistant material for and technology to reduce diameter of the valve guide for the valve drive system and introduced them in actual race events assessing productivity of them aiming at reliability assurance of ultra-high speed ENG. I intend to make use of the approach I acquired through F1 activities in various fields even in difficult manufacturing processes. I'd like to express my deep gratitude for that I could participate in Honda Formula One Third era activities.

Masakatsu Satoh

I wanted a little more time and it's regrettable that we couldn't achieve results. I intend to devote myself to everyday operations taking motive power from with the regret for the fourth era activities. While it was such a short period of time as a year and a half, I'm proud of having participated in the F1 activities desiring to by myself and I'd like to express my gratitude for the chance given to me.

Makoto Sawada

We hit a losing streak and withdrew from F1 in the end losing an opportunity for recovering lost ground. We were disgraced which will last me a lifetime. I'm now on the stage of development for mass-production. We are not allowed to play second fiddle in the technical field. I intend to see if there is any way in which I can be of help for becoming peerless.

Yosuke Sawada

I watched Formula One for the first time when I was the first grade of an elementary school when Maclaren Honda with Senna was flowering. I was a high school student when Honda made a comeback to Formula One in 2000. And I was a college student when Takuma Sato was taking an active part and Honda won a victory for the first time in the Honda Formula One Third era activities. I entered Honda in 2008. It was two months after I was assigned to the F1 engine design team of the Honda R & D that Honda decided to withdraw from Formula One. I feel regret but I'm looking forward to the fourth era activities.

Yuki Shigeta

I was transferred to the MS in January 2008 and I only worked just about a year for the F1 activities. I feel I did nothing but to learn the work in the F1 activities. Nevertheless, I expect that the experience in promoting development giving priority to speed and anticipating the future and how to have selections at an early stage will become my wealth for the future.

Kentaro Shimada

I enjoyed very much participating in the F1 activities which is my childhood dream. While it is regrettable to withdraw from Formula One without winning the championship, the experience of the vertiginous busyness in tests and the feeling of tension on the site of racing event and in particular that I was on the spot when Honda won a victory in Hungary are unforgettable memories.

Takashi Shimada

It was on August 30, 2007 that I was entrusted with MS_BL which may best be described by the expression "thunderbolt from a clear sky". I received a telephone call communicating me a business order that I should be Operating Officer from September 1. That was from the day after the next day. A year and a half since then had elapsed before I knew it. In "Barcelona" where we got points for the first time, I felt so excited as if my heart was leaping into a mouth. And that was only for the sixth place. I wondered how I should be if we were competing for victory. While we couldn't achieve good results the year, our competitive power was surely improved inviting Ross Brawn. It's very regrettable that we no longer have a chance to deliver the product next year. I'd like to finish taking operational responsibility in the MS expecting everybody to take an active part to get your own way that much, I believe you will naturally.

Katsuhiko Suzuki

I got stimulated competing on the world stage. I understood well at which point my ability is. I still have a lot of things I want to do though I have to leave without fulfilling my life ambition. I feel contented that I could work in such an environment that makes me have such feeling. I'd like to gain more victories next time.

Shiroh Takasaki

I applied for the project recruiting staffs who wish to participate in the F1 activities and was assigned as a member of the Racing Development Div. in October 2001. From then, I participated in the LIC Racing Car Components Sec. for seven years. While the section was taking charge of works doubly behind the scenes, we insatiably pursued how to respond to real-time to information and materials flow that changes every hour. Regrettably, Honda decided to withdraw from the F1 activities before startup of the fastest physical distribution system, I learned through the system development and physical distribution promotion works that "victory can only be achieved on confidential relationship and collaboration between people". I'm looking forward to the fourth era activities.

Ken Takahashi

The suddenly started and suddenly ended ten years have elapsed in just about no time.

I was made re-realize the importance of improving individual technologies at the same time the profundity of driving a car at a high speed and racing where people compete through the experiences that we couldn't win only with individual good technologies.

It is regrettable that I have to leave F1 activities without knowing everything of difficulties and fascinating aspects of Formula One.

Hiromasa Tanaka

Participating as an engine designer, I leaned the difficulty of making design being conscious of the "Number one in the world". I also had very precious experience of coming in touch with new attitude of the different idea/viewpoint unique to the EU.

Katsuyoshi Chubachi

I competed for F1 World Championship in one place after another as an engine technology staff of the Super Aguri F1 Team. I'm proud of getting a place by an all Japan team (team with nationality of Japan, Japanese team representative, Japanese driver, Japan-made tires and Japan-made engine). Thank you very much for your cheering.

Yoshito Tsukamoto

I'd like to thank our business partners who supported our development activities and staffs of other divisions and sections who always willingly aided us in consultation.

We could commit to development to the full thanks to the intensively allocated resources. We couldn't be happier as engineers.

Masayuki Tsuchiya

I was very happy that I could work on the childhood dream stage. While the withdrawal of this time is regrettable, I'm proud of the experience of participating in Formula One and I'm determined to do what I can do to the full extent of my power. I'd like to thank those fans who have supported us until today. See you in the fourth era activities.

Hiroki Tsuboi

I participated in the development of KERS. While it was such a short period of time as no more than a year and a half, I'd like to express my gratitude to all those who supported me for enabling me to have very dense experience. I'd like to make the experience I gained more valuable making best use of it for activities in the future. Thank you very much.

Masaya Deguchi

I played a part to the best of my ability for F1 technologies as a race car electrical component engineer.

I came to participate in supporting SAF1 in 2006 to witness the F1 technologies and felt much of the Three Reality Principle.

I intend to make efforts to reflect the technologies and the spirit I cultivated in the F1 activities in everyday life in the near future.

Akiko Tokimoto

I took charge of the ENG casting dies and could make significant contribution to improvement in efficiency/improvement in initial quality of casting by unification of design/manufacturing 3D data by CATIAV5 and complete application of casting coagulation CAE. I'm proud of this and I'd like to make its use for the job in the future.

Yousuke Tokiwa

Taking charge of casting core for F1 engine, I could introduce new core technologies such as the closed block between narrow shafts introduced in EUR GP in 2002 and the complex core cylinder head introduced later. In the Honda Formula One Third era activities, I could experience challenge spirit of never give-up and ideas becoming reality/actual product.

Masanori Totsuka

While I only participated in KERS development for 17 months, it was a happy experience to have participated in the development until it is almost completed for introduction into race event in '09 performing development with unprecedented density feeling passions of members who wish to avoid being defeated in the hybrid technology aiming at number one in the world.

Yasuhiro Toyoda

I could participate in the aerodynamics development for Formula One I desired to from when I entered the company and finally realized to reside in HRF. It was very precious time in my life.

If, some day, Honda challenges Formula One again, I want to participate in it again.

Yusuke Nakai

I was given a chance to take charge of engine design and gear box design on the supreme stage of Formula One to compete the number one in the world which is my childhood dream. I'd like to thank those who supported me for letting me have the very precious experience. Thank you very much.

Yasuo Nakagawa

I participated in the development of F1 car gear box for about seven years from 2002.

To say nothing of durability and preciseness of F1 car components and high-level control program, I learned various matters including development speed and processes until introduction to race events.

I'll never forget the excitement we got when we overcame difficulties.

Hiroshige Nagata

My name is Teruoki Naganuma. I took charge of the development of gears. While we couldn't achieve results in the Honda Formula One Third era activities, I'm glad that the results we achieved are shared by way of this paper. Personally, I was very happy that I could participate in Formula One which had been my adoration since my childhood. I'd like to express my heartfelt gratitude to the comrades of the F1 team.

Teruoki Naganuma

In 2002, when I was a J.V Race engineer, Jacques Joseph Charles Villeneuve mocked that HONDA ENG is not a horse but a donkey in his comment on the ENG at that time after running as I remember.

While we tried new items such as Rotary Throttle at the time to introduce to the race car, we couldn't achieve good results and we of the Honda R & D and actual running members had very tough time.

Spring back from the vexation, we succeeded to introduce a new ENG in the middle of the season which made Jacques not use the word Donkey, and the ENG was developed into one without problem both in the Power and DR. While we couldn't achieve results in race events, we of the ENG producers take pride in that even drivers recognized the result of taking on challenges.

Satoshi Nakamura

I participated in manufacturing the exhaust pipe referred to as a craftwork in the Honda Formula One Third era activities. We felt pleasure and accomplishment supporting the development by solving number of challenges and realizing technologies including creation of alternative plan, joggling/bulkheading/down-sized construction by might and main to improve ENG performance aiming at pulsation effect. I intend to make use of the developed EXPI and cultivated K/H in the future.

Takumi Nakayama

In the F1 activities, I experienced on site various works which were very beneficiary to me. While I experienced both hard times and enjoyments, those works proved worthwhile and enabled me to build self-confidence. I believe that the experience on site will be of use for work in the future.

Katsura Nariai

I could take on various challenges participating in F1 activities for five years from September ’03.

In the night in Suzuka in ’04 after calm down from the excitement of the race event, there had happened a sudden HONDA call from the main stand. We got stimulation/support from passion/expectations of fans we of all members directly felt going out to the race track.

Yuji Niihara

Finishing the development of CART ENG in 2002, I joined together with the F1 development team. I committed to the development of cladding type valve seat and completed the work supported by the team, the Prototype Sec., EG and subcontractors. I felt special accomplishment when the part was put into practical use taking pains.

Hiroaki Nishida

Honestly speaking, visionary prototype V8 engine stands out in my memory. The engine, put on ’04 chassis by force ahead of regulation modification in ’06, was the first compliant ENG in the world as with Honda’s KERS. While I’m going to be transferred to the GT Div., I intend to continue to pursue becoming the “First to arrive”.

Ken Nishimori

I pride myself on participating in the F1 activities, if indirectly, as a facility infrastructure exploratory member from the inauguration of the HRD in ’99 until the HRD & F integration plan in ’09 and it gave me a precious experience. In addition, I feel I’m lucky as a facility management personnel as the facility I introduced is producing results still now. I intend to respond to this by handing down the precious experience to the posterity when Honda makes comeback to the stage of Formula One.

Masamichi Hagiwara

“Participating in the F1 racing car development job for five years from the time I entered the company, I’m grateful to the Honda Formula One Third era activities that gave me various precious experiences which cannot be obtained other than through racing including the unique feeling of tension, importance to carry through to the end there and pleasure of watching the race car running with the part I developed installed.”

Junichi Hashimoto

I participated in the Prototype Group from ’03. I’m proud of being able to do parts delivery with confidence assuring the quality. I could hang in there with utmost effort despite the tight schedule owing to the comrades who were doing their best on the circuit and many people who supported us. I’d like to extend my thanks to Jenson Button who won a victory at which I became teary-eyed as well as to the Honda Formula One Third era activities.

Yoshihisa Hayakawa

I’m grateful to Honda Formula One racing activities which subjected me to various and precious experiences.

Takuji Hiroma

I’m grateful to a month I passed in the MS meeting capable associates whom I should have in my sight and precious experiences I had in the course of my work. I’m looking forward to a day when HONDA makes comeback to Formula One. I am going to make every single effort to better myself everyday so that I will be able to run about with an MSD seal affixed to my PC.

Gakuyo Fujimoto

There are things which can only be understood by being on the spot and cannot be understood only by watching from outside. Whether it is a good thing or bad thing, everything will enrich your body and soul in your later life. I’d like to make full use of what I gained in the future life. HONDA RACING SPIRITS passes for at any time and place. I wish good luck to my comrades and I will do my best myself.

Ryuichi Furukawa

While we couldn’t achieve results worth the name of Honda, I was happy being able to work as an engineer in Formula One, my personal dream. I’d like to express my heartfelt thanks to all in and out of the company who supported me. I want to realize subsequence of the dream some day with new experiences I will have in the future.

Yasutaka Masumitsu

I took charge of F1 engine valve drive system in 2003, 2005 and 2006 committing to racing engine since when I entered the company. Particularly in 2005, Honda introduced a new valve drive system for the first time in the past 13 years to largely reduce weight of the ENG HEAD as well as to enhance the power and the ENG was appreciated as the mightiest engine in the world of Formula One in the last year of V10 engines and in which I take pride.

Hirotake Matsubara

I'm very pleased to have participated in the Formula One project for six years from autumn 2002 and have passed the time with good comrades. The network I constructed with many people of the indirect divisions, Body Manufacturing Div., Materials Handling Div., Electrical Equipment Div. and Prototype Div. I started from device development for sensing high-speed world and I got various experiences including examining buildings for the job. I was also moved by dedicated efforts on the part of engineers working behind the scene to feel the sense of pleasure and accomplishments. I believe that if Honda makes come back to run at full speed on F1 track, we can further make great strides making use of the experiences gained in the Honda Formula One Third era activities.

Keiji Miura

Hollow connecting rod which was developed as a result that the words "box shape is best suited for light-weight and high-strength" inspired the manufacturer spirit. We introduced the component in '03 season race events after spending about two years. Then, we also committed to hollow crank. We had almost completed the development of the component when junction was banned and it was abandoned before introduction to actual race cars. The pick off by the regulations is the proof that our technology was recognized by the world.

Ken Mizogawa

In the ten years when I participated in the Honda Formula One Third era activities, I experienced many joys and difficulties of taking on a challenge and have greatly grown up as an engineer.

I'd like to thank all people who supported Honda up to now.

Powering dreams one lap at a time

Sadami Minato

In the busy days where racing events don't wait for us, we passed hearing BGM of high-pitched sound from the bench. When I got positively turbulent in the circuit PIT, I got back to myself glued to Formula One in my boyhood (The Power of Dreams). I'm very proud of having been able to participate in the F1 activities as an employee of Honda. I'd like to win a victory to make it the wealth of my life.

Takahiro Minezaki

I was very sorry when I heard the news of withdrawal from the F1 activities. I was so delighted when Honda won the first victory in the Honda Formula One Third era activities in the 13th race event Hungary GP in 2006 that I shouted loud in front of TV of which I have a distinct memory.

Hiroyuki Miyake

In the Honda Formula One Third era activities, we supported enormous progress of ENG by element development and manufacturing at fierce speed in response to various changes to regulations. The experience, among others, of having contributed to the development of apparitional '09 NEW ENG with the maximum speed and quality and passion is a very precious experience which can be made use of for the future.

Kazufusa Miyajima

Technology development in the F1 activities was really a competition in a state-of-the-art area of the world. I will feed back the precious technologies I studied in the F1 hybrid material development to mass-produced cars commencing with INSIGHT in the future.

Takehiro Miyoshi

It must be seldom that we become so vexed or happy that we feel like crying. In the "F1 activities that Honda joined in competition advocating development of young engineers", I could experience both. I'm grateful to Honda for giving me such a chance.

Shigeki Morie

I learned importance of personal initiative and responsibility, ability to take action taking initiative and giving output on a timely basis from senior associates who have high motivation never saying "No" granted that they may say "Yes, but".

Since my objective and intention toward technical development must be invariable, I intend to make use of them for the future.

Akihiro Yanase

The Honda Formula One Third era activities were actually global competition for material engineers. We rivaled global competitors to keep on evolving component specs making full use of advanced material technologies. I learned how it is important to continue to take on a challenge in race events.

I'd like to repay the favor by handing down the Honda DNA I have taken over from Mr. Matsubara and senior associates to junior fellows.

Hiroshi Yamada

It was such a short period of time as a little less than a year from January to December 2008 that I took charge of making arrangements for overseas business trip for F1 activity staffs.

The period of a year when I felt Formula One familiar to me though I was ignorant to Formula One in the past was all fresh to me and I have become interested in Formula One. Thank you very much.

Mari Yamada

I fully understood what work is, what a car is and what racing is. What I learned here will flow in my future life like vein. Hopefully, the water will blast out again into the field of Formula One. I'm grateful to the precious days.

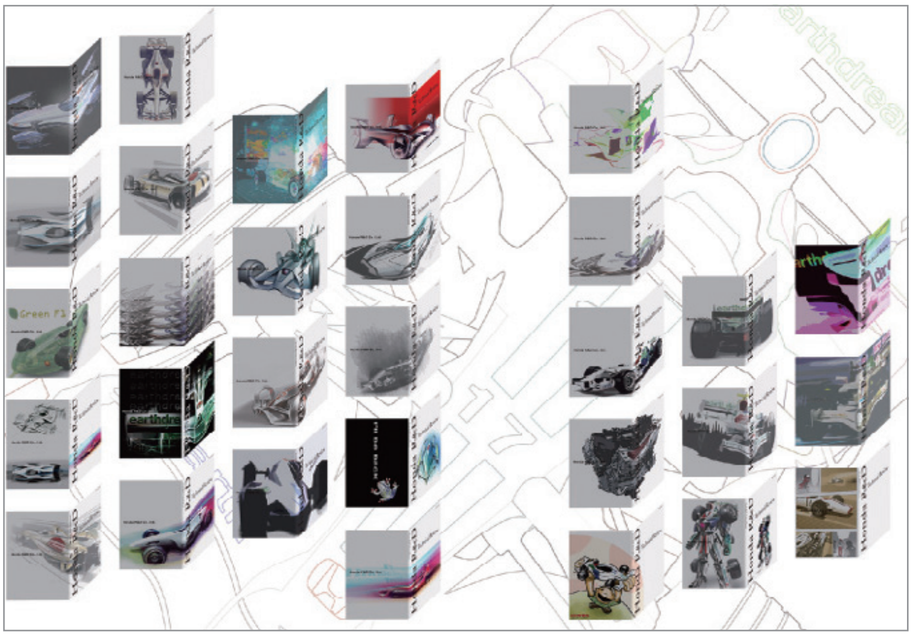
Ryozo Yamamoto

I applied for the project recruiting personnel who wish to participate in the F1 activities and was assigned to the Racing Assembly Div. transferred from the Service Div. of the Head Office. I found there that the racing ENG had remarkable quality and the race site harshness was out of ordinary. Instead, the sense of fulfillment and the contentment gained by carrying through those difficulties have invaluable allurement. The experience and the memory at the time that those who have passion gained working hard together will never fade away.

Makoto Yoshino

[Glossary]

ENG	Engine
GBX , G/BX	Gearbox
TH	Throttle
HEAD	Cylinder Head
EXPI	Exhaust Pipe
K/H	Know-How
J.V	Jacques Villeneuve
MSD, MS, MS_BL	Motor Sports Division
HRF	HONDA Racing F1 Team home base
Facility control expert	Facility Control Section expert
LIC	Parts Control Division
EG	Honda Engineering Co., Ltd.



Comment on the Cover

Honda Designers' F1 Images, from the Heart

It's a Honda tradition to put your heart and mind into your work, which motivates us and makes work more satisfying.
We may face demanding but satisfying tasks every day that call for fresh thinking, and of course, F1 presented many such challenges. Honda F1 has excited and inspired a lot of people, and our designers all over the world have poured their hearts into cover illustrations in commemoration. In turn, the images portray a tradition people have poured their hearts and minds into, thrillingly beautiful for many years past and many years to come.

Shigeo Ueno

Designers who drew the cover illustrations

Honda R&D Co., Ltd.
Honda R&D Technical Paper Committee
Automobile R&D Center
Editorial Committee for F1 Special (The 3rd Era Activities)

Afterword

Almost a year has passed since Honda suddenly announced its withdrawal from Formula One, and we commenced planning this special issue of the Technical Review. An older colleague once told me that being guaranteed a position in the company and being able to participate in race development was great good fortune for an engineer; for me, being involved in race development and being able to bring together the outcomes of those projects as an editor has been an even more precious experience.

Today, the gulf between race technologies and mass-production technologies is pointed out to be great in comparison to Honda’s first and second Formula One eras. While their applications and effects may not immediately be apparent, the technologies that we produce by reaching for the ultimate will be useful and valuable in some way to the future, and I will be happy if their recording here in this form can show this to be the case.

On the final note, I would like to take this space to remember our friends and colleagues who, to our sadness, passed away during the time that they participated in Honda’s Formula One project: Tomohiro Kumagai, who was involved in the development of both hardware and software for vehicle bodies and gear boxes; Tadasu Takahashi, who was involved in engine research based on track running and carried out research on gear boxes; Kenji Nakashima, who was involved in engine research; and Masanori Wakasa, who was involved in the development of control systems. This volume is dedicated to their memory.

(Hiromasa Tanaka)

Honda R&D Technical Review

Published by:
Honda Motor Co., Ltd.
2-1-1, Minami-Aoyama, Minato-ku,
Tokyo 107-8556, Japan

Edited by:
Honda R&D Co., Ltd.
Honda R&D Technical Paper Committee

Printed:
Dec. 1, 2009


Honda R&D Co., Ltd.
Honda R&D
Technical Review
Honda R&D Co., Ltd.
Green F1
earthdre
I'll be back