
Technical Description of Formula One Engine Structural Design

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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.

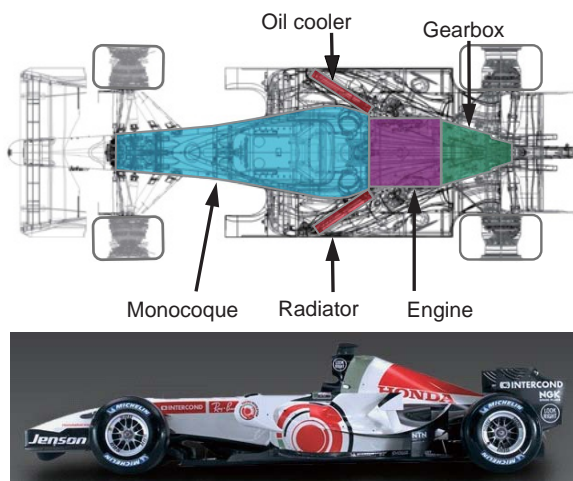


Fig. 1 F1 chassis and powertrain configuration

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

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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				

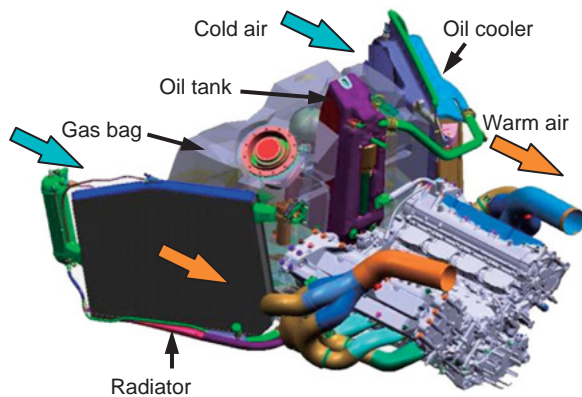


Fig. 2 Engine installation with cooling system

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

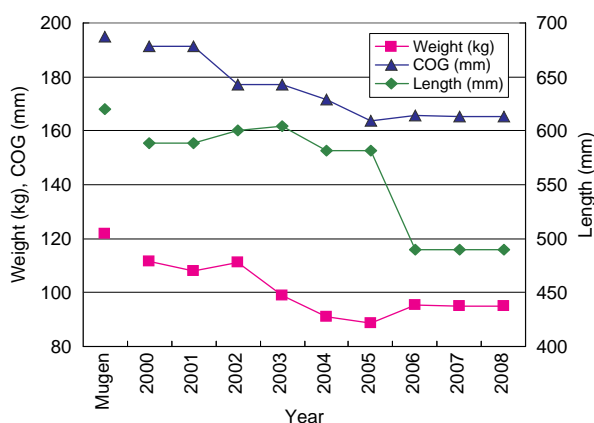


Fig. 3 Change in engine length, weight and COG

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 $\phi 95$ in 2000 to $\phi 97$ in 2002 during the third era of Honda Formula One. $\phi 97$ was an optimal setting for engine speed of 19000 rpm. All engine constructors appear to have used bore diameters of between $\phi 97$ and $\phi 98$ 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.

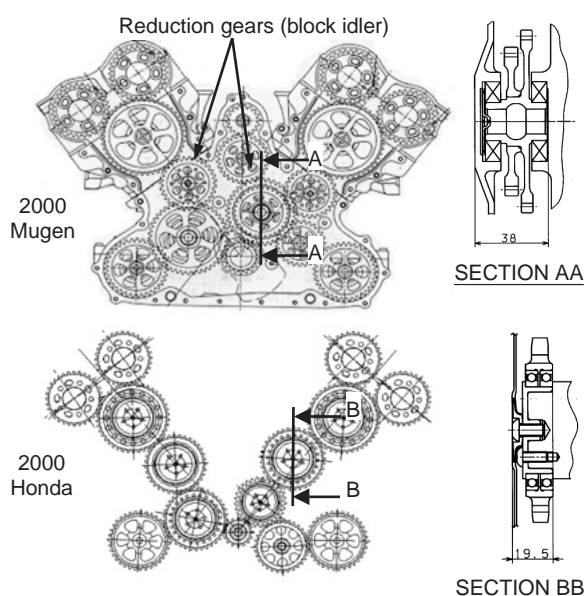


Fig. 5 Gear train configuration

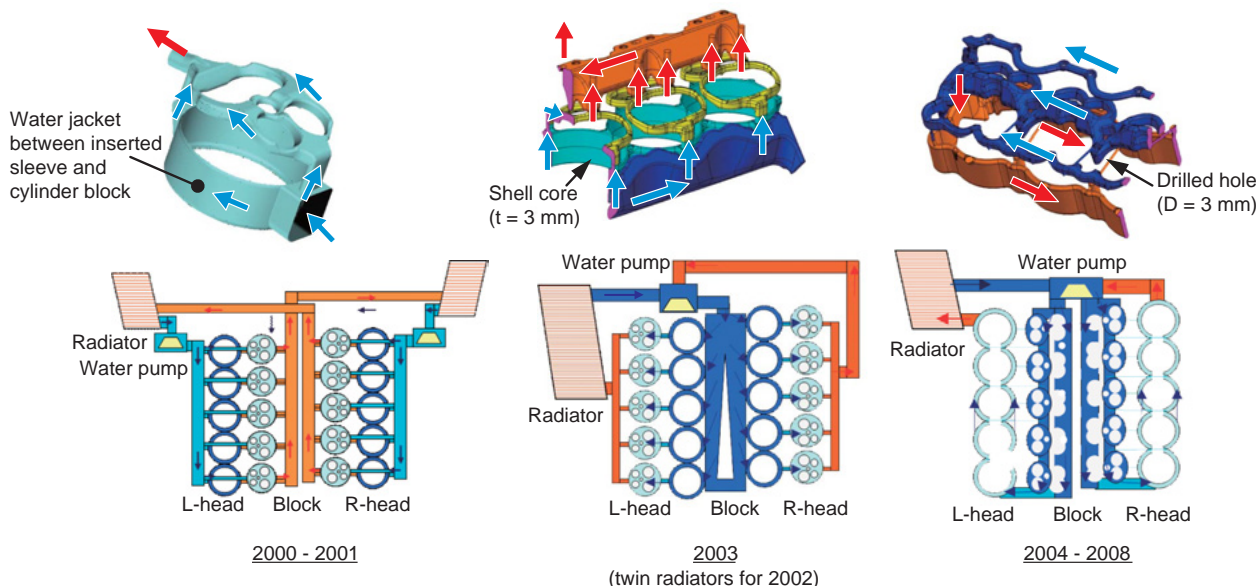


Fig. 4 Schematics of water cooling systems

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 $\phi 40.6$ to $\phi 41.6$, and in the case of the exhaust valves, from $\phi 31.8$ to $\phi 32.6$), 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.

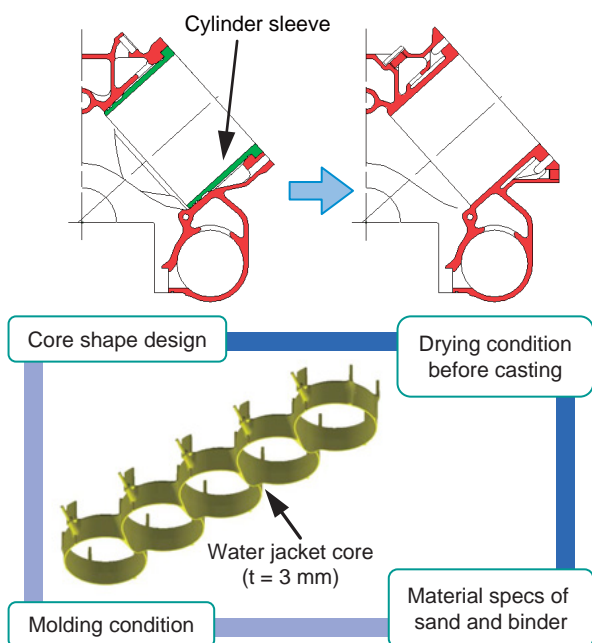


Fig. 6 Closed deck cylinder block and water jacket core

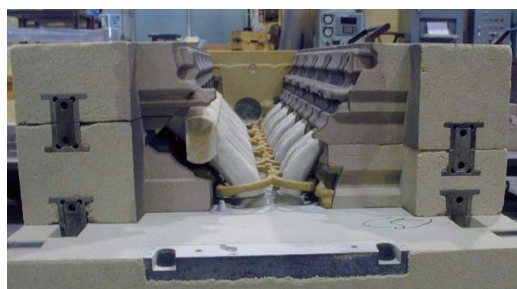
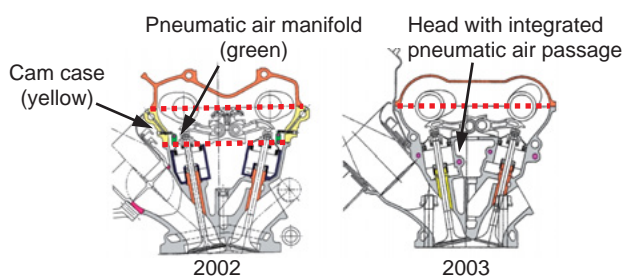


Fig. 7 Cylinder head with integrated PVRS

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).

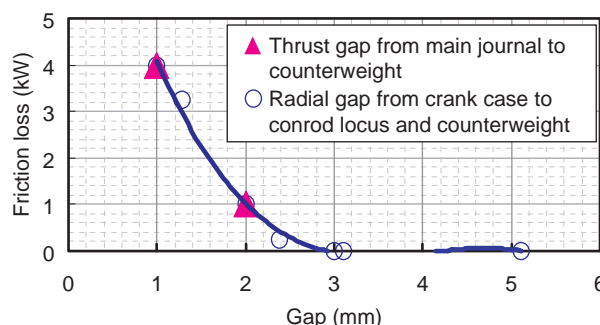


Fig. 8 Friction loss against closest gap

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

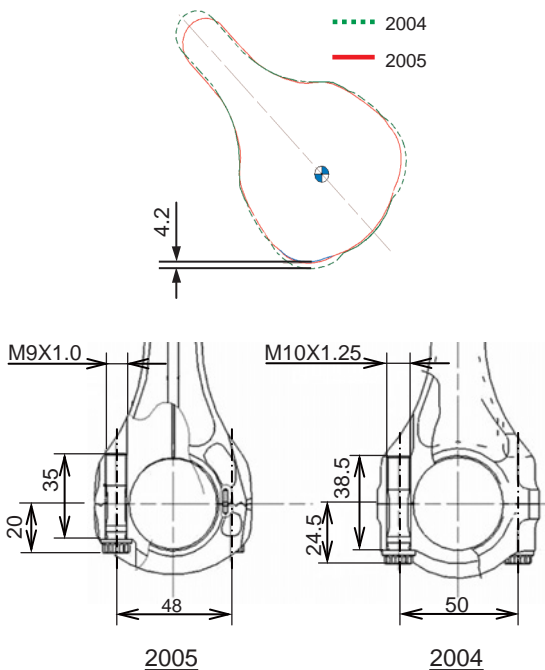


Fig. 9 Reduction of conrod locus

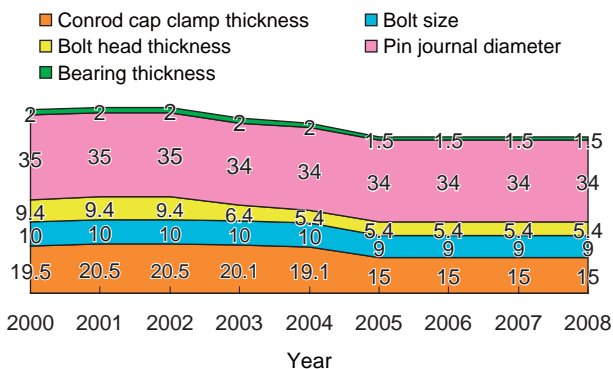


Fig. 10 Progress of conrod configuration

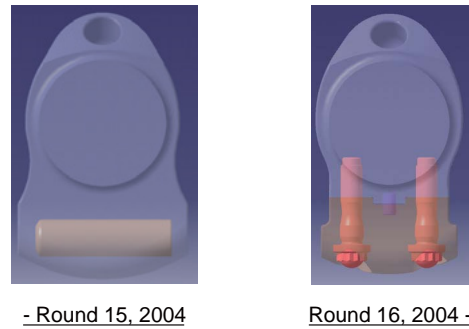


Fig. 11 Schematics of crankshaft counterweights



Fig. 12 Lateral bend calculation

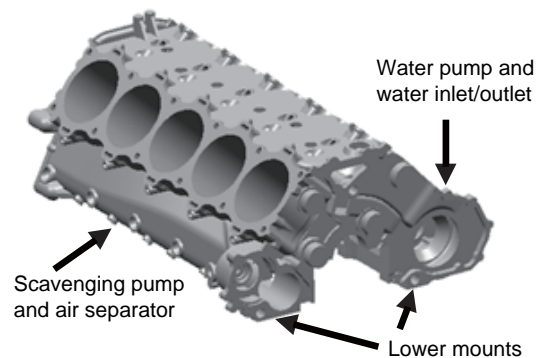


Fig. 13 Cylinder block with integrated oil/water pump

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

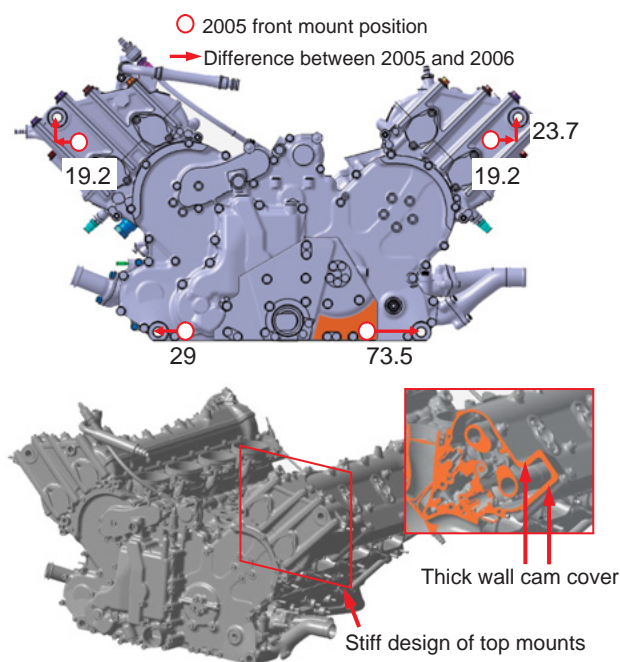


Fig. 14 2006 engine characteristics for increased stiffness

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

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.

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 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

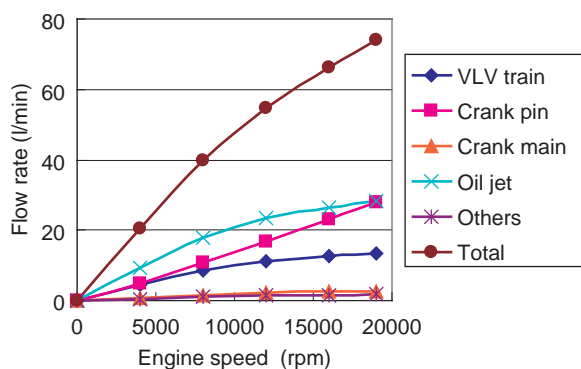


Fig. 15 Oil distribution

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 $\phi 12$ 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 $\phi 8$ 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)

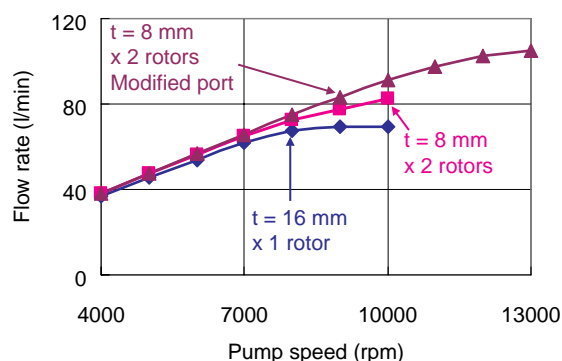


Fig. 17 Effect of twin rotor and modified port shape

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.

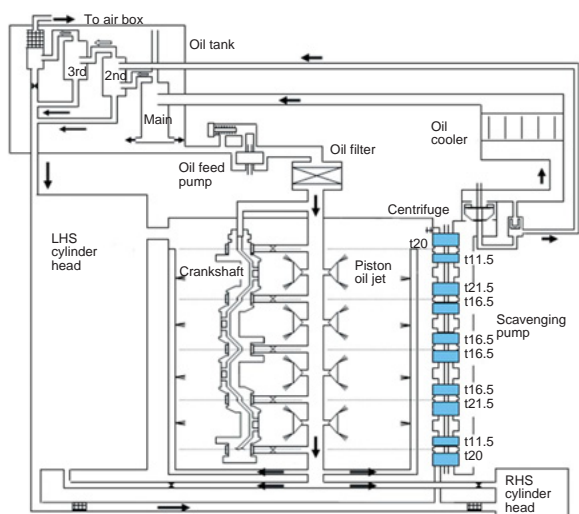


Fig. 18 Oil system

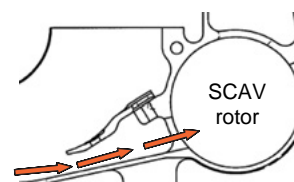


Fig. 19 Straight suction

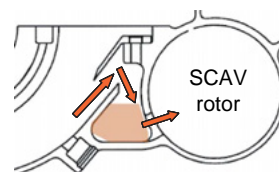


Fig. 20 Trap-type suction

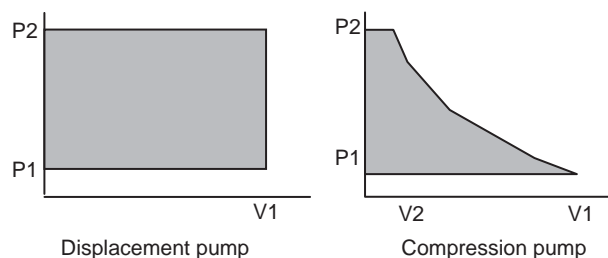


Fig. 21 Comparison of pump work

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

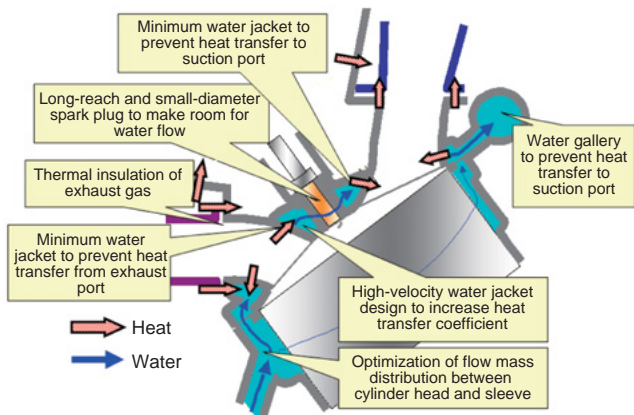


Fig. 22 Design concept of water jacket

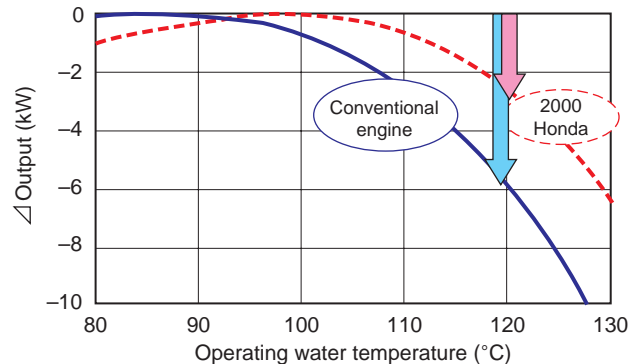


Fig. 23 Effect of water temperature on engine performance

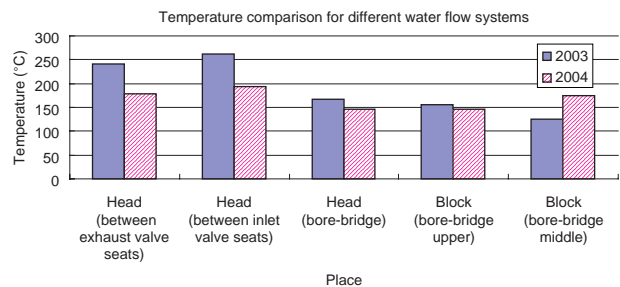


Fig. 24 Temperature comparison between 2003 and 2004

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.

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