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

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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.

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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	

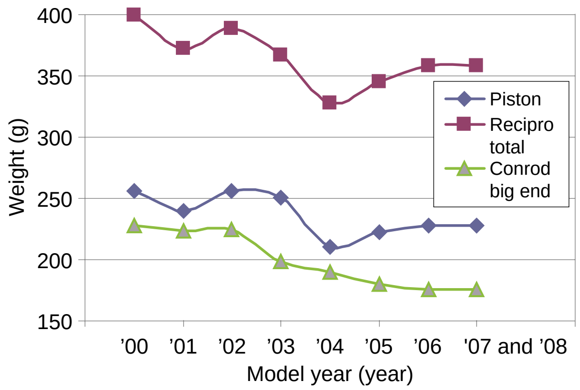

Fig. 1　Reciprocating parts weight history

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

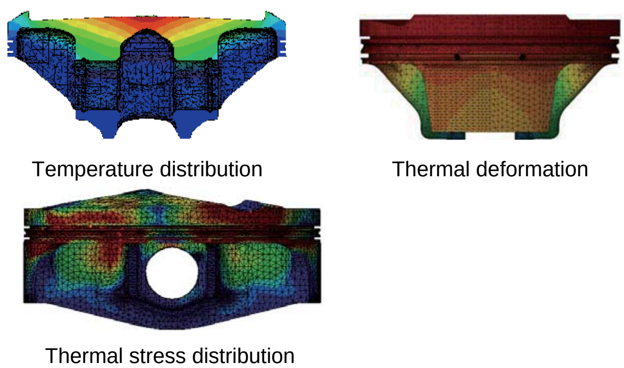

Fig. 2　Examples of piston CAE results

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

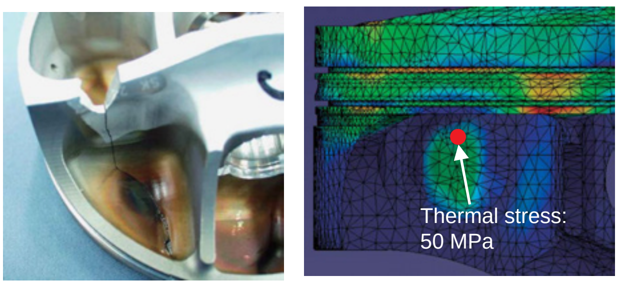

Fig. 4 Crack due to thermal stress in thin area of rib

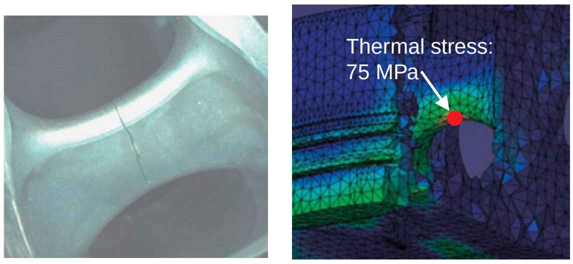

Fig. 5 Crack due to thermal stress in bridge rib

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.

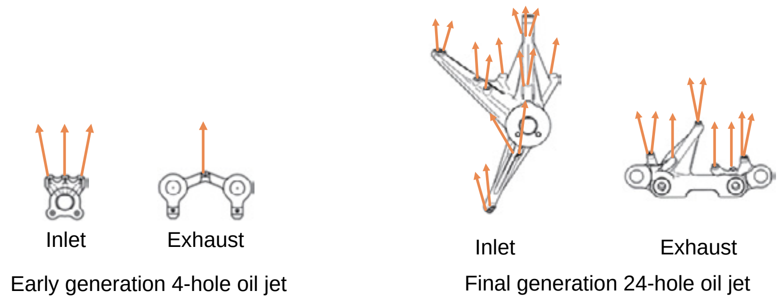

Fig. 6 Evolution of oil jets

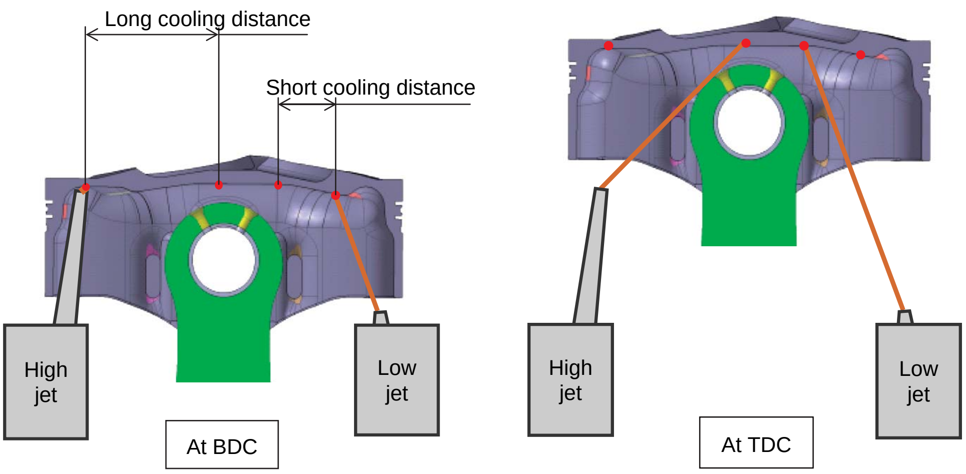

Fig. 7 Efficiency of high oil jet

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

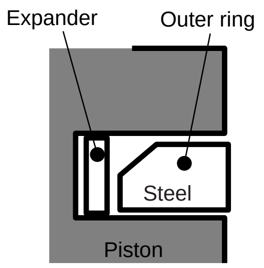

Fig. 8 Expanded piston ring

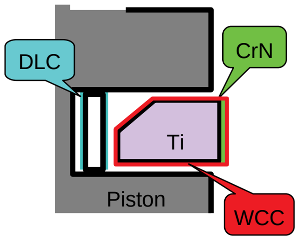

Fig. 9 Titanium expanded piston ring

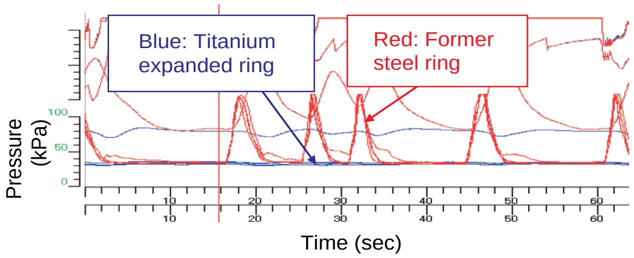

Fig. 10 Rise of crank case pressure due to piston ring fluttering

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.

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.

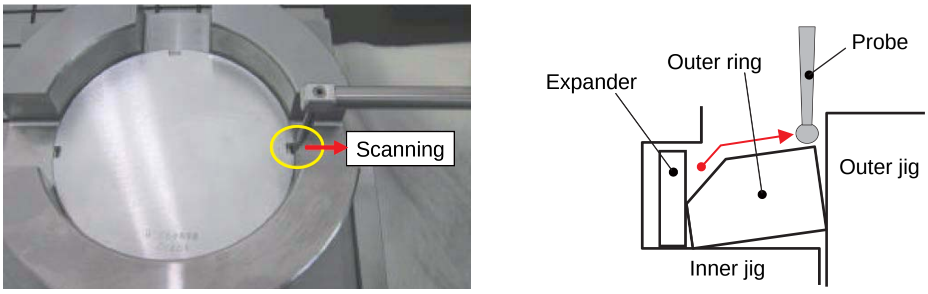

Fig. 11 Expanded ring taper angle measurement jig

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.

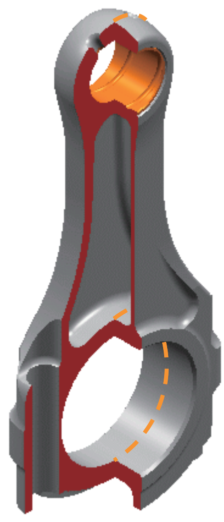

Fig. 13 Hollow conrod

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

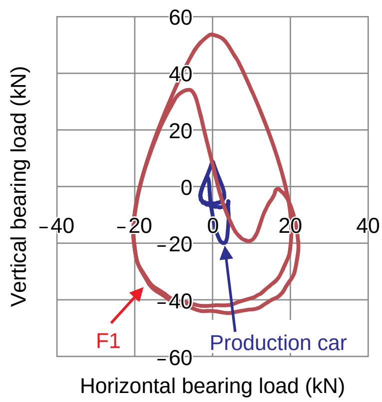

Fig. 14 Conrod bearing loads at WOT, Ps peak revolution

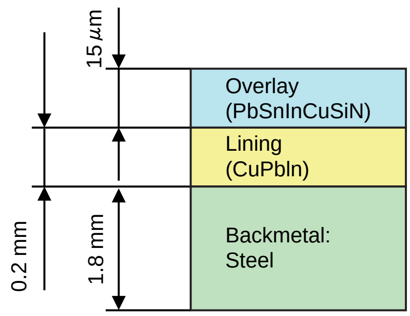

Fig. 15 Former bearing layer

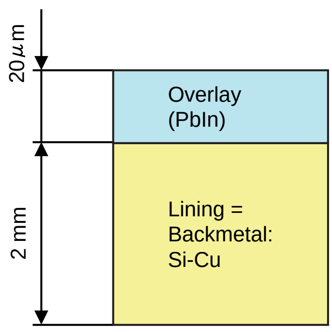

Fig. 16 Si-Cu bearing layer

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

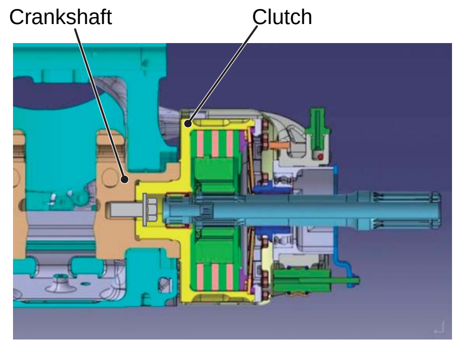

Fig. 17 Fastened clutch

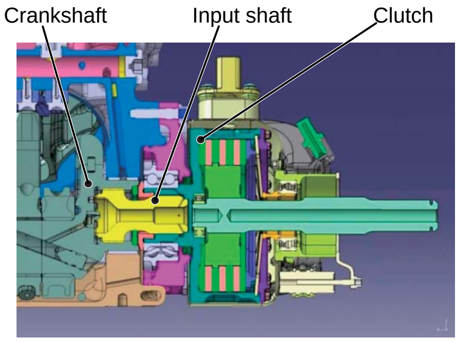

Fig. 18 Transmission with input shaft clutch

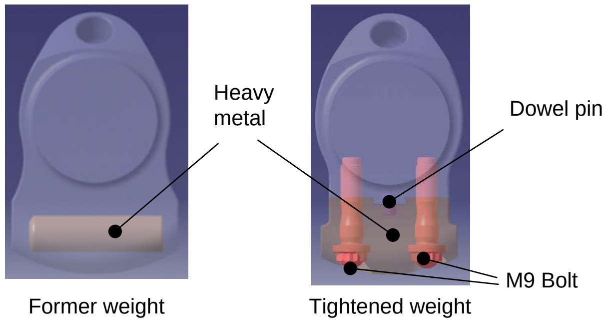

Fig. 19 Crankshaft counterweight history

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.

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.

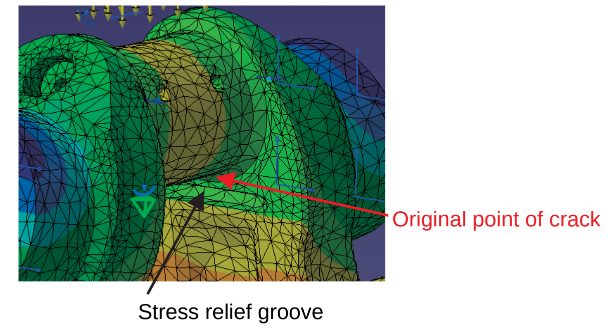

Fig. 20 Crankshaft stress relief groove

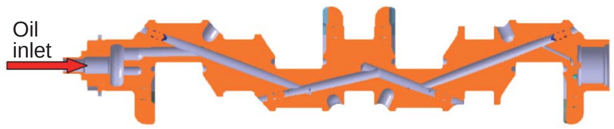

Fig. 21 Crankshaft center oil supply passage

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

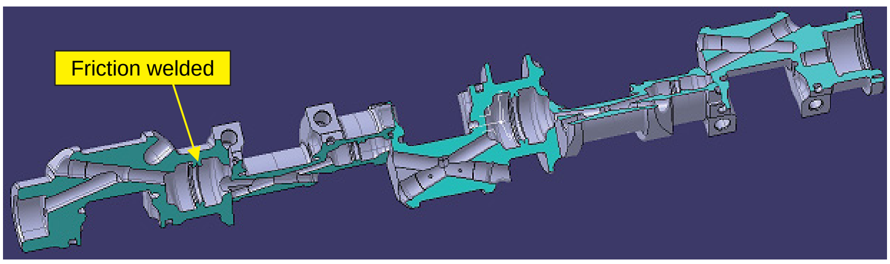

Fig. 22 Hollow crankshaft

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.

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