

Development of V10 500-HP Engine – Technologies for High Performance and Reliability –

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ABSTRACT

This article introduces technology related to the power output and reliability of the V10 500-HP engine developed for use in super sports cars. In order to achieve high power exceeding 500 HP, thoroughgoing measures were taken to reduce interference of intake air and exhaust air between cylinders, and laser clad welded valve seats were adopted. Individual throttles were adopted for the purpose of high response, which is of similar importance to power output for sports cars, and comprehensive weight reductions were made in major moving parts. As to reliability assurances, an examination was made of how the successor vehicle to the NSX, which was slated to have this engine mounted in it, would be used. Test criteria were formulated on the basis of actual running data to enable issuance of a guarantee in all areas, including running on the circuit, and testing was carried out. The above measures produced an engine that would allow enjoyment of running with the high degree of attractiveness set as an objective and with a sense of security in all situations.

1. Introduction

The principal technologies in the V10 500-HP engine are being introduced in two parts⁽¹⁾. This article will describe the technologies for high performance and reliability.

A super sports car realizes its true value when it runs on the circuit at speeds so high that it is in a different dimension from ordinary automobiles. This engine, which had been slated for mounting in the NSX successor car, had to have both the high power output and the high response of a racing engine in order to provide that kind of pleasure in running. As a high-performance sports car engine, this engine also required a conceptual approach and technologies different from an ordinary car for its reliability guarantee. The following will introduce those technologies.

2. Power Performance

2.1. Overview of High-performance Technology

Figure 1 shows the engine performance characteristics. Table 1 shows the major specifications.

Enhancing intake and exhaust efficiency, enhancing thermal efficiency, and reducing friction are important in achieving high performance. The technologies adopted with this engine for these purposes will be introduced, in that order, below.

Figure 2 shows the overall layout of this engine's intake and exhaust system. The intake system was divided into independent left and right banks in order to reduce interference between cylinders. The intake chamber volume was increased to obtain fully adequate capacity that would not cause diminished power output, because the adoption of individual throttles helped resolve the issue of a reduction

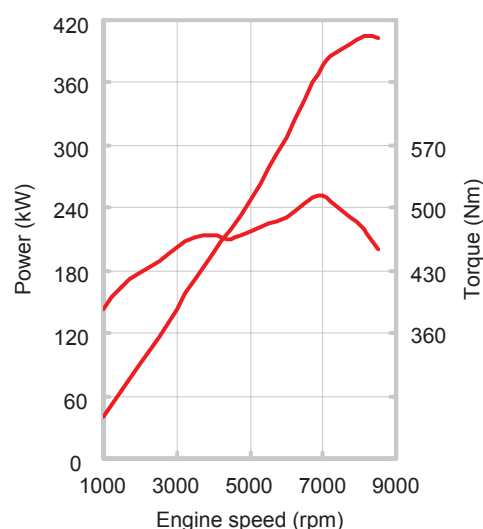


Fig. 1 Engine performance

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Table 1 Engine performance specifications

Max. power (kW/rpm)		405 / 8200
Max. torque (Nm/rpm)		513 / 7000
Displacement (cm ³)		4993
Bore × stroke (mm)		90 × 78.5
Compression ratio		12.3:1
Main part	Valve (mm)	Inlet: $\phi 37 \times 2$ Exhaust: $\phi 31.5 \times 2$
	Piston & Ring	Forged & 3 Ring
	Connecting rod	Titanium
	VTC	Operation angle: 50 degrees
Air intake system	Surge tank	Product made of resin (vol: 9.4 L × 2) Each bank independent
	TH/B : Pri	$\phi 38 \times 2$
	TH/B : Sec	$\phi 50 \times 10$
Exhaust system	Exhaust manifold	$\phi 38.1 \times 440$ Thickness: 1.0-1.5
	C.C.	2BED (0.4+0.6 L) 600/1.2 metal +600/3.5 ceramic
	Exhaust pipe-A	$\phi 60.5$ Thickness: 1.2
	U.F.	$\phi 118$ race track 1.4L 300/1.2 metal
	Exhaust pipe-B	$\phi 65$ - Exhaust chamber $\phi 115$ Each bank independent

in response when the capacity was increased. The exhaust system was also divided into independent left and right banks in order to reduce interference between cylinders. An exhaust manifold of the long 5-1 type was adopted, and where the five pipes come together, a stainless steel chamber made by lost-wax casting was installed to reduce interference between cylinders due to the exhaust. This is the GT chamber shown in Fig. 3. In addition to these means used to reduce interference between cylinders in the intake and exhaust system, laser clad welded valve seats were adopted and the shape of the area around the intake port valve seats was refined, further enhancing intake and exhaust efficiency.

Thermal efficiency was enhanced by the use of spark plugs with fine center electrodes and aluminum sleeves, and

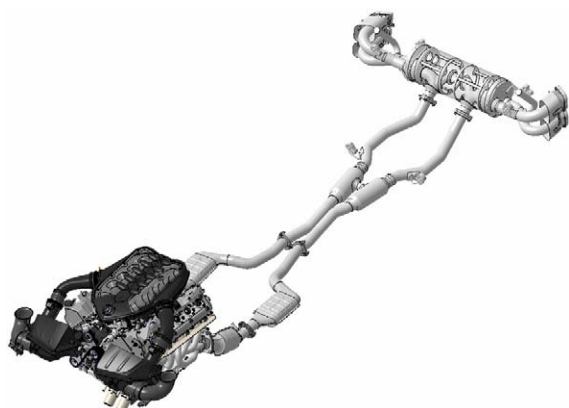


Fig. 2 Engine inlet and exhaust L/O

shape optimization of the combustion chambers and water jackets achieved a compression ratio of 12.3:1.

Measures taken to reduce friction include extensive reduction in the weight of major moving parts together with refinement of passages between the cylinders in the cylinder block and of the layout of oil return passages from the cylinder heads to the crankcase.

2.2. Laser Clad Welded Valve Seat (LCVS)

One of the main technologies for enhancement of engine performance is the laser clad welded valve seat (LCVS), and this technology will be explained here. The valve seats were formed by welding powder metal, and for this reason, the LCVS⁽²⁾ used in Honda F1 engines were able to have the seat material occupy a smaller area around the valve seat location than in conventional press-fit seats, allowing increase of the valve diameter. In this engine, a 0.7 mm increase in the diameter was realized.

With press-fit seats, it is most common that the connection between the rotary processed shape of the valve seating area of the valve seat and the cast port shape was performed by rotary processing. With the Honda F1 engine using the LCVS, on the other hand, the entire assembly from the valve seating area of the valve seat to the port could be continuously worked for optimization in three dimensions, providing a stable port with a high flow coefficient. In the present engine, the processing that connects the rotary processed shape of the valve seating area of the valve seat on the intake side and the cast port shape is the three-dimensional funnel shape shown in Fig. 4, which was intended to provide stable enhancement of the flow coefficient.

Funnel-shaped processing together with expansion of the valve diameter by the adoption of LCVS yielded a confirmed 18-kW enhancement in power output.

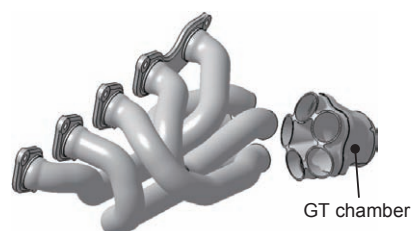


Fig. 3 Exhaust manifold and GT chamber

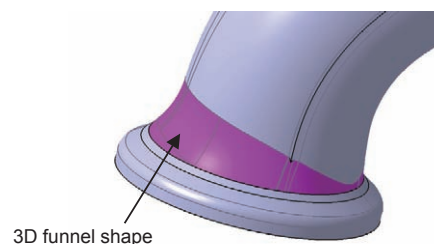


Fig. 4 Funnel shape of LCVS

3. Technology to Enhance Response

Although one of the conditions for running with greater speed on the circuit is to have high performance, it is also important to have positive response whereby engine speed increases and decreases according to the opening or closing of the throttle (TH). It has been conventional in racing engines, therefore, to employ an individual TH in order to enhance response by having a small air intake volume downstream from the TH valve. This engine also employs such a mechanism. Figure 5 shows the results for response performance measured under snap acceleration with no load.

Having individual TH can enhance response, as described above. However, this can also make it more challenging to control small volumes of intake air in the low-flow region, and differences in valve openings at individual TH can tend to result in differences in the air fuel ratio between those cylinders. Differences in air fuel

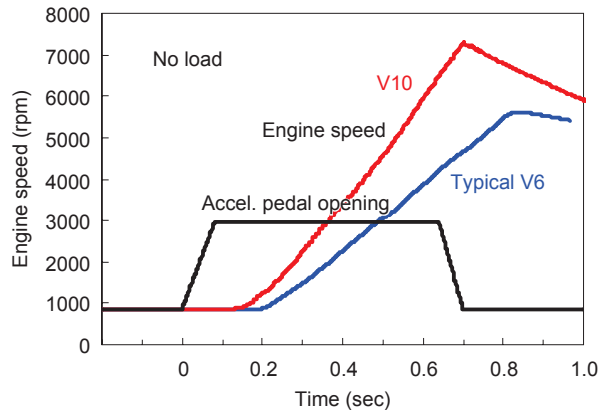


Fig. 5 Engine response

ratio can make it more complicated to conform to exhaust emissions controls. The type of configuration shown in the diagram in Fig. 6 was therefore employed to limit those differences, and this system will be described below.

- (1) The individual TH is used primarily to control air intake volume in the high-flow region, and it was designated the secondary TH.
- (2) Control of air intake volume in the low-flow region, including idling, is performed by a single TH placed on each bank of five cylinders in order to reduce air fuel ratio differences between cylinders. This was designated the primary TH, and there is one installed on each bank.
- (3) Purge and PCV circuit are installed only on Bank 1, where cylinder deactivation does not occur⁽¹⁾.

Figure 7(a) shows a model of parts on the Bank 1 side as actually designed for the above configuration. Similar

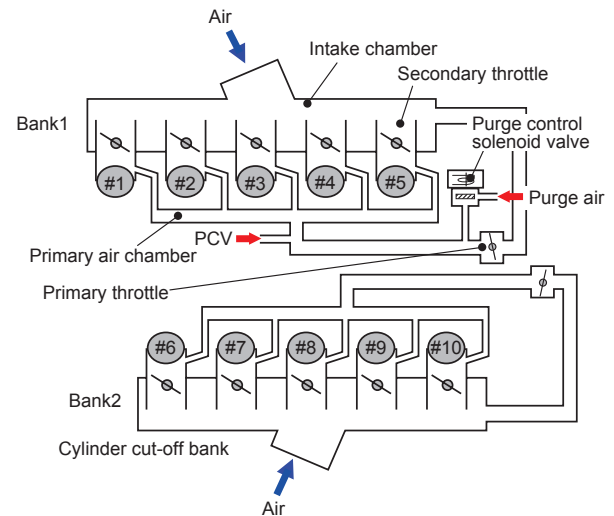


Fig. 6 Schematic diagram of throttle system

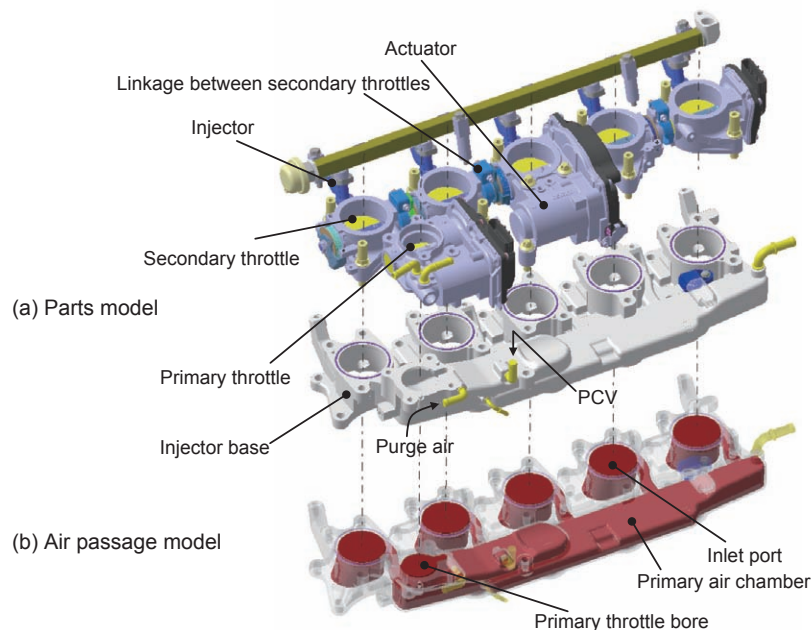


Fig. 7 3D model of throttle system

parts were installed symmetrically to these on the Bank 2 side. Details of this design are explained in the following.

(1) Primary TH

With a bore diameter of 38 mm, this was designed for a maximum flow of 1450 L/min, which should largely cover general driving in the US. As shown in Fig. 8, the system is configured so that the secondary TH does not open until the primary TH is fully open. This greatly limited air fuel ratio differences between cylinders during ordinary driving.

(2) Secondary TH

The independent TH body is bolted to the injector base (INJ base). The actuator that operates the TH is located in the center bore. A TH and its adjoining TH are joined by links.

The TH in each cylinder is fitted with a return spring. This is designed so that if the return mechanism malfunctions, the TH will return to the fully closed position.

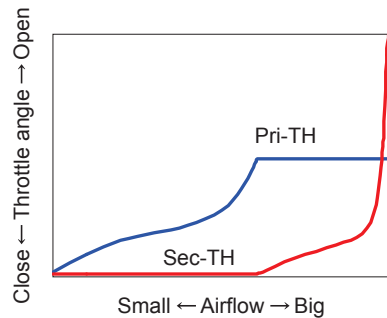


Fig. 8 Throttle open pattern

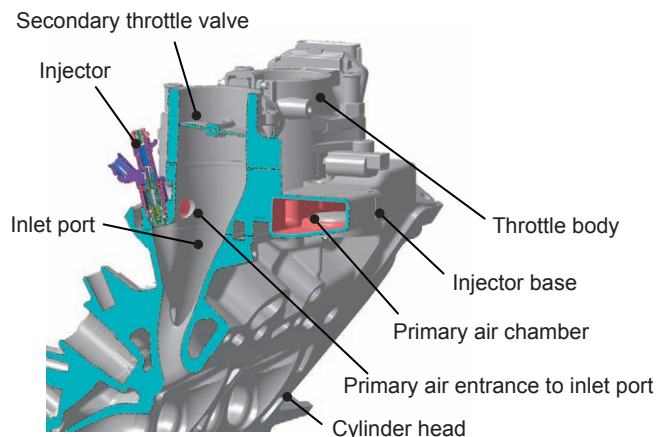


Fig. 9 Primary air passage and inlet port

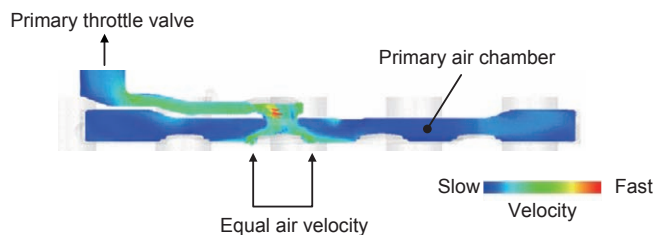


Fig. 10 Air flow analysis of primary air passage

(3) INJ base

As shown in Fig. 7(b), the air passage from the primary TH passes through the primary air chamber inside the INJ base and connects to each cylinder port.

The primary air entrance to the inlet port was placed upstream from the INJ in the location shown in Fig. 9 so as not to affect the fuel injection form.

Important considerations in determining the shape of the passage from the primary TH to the chamber are making the flow of air into each cylinder equal in order to reduce air fuel ratio difference, and mixing purge gas and air to make the mixtures homogeneous. Figure 10 shows an example of CAE performed to make the flow of air to each cylinder equal. It shows that the air flow from the merge location to either side of it was made equal by determining the appropriate location for merging the air passage into the chamber.

4. Weight Reduction in Major Moving Parts

Reducing the weight of major moving parts is an effective means of reducing inertial force so as to enhance power output and response. This approach also reduces the diameters of the crankpin and the main journal, resulting in reduced friction and enhanced fuel economy. The following will describe weight reduction measures taken with the present engine.

4.1. Pistons

Measures taken to reduce weight were, employing forged pistons, application of CAE for thermal stress to optimize the thickness of each part and application of piston behavior CAE to minimize the area of the piston skirt. Figure 11 shows the results of piston behavior CAE to configure the piston skirt with the appropriate stiffness and form. The result, as seen here, was a favorable surface pressure distribution with a maximum surface pressure area at the center sloping gently outwards to reduced pressures around it.

4.2. Connecting Rods (Conrods)

A weight reduction of 18% relative to steel was sought by the adoption of titanium conrods (Fig. 12) connected by bolts that do not employ nuts. The thrust-receiving surface on the large end of the conrod was given a diamond-

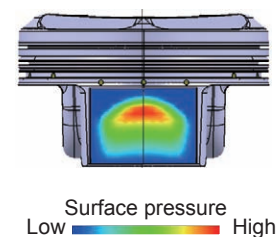


Fig. 11 Piston skirt surface pressure distribution analysis

like carbon (DLC) coating as a measure against seizing. A molybdenum disulfide surface treatment with a stable torque coefficient was also applied to the conrod bolt so that even repeated tightening would not cause scoring.

4.3. Crankshaft

As shown in Fig. 13, the stress relief groove that had been adopted in the Honda F1 engine was situated on the weakest crankshaft web. Carving out a groove reduces the stiffness of the area around crankpin fillet R, where stress figures rise high. This also mitigates the tendency for deformation to be concentrated on that fillet R and reduces the stress. This approach yielded a 7% reduction in stress, allowing the crankpin diameter to be reduced by 3 mm.



Fig. 12 Titanium connecting rod

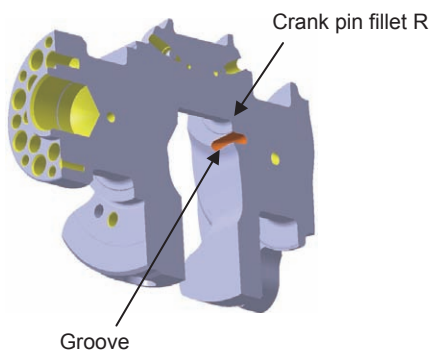


Fig. 13 Stress relief groove of crankshaft

5. Reliability Assurance Technology

5.1. Conceptual Approach for Reliability Assurance

The basic purpose of reliability assurance is to provide the full necessary assurance covering a wide range of operating conditions, from driving at the limit, as done in racing cars on the circuit, to everyday driving on city streets.

The race circuit is an operating environment that cannot be envisioned for ordinary engines, and in order to assure reliability for driving at the limit in that environment, frequency of use in terms of engine speed, engine load, engine oil temperature, and the like when driving on the circuit were determined. This was done by actual driving and by simulation on major world circuits such as Suzuka

in Japan and Nürburgring in Germany. Figure 14 shows one example of engine speed frequency when driving at the limit on the Nürburgring Circuit. It is apparent here that the engine is constantly being operated close to its upper speed limit. Figure 15 shows one example of a dynamometer durability testing mode used to assure reliability when driving under extreme circuit conditions like these. This durability test models continuous operation for 48 hours over the fastest lap on the Nürburgring Circuit.

The goal for service life was to satisfy the standards of the US ACURA for the wide range of use extending from driving on the circuit to driving for everyday transportation, and development was pursued accordingly. On the other hand, excessive quality for durability can sacrifice performance, and the durability testing standards were therefore considered carefully in light of data from actual driving on public roads, and the test was configured differently than for ordinary engines.

First of all, Fig. 16 shows the frequency of power output and engine speed when driving on the Autobahn in Germany as organized by the engine's highest performance. The vertical axis on the left shows the engine working ratio, which is the amount of engine work for driving a certain distance on the Autobahn, normalized by the amount of

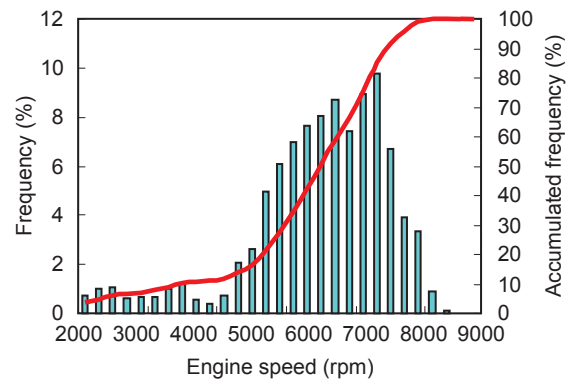


Fig. 14 Engine speed frequency of circuit driving

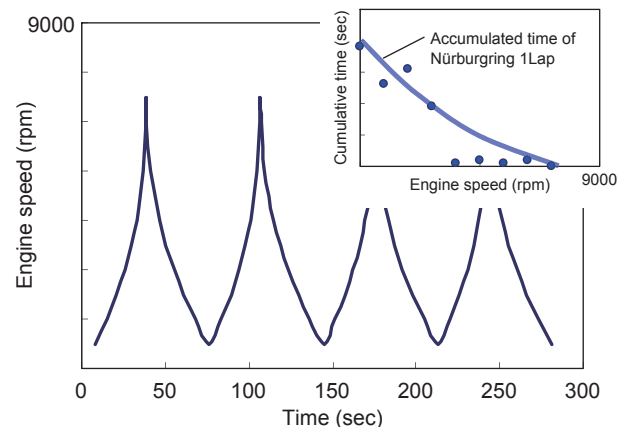


Fig. 15 Simulated circuit driving test mode of durability on dynamometer

work done when driven for a certain period of time with the engine at maximum power output. The vertical axis on the right is the frequency of use of high engine speed when driving on the Autobahn. The horizontal axis is the maximum power output of the engine in the test vehicle. It is apparent that in cars with higher maximum engine power, both the work ratio and the high engine speed frequency are lower.

Next, driving simulation was used to calculate the high engine speed frequency in ordinary driving on city streets and on highways in North America. The results show that 99% of use is at 3000 rpm or under, and the average engine speed came out to about 1800 rpm.

In addition to the above results, the circuit driving frequency discussed above was also taken into consideration. The reliability assurance test standards were therefore determined from the following three perspectives.

(1) Determination of durability conditions using engine work as a standard

It is apparent from the work ratio trend for driving on the German Autobahn, as shown in Fig. 16, that the frequency of use in the engine's high-performance region will be lower in cars that have higher maximum power output. It is therefore necessary to reflect this point in the durability conditions. As a result, the power output conditions for dynamometer durability testing, which is ordinarily conducted at maximum power output, were set lower to match the work ratio described above.

(2) Determination of high-speed durability conditions

As is apparent from Fig. 16, the frequency of use in the high engine speed region is also lower in cars that have higher maximum power output. The time of continuous dynamometer durability testing at high engine speed was therefore made shorter than usual. As discussed above, however, as circuit driving makes more use of high engine speeds, an additional, special durability test was set up for the limit engine speeds shown in Fig. 15.

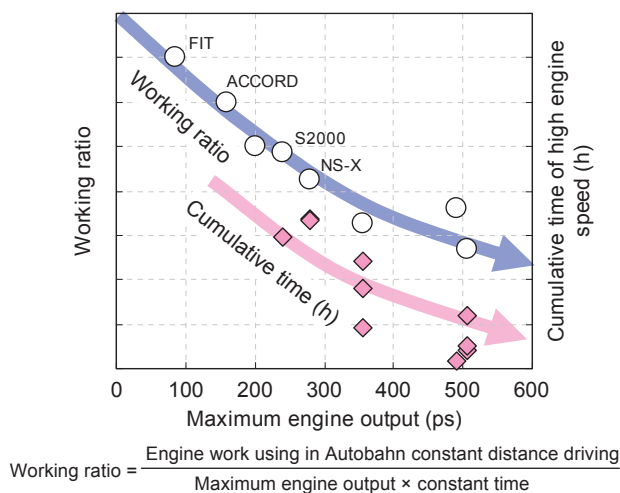


Fig. 16 Working ratio and high engine speed characteristic of high-power vehicles

(3) Determination of durability conditions to guarantee driving on ordinary public roads

Ordinary driving in North America involves a higher frequency of use at low engine speed. This characteristic is common to ordinary roads and expressways in all the countries of the world, with the exception of the Autobahn in Germany. This was addressed by using a longer than usual durability time in the evaluation mode for friction in moving parts, as represented by the valve assembly.

Figure 17 shows the concept of reliability assurance shown above as expressed in terms of engine speed and load frequency. As discussed above, the reliability assurance test standards were set up for the conditions of driving on the circuit, driving on the Autobahn in Germany, and ordinary driving in North America. These standards are considered to have enabled the full reliability assurance necessary for all operating regions in which this engine will be used.

5.2. Crank Lubrication Specifications

The topic here is determination of specifications for crankshaft lubrication, which is of particular importance in assuring reliability when driving on the circuit.

The first step was to show how this engine is situated in terms of load by comparing the PV value of the crankpin bearing in the vicinity of the red zone for this engine with those of mass-production engines, as shown in Fig. 18. This engine has the highest PV value, indicating the necessity for

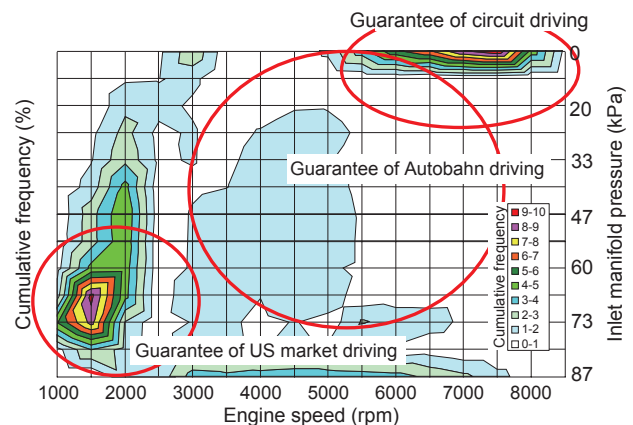


Fig. 17 Concept of reliability guarantee

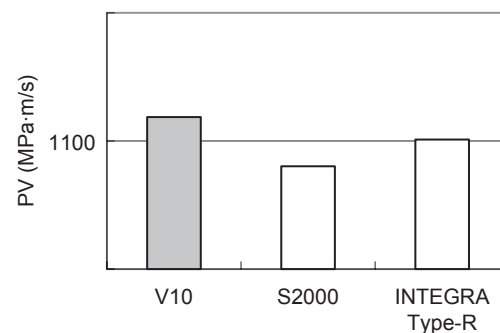


Fig. 18 PV value of crankpin bearing

new technology for reliability assurance.

Ordinary lubrication from the main journal was used for the crankpin lubrication layout in the present engine. During initial development, the engine started out with a center lubrication system of the kind commonly used in racing. As shown in Fig. 21 (to be discussed below), however, this did not have sufficient aeration toughness, and this approach was abandoned as inappropriate for mass-production vehicles, which did not provide the stringent degree of oil volume control it required. Figure 19(a) shows that a characteristic of the lubrication layout is that pressure loss due to centrifugal force can be reduced by placing the branching point between the lubrication port in the main journal of the crank and the lubrication port in the crankshaft web as far out toward the periphery as possible. Figure 19(b) further shows how steps were taken to enhance oil volume and oil pressure due to the “scraping in” effect by refining the shape of the area where oil flows in from the crank bearing through the lubrication port placed in the main journal and making the opening wider in the direction of rotation.

These measures caused the oil pressure immediately before the crank pin bearing to increase 1.2 times in the average value and 2.0 times in the minimum value, as shown in Fig. 20. The temperature in the crank pin bearing was reduced by approximately 10°C.

Figure 21 shows how it was confirmed that this crank specification was advantageous even when aeration occurred. These data represent measurements made of changes in the oil pressure in the main gallery and the oil pressure immediately before the crankpin bearing, as well as of the aeration rate, when air was blown into the main gallery before oil entered the crank. This can be explained in terms of the blue line, which represents the oil pressure in the present engine. Point A represents the oil pressure in the main gallery (horizontal axis) and the oil pressure immediately before the crankpin bearing (vertical axis)

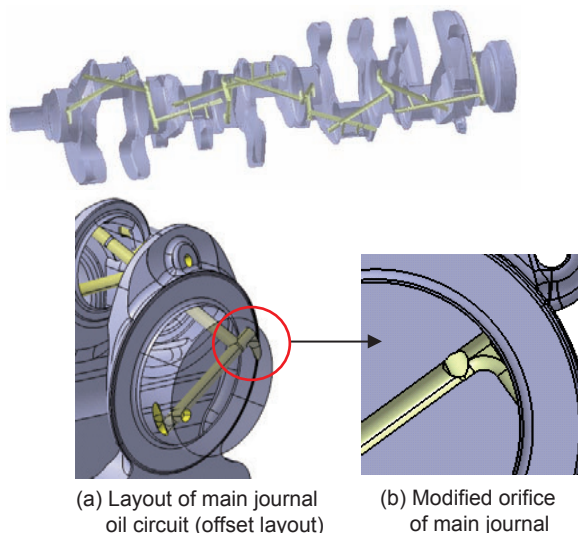


Fig. 19 Layout of crankshaft oil circuit

when aeration is zero. When aeration is increased, the oil pressure shifts the blue line to the left. With the present crank specifications, there is only a slight decline in the oil pressure immediately before the crankpin bearing when aeration increases, and it is apparent that reliability is high under limit conditions.

5.3. Measures to Counter Spark Plug Ground Electrode Damage

Finally, the measures taken to counter damage to the spark plug ground electrode will be described. This damage became an issue during the high engine speed durability testing performed for the purpose of providing a guarantee for driving at the limit on the circuit, as mentioned in the foregoing section. It had long been recognized that one of the reasons for damage to the spark plug ground electrode was that vibrations generated by the impact force of the intake and exhaust valves seating when the engine was running in the high-speed range were transmitted to the spark plug. Only rarely, however, had this been expressed in quantitative terms. For this engine, it was considered necessary to obtain a quantitative grasp in order to take

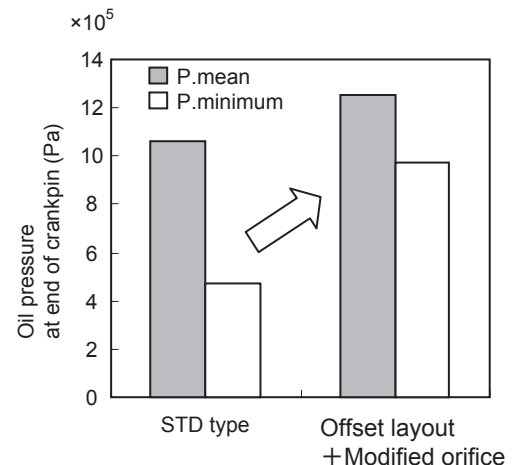


Fig. 20 Oil pressure at end of crankpin

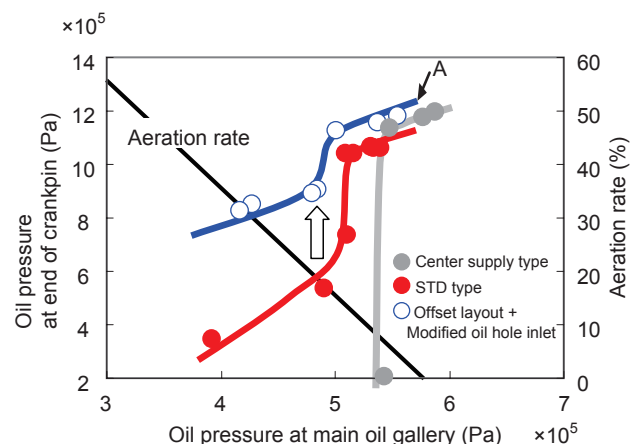


Fig. 21 Toughness of aeration

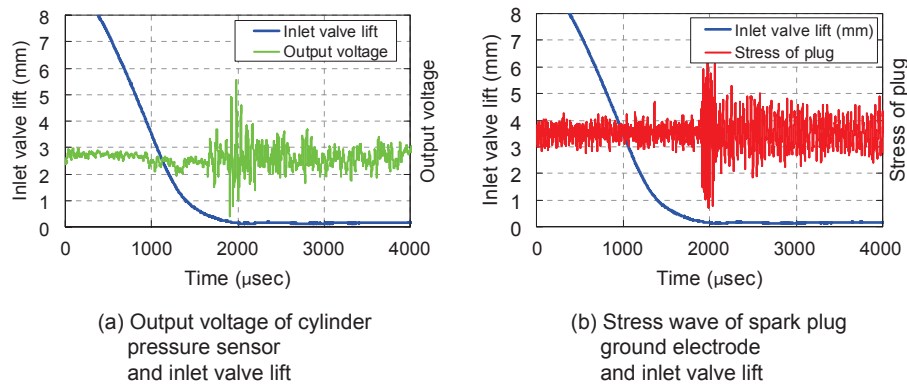


Fig. 22 Analysis result of spark plug ground electrode

proper measures against that phenomenon. Measurements were therefore made of the vibration generated in the spark plug seating surface due to the impact force arising when valves seated. Measurements were also made of stress in the spark plug ground electrode, for higher frequencies up to 200 kHz.

Figure 22(a) shows valve lift characteristics at high engine speeds and the output value from a cylinder pressure detection system⁽¹⁾, which represents spark plug seating-face vibration during motoring. Figure 22(b) shows valve lift characteristics and stress at the location of spark plug ground electrode damage. From this it could be confirmed that vibration was generated by the impact of valve seating, and synchronously with that, stress was generated in the ground electrode of the spark plug. Figure 23 shows the results from frequency (PWR) analysis of the stress waveform in Fig. 22(b). It was ascertained from Fig. 23

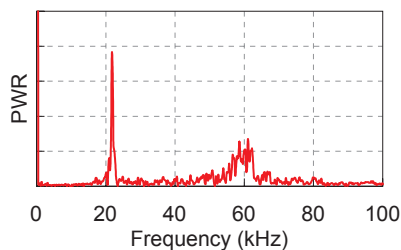


Fig. 23 Frequency analysis of spark plug ground electrode stress

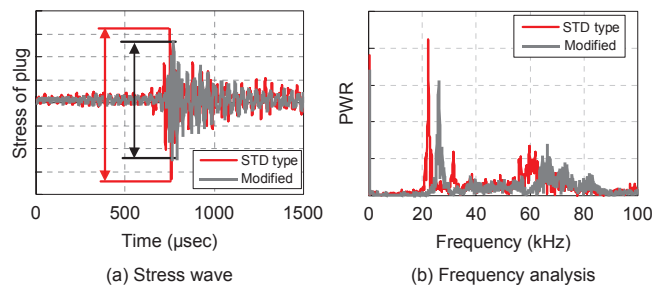


Fig. 24 Analysis result of before and after shape change

that the frequency components from 20 kHz to the vicinity of 60 kHz were the determining stress components. When CAE was used to find the resonance frequencies in the spark plug ground electrode, the first resonance frequency was found to be approximately 20 kHz and the second resonance frequency approximately 60 kHz. It was confirmed from the preceding that the excitation force from the valve seating caused sympathetic vibrations on the outside of the spark plug that tended to result in damage. Alteration of the cam profile was considered as a countermeasure, but this resulted in diminished power output. It was decided, therefore, to change the shape of the spark plug ground electrode to raise the resonance frequency. This change was confirmed to have raised the first resonance frequency by approximately 3 kHz and the second resonance frequency by approximately 12 kHz.

Figure 24(a) shows the results from stress measurement before and after the shape change. Figure 24(b) shows the results from frequency analysis of the waveform in (a). It could be confirmed that the stress frequencies had shifted toward higher frequencies after the shape change, and that the amplitude of the stress had diminished as well. Durability testing was conducted with items incorporating the countermeasure, and no damage occurred.

6. Conclusions

The following achievements were realized regarding power output and reliability.

- (1) High performance: Power of 405 kW (550 PS) achieved
- (2) High response: Developed an individual throttle that complies with California State LEV2-ULEV
- (3) Weight reduction: Change of material and optimization of shape by means of CAE resulted in comprehensive weight reductions of major moving parts
- (4) High reliability: Durability tests were conducted based on actual operating data from both driving on the circuit and ordinary driving, and fully adequate reliability was established.

Given the above results, it can be concluded that an

engine was developed that achieves a balance of high performance and high reliability, such that it can provide the pleasure of driving on the circuit as in a race car with performance in an entirely different dimension from that of an ordinary car, and on the other hand can be used like an ordinary automobile for everyday transportation on expressways and city streets.

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