

Development of i-DSI 2Plug Engine for 2004 Model Year Honda LIFE

Yusuke WADA*

Kennosuke NARITA*

Keiji OHTSU*

Toshimitsu SHINOHARA*

ABSTRACT

A 660 cm³ 3-cylinder engine has been developed for the new model of the Honda Life. The basic structure of the engine has been redesigned to achieve increased environmental performance, increased compactness, a reduced weight, and a higher level of quietness.

The development concept was to produce an engine that excelled under normal driving conditions. To respond to the high frequency of high engine loads and engine speeds in minicars, the i-DSI engine employs a short stroke to reduce friction. The use of i-DSI twin plug ensures rapid combustion, enabling fuel economy and output to be increased through the use of a high compression ratio.

The innovations outlined above have increased torque by 7% under normal driving conditions, and enabled the achievement of a class-topping fuel economy of 19.8 km/L in Japanese 10 - 15 mode. The increased idling fuel economy achieved by improved combustion and reduced emissions, and the reduced noise and vibration make the new engine an extremely high-quality unit.

1. Introduction

With the goals of environmental protection and resource conservation, Honda has developed the “i-series” of power trains - lightweight, compact power trains giving lower emissions, enhanced fuel economy, and better drive performance. With the lean-burn i-VTEC used in the Stream, the i-DSI⁽¹⁾ providing rapid combustion through dual and sequential ignition used in the Fit, and the i-VTEC cylinder idling system used in the INSPIRE, these new engines were launched into the market providing advanced fuel control.

In the development of this engine, studies were carried out concerning how minicars are used and what would be the best type of engine for them. Development proceeded with the goals of a simple structure and high performance. This paper reports on the engine concept, as well as on the technologies introduced and the performance achieved.

2. Development Goals

The 2004 Model Honda Life, for which this new engine was designed, was developed according to the concept “Pride and Dream.” Corresponding to this, studies were made into engine specifications to achieve a small vehicle with low energy consumption that is friendlier towards the environment and mankind. In particular, the following four items were set as environmental performance targets for the small car engine.

- (1) Environmental performance; improved fuel economy and conformity to ultra-low emissions standards
- (2) Dramatically improved torque in the practical range
- (3) A lightweight and compact engine
- (4) Reductions in engine vibration and noise

* Tochigi R&D Center

3. Overview of the Engine and Specifications

3.1. Overview

To study the engine operating range, minicar driving load information was analyzed. Figure 1 shows the results of a comparison of driving frequencies when driving in the Japanese 10 - 15 mode, by engine displacement. For 660 cm³, there is a high usage frequency of relatively high load at high engine speeds.

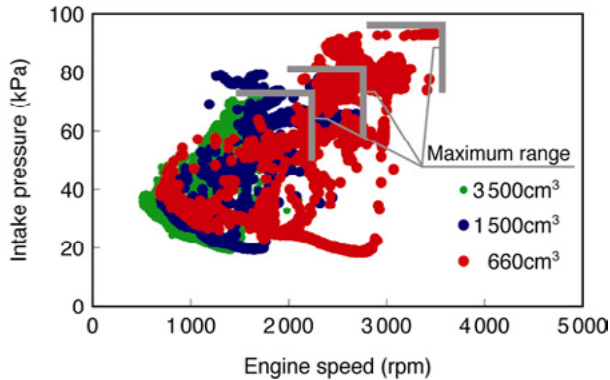


Fig. 1 Comparison of range of operating loads for engines of different displacement in 10 - 15 mode

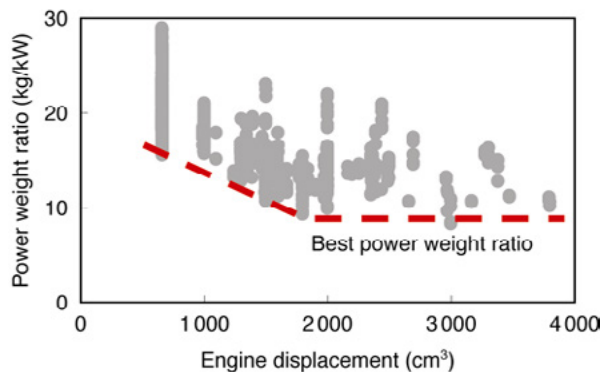


Fig. 2 Best power weight ratio for engines of different displacements

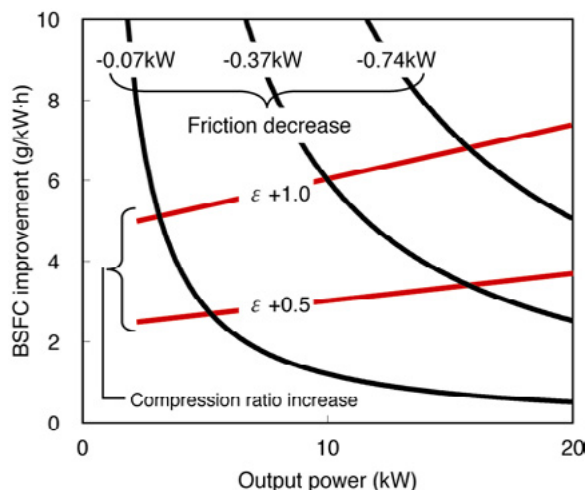


Fig. 3 Improvement of BSFC by friction and compression ratio

Table 1 Engine specifications

Engine	Developed model	Previous model
Cylinder configuration	In-line 3-cylinder	In-line 3-cylinder
Bore × stroke (mm)	71 × 55.4	66 × 64
Displacement (cm ³)	658	656
Compression ratio	11.2	10.5
Valve train	SOHC 2 valves	SOHC 4 valves
Number of valves	2 per cylinder	4 per cylinder
Valve diam. Inlet (mm)	35	24.5
Exhaust (mm)	30	21
Fuel supply system	Programmed fuel injection	Programmed fuel injection
Ignition system	Dual point sequential ignition	Single point ignition
Max. power (kW/rpm)	38/6700	38/7200
Max. torque (N-m/rpm)	61/3800	61/4500

Figure 2 shows the relationship between naturally aspirated engine power to weight ratio and engine displacement. The minimum power to weight ratio for minicars is equivalent to 1.5 times that of a vehicle of 15 kg/kW and 1 800 cm³ or larger. This is the reason for the increase in use frequency of high load, high engine speed in minicar.

Focusing on environmental performance, particularly on the improvement of fuel economy, it is best to have both rapid and stable combustion, regardless of the engine displacement. On the other hand, focusing on minicars, high engine loads are frequently used and therefore the effectiveness of pumping loss reduction methods, such as large volume EGR, is limited. Therefore, the effect of fuel economy improvement methods was checked against desired output.

Figure 3 gives the tests results concerning Brake Specific Fuel Consumption (BSFC) improvements, using a previous 660 cm³ Engine. BSFC improvements through friction reduction are limited when accompanied by an increase in the required output. In contrast, it is clear that using a higher compression ratio to reduce BSFC is an effective method, showing improvements accompanying an increase in required output.

Consequently, it is clear that for minicar engines, the following methods are effective in the significant reduction of high engine speed friction, while avoiding knocking and at the same time setting a high compression ratio.

- (1) High engine speed friction reduction, through the use of a short stroke.
- (2) Pumping loss reduction by larger valves using a bigger bore.
- (3) High compression ratio, through a 2-valve combustion chamber, using 2 plugs for both dual and sequential ignition.

From above, the short stroke frame and i-DSI adoption was determined for the engine basic concept. Table 1 shows the main specifications, and Fig. 4 shows the appearance of the engine.

The features of the engine combustion chamber and the area around it are given in Fig. 5. First, note the big bore, short stroke frame. By applying a bore stroke ratio of 0.78 and a low average piston speed of 12.4 m/sec at the maximum output, friction at high engine speeds was reduced. The adoption of a 2-valve configuration with large diameter

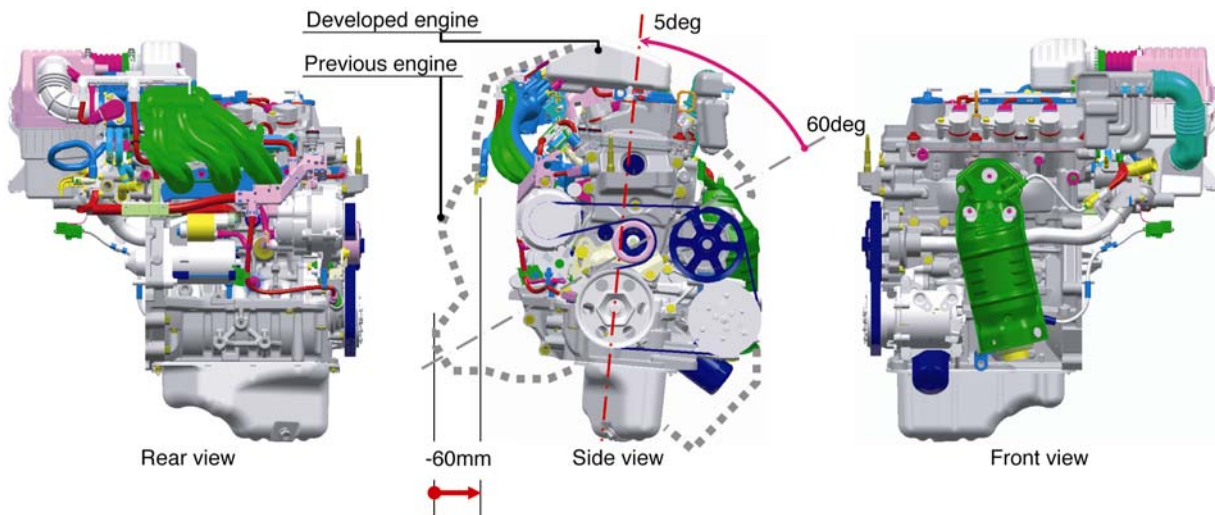


Fig. 4 View of new engine

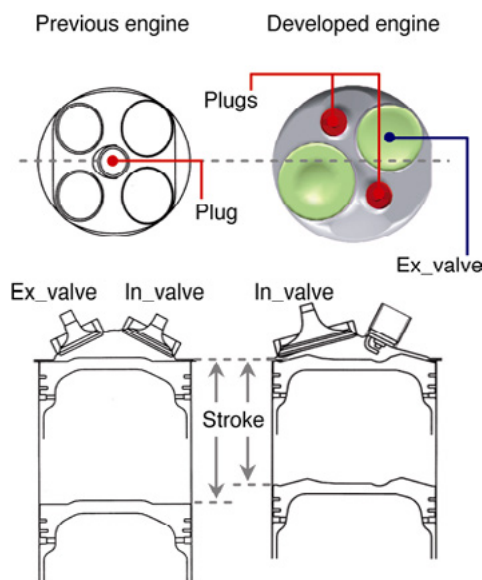


Fig. 5 Comparison of combustion chambers configuration of previous and new engines

valves enabled any increase in pumping losses to be kept to a minimum.

In addition, the i-DSI, already used in the Fit, which has two spark plugs placed diagonally in each cylinder, was adopted for the important purpose of creating a high compression ratio through dual and sequential rapid combustion characteristics.

As a result, the compression ratio was set at a maximum level of 11.2 for the regular gasoline (RON91) specification. In addition, mid to high engine speed torque (which has a high usage frequency) was raised, while keeping the maximum torque and power equivalent to the output of the previous model.

3.2. Reduction in Weight and Size

It is very important that minicars are given small engines, in order to maintain a reasonably sized cabin space within the vehicle. Innovations that result in a compact layout include

direct attachments of auxiliary equipment to the cylinder block and an exhaust manifold-integrated cylinder head connecting directly with catalytic converter. Changing the engine-fitting angle from a forward 60° tilt to a forward 5° tilt enabled the total fitting length to be shortened by 60 mm.

In addition, placing the concentrated resonator type air cleaner on top of the transmission contributes to making the total power train more compact.

Moreover, by optimizing the shapes and layout of the areas around the crank shaft and cylinder block, and by making the intake manifold, cylinder head cover, oil strainer, and oil level gauge out of plastic, an engine weight reduction of 9% was achieved.

4. Achieved Performance

4.1. Improved Fuel Economy

Figure 6 shows engine unit fuel consumption and combustion stability under a light load. A 7% reduction of BSFC in stoichiometric air fuel ratio was achieved, and combustion limit air fuel ratio (A/F) was made a leaner 3 A/F. In addition, combustion duration of stoichiometric air fuel ratio was shortened by 13°. This is the effect of rapid combustion. The following are details of the element technologies.

4.1.1. i-DSI application effects

When applying the i-DSI to the short stroke frame, studies were made of fuel consumption effects from the i-DSI cylinder head spec in comparison with the 4-valve cylinder head spec represented in Fig. 5.

The test engine used was a previous engine modified as much as possible to give it a short stroke. Specifically, a bore diameter of 68 mm, stroke of 60.4 mm, compression ratio of 11.5, and valve pinch angle of 30° were set. The inlet and exhaust valves were with diameters of 25 mm and 21 mm for 4-valve spec, and 33 mm and 28 mm for the i-DSI spec.

To evaluate the fuel economy effects accompanying this change, 3 modes were established, including the minicar actual driving range.

- | | |
|--------------------------------|-------------------------|
| (1) Combustion evaluation mode | 1 500 rpm, BMEP=358 kPa |
| (2) Light load mode | 2 650 rpm, BMEP=171 kPa |
| (3) High load mode | 4 000 rpm, BMEP=838 kPa |

As shown in Fig. 7, the i-DSI spec achieves a 2 - 6% reduction in Indicated Specific Fuel Consumption (ISFC), and a 20 - 50% reduction in HC emissions. These reductions are due to the benefits of increased port flow speed with a single inlet valve, giving a stronger flow within the cylinder, and rapid combustion created by having dual and sequential ignition. The details of this rapid combustion are shown in Fig. 8. In the combustion evaluation mode, it is possible to fully understand rapid combustion, whereby the total combustion period is shortened by 8%, delaying the Minimum Spark Advance for Best Torque (MBT) ignition timing and advancing the maximum cylinder internal pressure generating angle. As a result, fuel economy was improved by raising the average effective pressure through raising the maximum cylinder pressure.

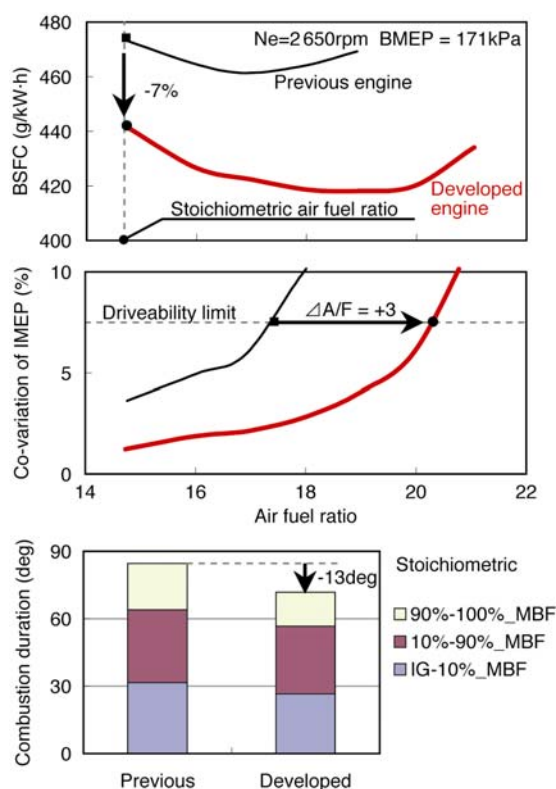


Fig. 6 BSFC and combustion characteristics

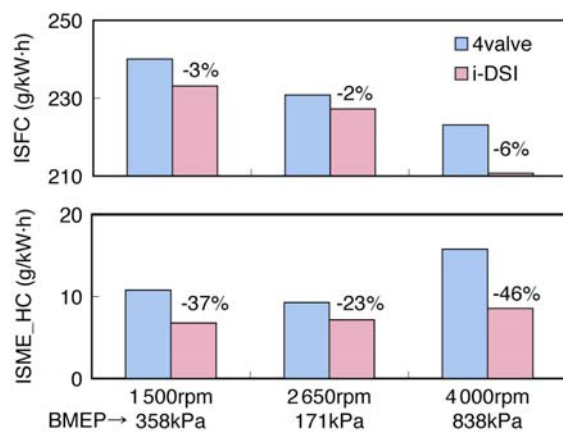


Fig. 7 Effect of i-DSI on ISFC and ISME_HC

The results of i-DSI rapid combustion benefits are clear. The mass production spec, which aims at a further reduction in friction, has a bore pitch of 80 mm, a cylinder bore diameter of 71 mm which makes the distance between bores 9 mm. The stroke length corresponding to this is 55.4 mm, fulfilling the short stroke specification.

Next, studies were carried out to optimize swirl and tumble ratios through inlet port modifications aiming at improved fuel economy.

- (1) a_Port: Tumble port (base spec)
- (2) b_Port: Strengthen swirl ratio to 4 times that of a_Port.
- (3) c_Port: Further enhance tumble ratio 25%, swirl ratio 12%.

Figure 9 shows the results of a distribution simulation of intake airflow velocities around the inlet valve seat, during

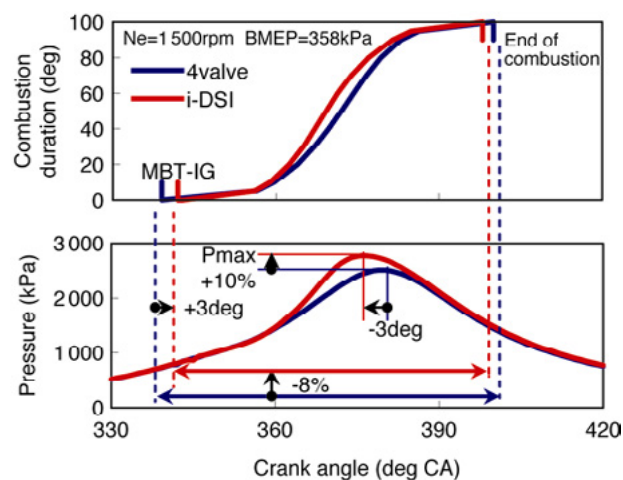


Fig. 8 Effect of i-DSI on combustion characteristics

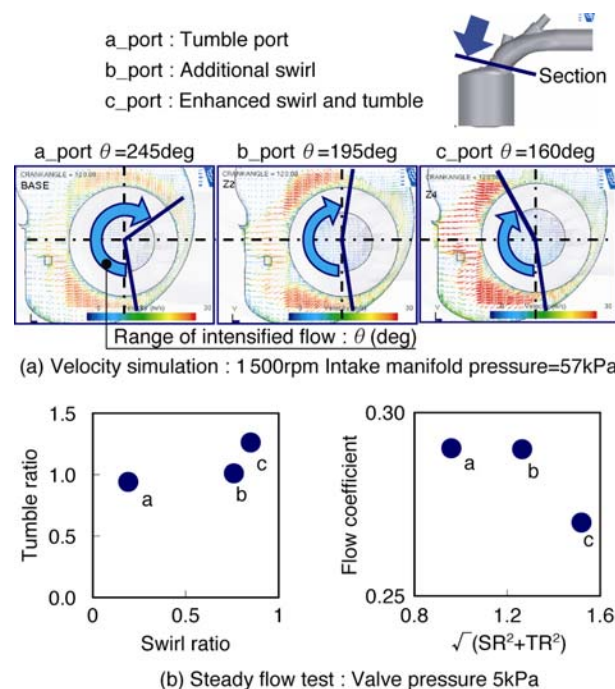


Fig. 9 Inlet port flow characteristics

the intake stroke. It is clear that in the proposed port order of a, b and c, the flow directivity is strengthened. Comparing flow characteristics in terms of average flow coefficient and vortex intensity⁽²⁾, there was a drop in the average flow coefficient against a relative rise in c_Port vortex intensity, so a drop in output was expected. The vortex intensity used here is defined as composition value $\sqrt{(SR^2+TR^2)}$, with SR the swirl ratio TR the tumble ratio.

Figure 10 shows the result of actual ISFC measurements. The ISFC reduction allocation against a_Port is 0.5% at b_Port and 1.0% at c_Port, so by raising vortex intensity, ISFC was improved. In the end, port specifications showed a 2.5% drop in maximum power output with the c_Port, so the b_Port was selected for mass production.

4.1.2. Friction reduction

Aiming for a reduction in friction in the mid and high engine speed range, which sees high use frequency with minicars, a short stroke frame was adopted, and the crank journal diameter was reduced.

It is known that making a short stroke results in a reduction in piston speed, and there is a reduction in oil membrane shear resistance between the piston sliding surface

and cylinder sleeve wall surface, and a reduction in pumping loss. In addition by using a short stroke, it is possible to expand the overlap formed by journal diameter and pin diameter, thereby increasing crankshaft rigidity, which is beneficial in terms of crank rattling noise. As a result of studies of friction reduction in the crank journal part, higher crankshaft rigidity, and metal seizure, the diameter of the crank journal was reduced from 40 mm to 38 mm in the new engine.

Other adopted friction reduction techniques included the continued use of the roller follower used on the current model engine, and applying OW-20 low viscosity engine oil. Also contributing to reduced friction is the newly adopted plateau honing used on the Civic Hybrid's engine⁽³⁾, an offset cylinder structure, ion plated piston ring, main metal and connecting rod metal streaking, and molybdenum disulfide shot peening on the piston skirt⁽⁴⁾.

The friction reduction effects are shown in Fig. 11. These results are from a comparison of motoring friction when the throttle is fully open, including pumping losses. Throughout almost all of the engine speed range, there was a reduction in friction of 10% or more. Figure 12 shows the breakdown of friction reductions. Friction at 5 000 rpm showed a 13% reduction in comparison with the previous engine. Of this reduction 55% comes from around the piston, 11% from around the crankshaft, and a 34% from the valve system drive.

4.1.3. Idling stability improvements

Idling engine speed reduction is very important in reducing fuel consumption. The idling engine speed with the newly developed engine is 50 rpm less than the previous engine, making it 750 rpm. This is a benefit of the rapid and stable combustion of the i-DSI.

Figure 13 shows the engine speed fluctuation and its frequency for each target idling engine speed. The measurements were taken in a bench test with the engine

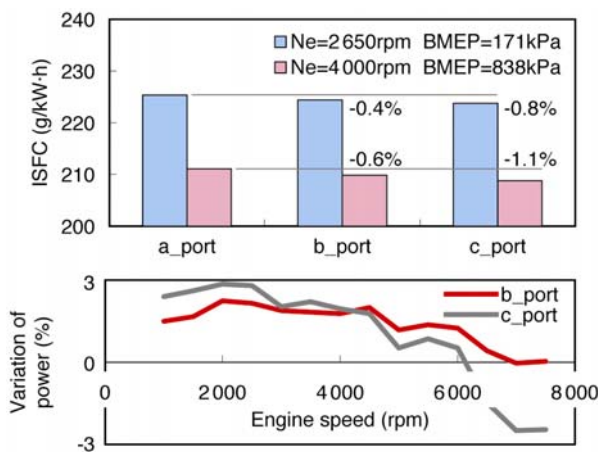


Fig. 10 Comparison of ISFC and power by inlet port

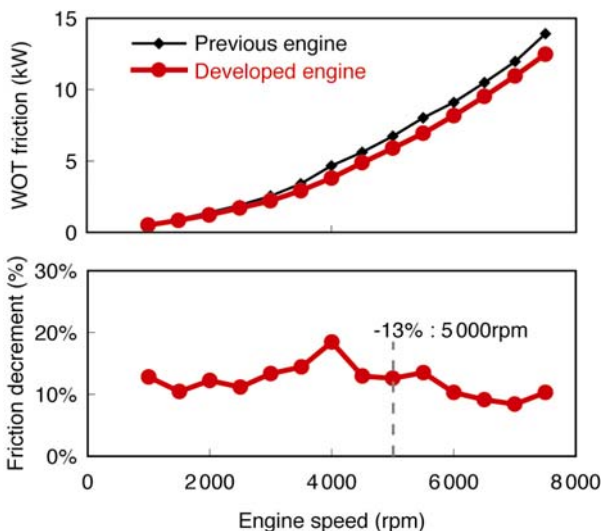


Fig. 11 Decrement of WOT friction

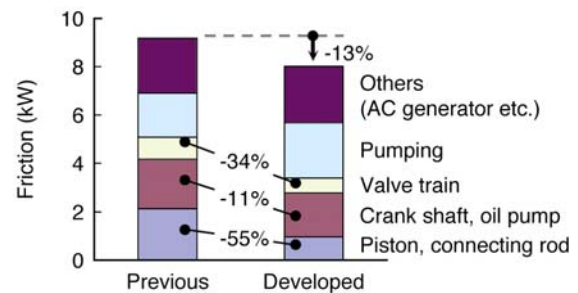


Fig. 12 Comparison of friction distribution

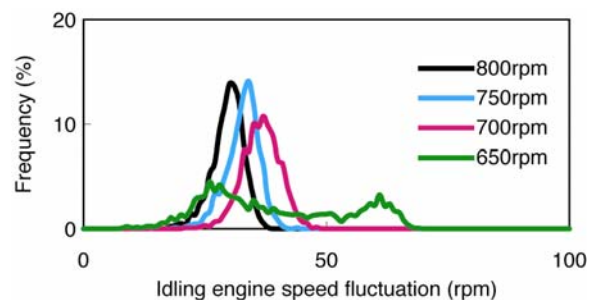


Fig. 13 Stability of idling engine speed

connected to an automatic transmission. The automatic transmission selector was set in the D range, and after fully warming the engine, the air fuel mixture was kept stoichiometric. In this measurement, a tendency was found that engine speed fluctuation range increases by lowering the target idling engine speed. At idling engine speed of 650 rpm, idle stability worsens due to a widening of engine speed fluctuation range and frequency dissipation.

In contrast, Fig. 14 shows the standard deviation of maximum cylinder pressure (Pmax) against target idling engine speed. At 650 rpm, Pmax standard deviation rises, and it enters the unstable combustion range. These results closely follow the tendencies in engine speed frequency characteristics previously noted, and combustion stability greatly affects idling engine speed stability.

The mass production spec sets the engine speed at 750 rpm, including idling controllability during transient periods such as electrical load changes, etc.

4.2. Power

Figure 15 shows the engine performance curve. As for power characteristics, resonant engine speed adjustments are carried out using intake system specifications. While the maximum torque value is equivalent to the previous engine, there is an average torque improvement of 7% in the engine speed range up to 4 000 rpm. In addition, the same maximum power value occurs at an engine speed that is 500 rpm lower,

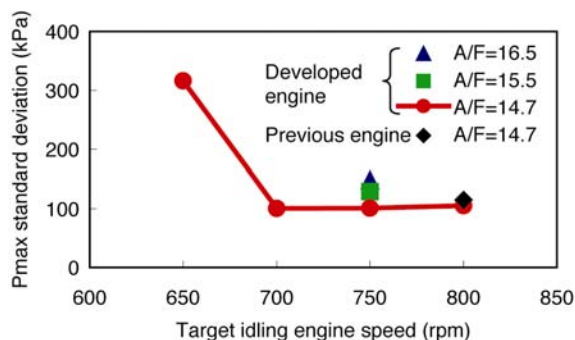


Fig. 14 Stability of idling Pmax

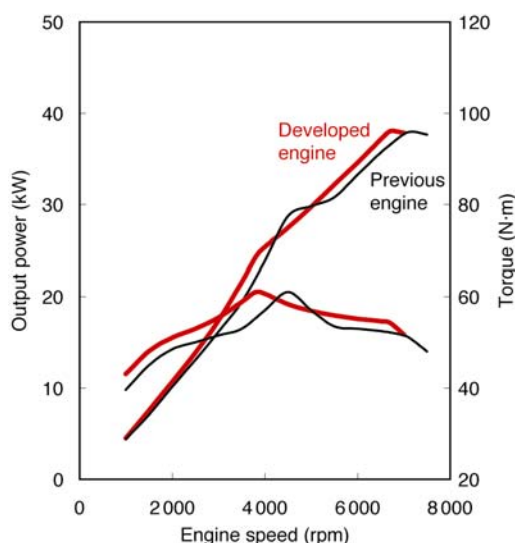


Fig. 15 Engine performance

giving practical power characteristics.

Furthermore, various types of knocking suppression technologies have also contributed to modifications to the new engine's power characteristics. The application of i-DSI rapid combustion, swirl and tumble ports, and a new coolant path are given as specific examples.

In the same way as with fuel economy improvement, the effect on power of the i-DSI cylinder head spec using short stroke spec modified from the previous engine was confirmed. Figure 16 compares the volumetric efficiency characteristics against engine speed. Thanks to the 2-valve spec, the i-DSI advanced volumetric efficiency in the engine speed range up to roughly 5 000 rpm. Figure 17 shows the MBT torque characteristics at 3 500 rpm, and the set ignition timing. According to this, with i-DSI spec, the high MBT torque occurs with the ignition timing on the delay angle side, and at the same time, because the knocking margin is the same, set torque is raised 3%.

As a result of volumetric efficiency improvements and knocking suppression effects, the power characteristics of i-DSI from low to high engine speed are equal to or better than those of the previous 4-valve engine.

Concerning the coolant path, modifications were made to suppress knocking. Figure 18 shows the changes to the coolant path. Rather than flowing first to the cylinder block as the conventional coolant path would, this system has the recirculated coolant from the radiator flow first to the cylinder head. Thanks to this, recirculated coolant at a comparatively low temperature is provided around the combustion chamber at a high flow speed. This means that

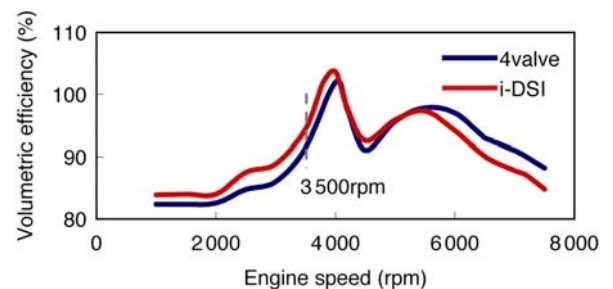


Fig. 16 i-DSI volumetric efficiency

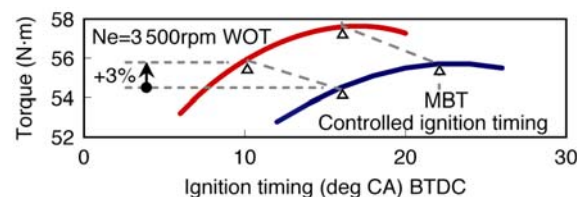


Fig. 17 Effect of i-DSI on torque

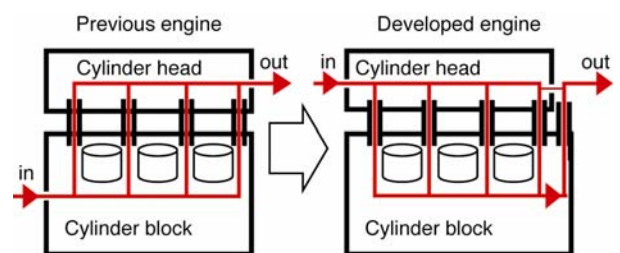


Fig. 18 Comparison of coolant circuit

there is a knocking suppression of 1-2° across the entire engine speed range, despite a higher compression ratio, which contributes to fuel economy and power improvements.

In addition, with the adoption of a cylinder head with integrated exhaust manifold, coolant flow speed distribution in the entire cylinder head water jacket was optimized so that it was possible to maintain sufficient cooling performance even with the heat flux from catalyzer flange to cylinder head. Specifically, through the coolant flow speed simulations in Fig. 19, visualization, and multi-point temperature measurements of actual engine firing, the shape was determined to minimize coolant stagnation and localized temperature rise.

4.3. Low Emission Technology

The major technologies for lowering emissions, as applied to the newly developed engine, are outlined in the four items below.

- (1) Highly accurate air fuel ratio control
 - 1-1 Highly accurate air fuel ratio control through the use of an A/F sensor before the catalyzer and an O₂ sensor downstream from the catalyzer
 - 1-2 Reduction in fuel transportation delay by placing multi-hole injectors in the cylinder head ports
- (2) A low heat mass structure around the exhaust system
 - 2-1 Cylinder head with integrated exhaust manifold
 - 2-2 Catalyzer located immediately under the engine
- (3) Stoichiometric air fuel ratio control immediately after cold start, through combustion stabilization
- (4) A large ignition timing delay angle control for early activation of the catalyst.

Here the results of application of early catalyst activation technology will be introduced. The goal of this technology is to reduce HC emissions immediately after cold start, as stated in (4) above⁽⁵⁾.

To have early activation of the catalyst, it is most effective to have an exhaust temperature rise through a large delay angle in ignition timing. However, in applying a large delay angle, it is assumed that stable combustion will be maintained.

Figures 20 and 21 show the results of engine unit tests of combustion stability, using the new engine at low water temperatures. With a water temperature of 35 °C, the ignition timing delay angle and intake air amount was adjusted so that the Brake Mean Effective Pressure (BMEP) is uniform. As a result, even with a delay angle of 25° from MBT ignition timing, the combustion fluctuation rate does not reach 5%

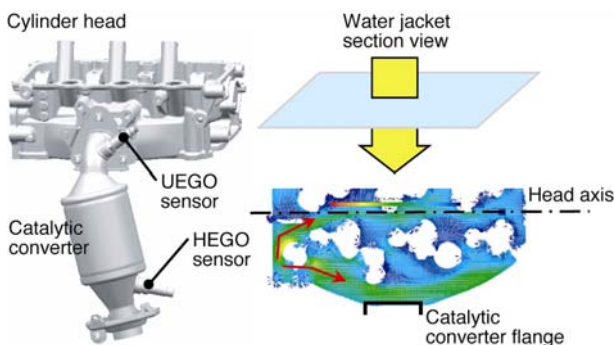


Fig. 19 Result of simulation of coolant flow to siamese exhaust port

and stable combustion is maintained, and an exhaust temperature rise of roughly 200 °C could be reached.

The results of applying this control to actual vehicles are shown below. Figure 22 shows the catalyst temperature and 11 mode HC emissions 54 seconds after engine cold start, against the ignition delay angle amount. In comparison with having no ignition timing delay angle control, even with a maximum 15° delay angle, the HC emissions reduction tendency is maintained. Eventually, the mass production spec

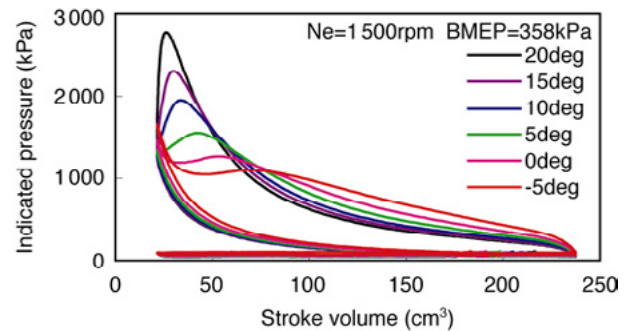


Fig. 20 Relationship between indicated pressure and ignition timing retard at constant BMEP

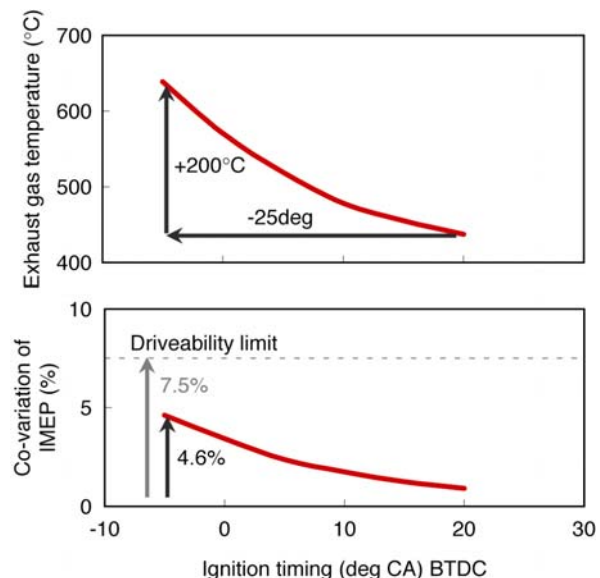


Fig. 21 Influence of ignition timing retard on exhaust temperature and combustion stability

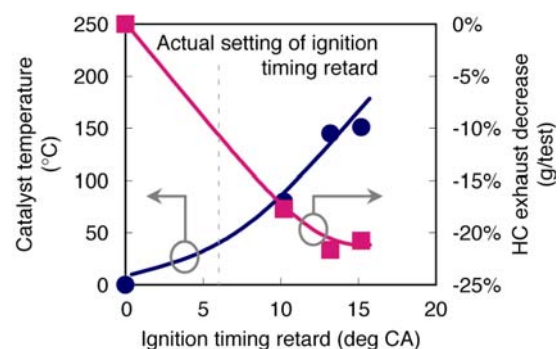


Fig. 22 Catalyst temperature rise and HC decrease

was set with an ignition delay angle of 6° , to balance fuel economy with catalyst early activation. In addition, as shown in Fig. 23, combined with the low heat mass structure around the exhaust system, a 220°C catalyst temperature rise was achieved, making the system conform to ultra-low emissions standards.

4.4. Engine Vibration and Noise Reduction Technologies

Concerning engine vibration and noise, in order to actualize a competitive engine structure for the future, progress was first made in raising rigidity and reducing

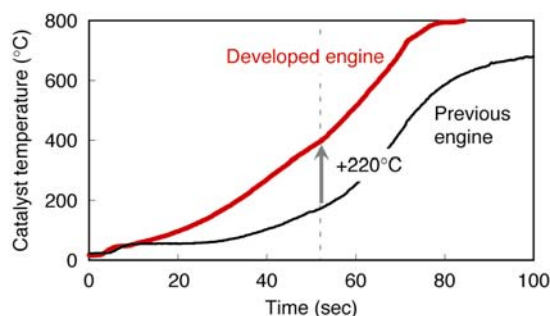


Fig. 23 Effect of siamese exhaust port and ignition timing retard on catalytic converter warm-up characteristics

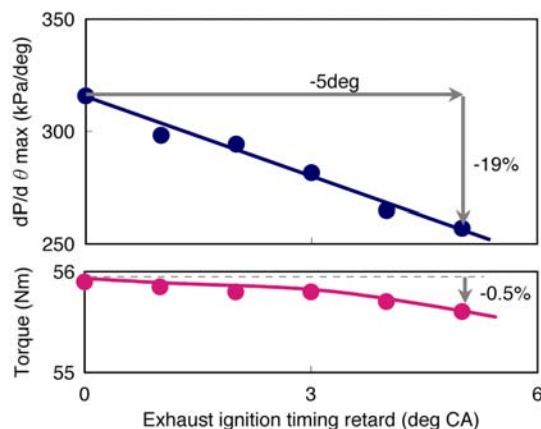


Fig. 24 $(dP/d\theta)$ max decrease and torque

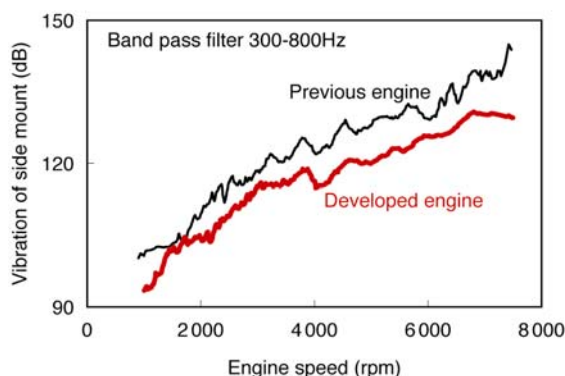


Fig. 25 Effect of i-DSI sequential ignition timing on improvement of engine noise

vibration around the major drive systems. To begin, high crankshaft rigidity was ensured through a short stroke structure, and vibration was suppressed through the use of full balance weights. By adding a twist process to the crankshaft forging and machining, a lighter weight was achieved while still having full balance. In 1-throw rigidity evaluations using computer aided engineering (CAE), crankshaft bend rigidity shows a 26% improvement over the previous model.

On the other hand, the lower block design in the cylinder block structure greatly contributes to engine frame rigidity. In this lower block, each journal is integrated into the lower block's outer wall, and through optimization of the curved surface shape and of the rib locations, the journal's bending stiffness is improved by 38%. In addition, CAE studies made it possible to maintain engine and transmission connection rigidity, without any need for auxiliary parts.

Secondly, there is i-DSI sequential ignition control for engine vibration and noise reduction. Figure 24 shows the simultaneous ignition of two spark plugs in each cylinder at 5 500 rpm WOT, and the $(dP/d\theta)$ max reduction effect through the delay angle control on exhaust side only. The mass production spec has, in contrast with simultaneous ignition, a 5° delay angle in the exhaust side ignition timing, thereby $(dP/d\theta)$ max is reduced by 19% and engine external noise is reduced by 17%, while the engine torque remains roughly the same.

By applying the above high rigidity engine structure and i-DSI technologies the mid frequency mount point vibrations, shown in Fig. 25, are reduced by 5 - 10 dB.

5. Conclusions

In the development of engines for minicars, the knowledge listed below was gained. By applying this knowledge, it was possible to achieve a commercial product that provides best-in-class environmental performance, excellent power, fuel economy and quietness.

- (1) Reducing piston speed by applying a shorter stroke enables a 10% reduction in friction. In addition, rapid combustion using dual ignition improves performance against knocking. Combined with a high compression ratio of 11.2, it is possible to improve BSFC by 7%, achieving a 10 - 15 mode fuel consumption of 19.8 km/L.
- (2) As for Wide Open Throttle power, in addition to the aforementioned rapid combustion and high compression ratio, volumetric efficiency is improved through a two-valve system, and knocking suppression performance is improved through changes to the coolant paths. These improvements mean that while reducing maximum torque engine speed by 500 rpm, there is a 7% torque improvement.
- (3) In terms of environmental performance, improvements include a catalyzer directly beneath the engine, a low heat mass exhaust system, and early catalyst activation through an ignition timing delay angle. Furthermore, the air-fuel ratio was stabilized using a highly accurate air fuel ratio control. These measures have enabled the vehicle to meet the ultra-low emission standards.
- (4) Changing the angle at which the engine is fit from 60° to 5° means that the front to rear length is reduced by 60 mm. Making the cylinder head cover and intake system parts out of plastic has reduced the engine's weight by 9%.
- (5) Examining noise and vibration performance, action was

taken to create a high-rigidity crankshaft and to increase the connection rigidity with the transmission based on CAE analysis. Moreover, by reducing $(dP/d\theta)$ max through the use of an ignition timing delay angle control the mount vibration could be reduced by 5 - 10 dB.

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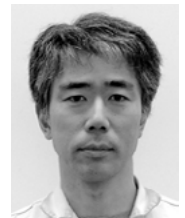
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