

# Development of a 1.3L 2-Plug Engine for the 2002 Model 'Fit'

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## ABSTRACT

A lightweight and compact, 1.3L 4-cylinder engine was developed as the new-generation i-Series power train. This engine has twin plugs for each cylinder, and achieves rapid combustion by means of high-precision dual-point sequential ignition control that responds to the engine's operating conditions. In addition, reducing friction and using the newly developed CVT have yielded the low fuel consumption of 23 km/L in the Japanese 10-15 mode. The engine has achieved 'J-LEV' certification by reducing exhaust emissions to 50% of the fiscal year 2000 emissions standards.

## 1. Introduction

In response to demands for environmental protection and lower resource consumption, Honda has developed the 'i-series' – a lightweight, compact power train series – which improves environmental (exhaust gas), fuel economy, and drive performance. This 'i-VTEC' combines VTEC with continuously variable control camshaft phase <sup>(1)</sup> and is fitted to the 'Stream'. The 'i-series' is the general name for the latest generation of engines that carry out optimum fuel control in response to the engine speed and load conditions.

This report introduces the new 'i-DSI', an 'i-series' engine for use on sub-compact cars (DSI: Dual and Sequential Ignition). This new engine is a 1.3L, SOHC, two-plug, four-cylinder engine that is lightweight and compact, which has achieved both low fuel consumption and low emissions.

## 2. Aims of Development

The development concepts for the new small car 'Fit' to which this engine is fitted are 'spacious cabin' and 'best fuel economy performance in the world'. In order to achieve this concept, the aim was to develop a compact design that responds to the vehicle packaging and lower emissions, by focusing on engine fuel consumption improvements through friction reduction, adopting a rearward layout of the exhaust system and an exhaust gas oblique intake flow catalytic converter.

The detailed objectives for the development of the engine were as follows:

- 1) Highest fuel economy performance in the world (Japanese 10-15 mode fuel economy: 23 km/L)
- 2) High torque performance
- 3) 50% reduction over FY2000 emissions requirements ('J-LEV' certification)
- 4) Compact engine design that facilitates a roomy cabin

## 3. Outline and Main Specifications of the Engine

Fig. 1 shows the appearance of the engine, and Table 1 compares the specifications of the new engine (L13A) to those of the previous engine (D13B).

As shown in Fig. 2, the lightweight, compact design is achieved through the adoption of bracket-less accessories and a serpentine accessory belt, a long, plastic intake manifold and cam chain drive. In comparison with the existing 1.3L D13B engine, the new design is compact, being 118 mm shorter and 69 mm narrower, and lightweight, with a total engine weight reduction of 8%. Also, as shown in Fig. 1, the air cleaner and intake resonator chamber are placed above the transmission, to also contribute to a more compact power train overall.

The special features of this engine are a higher compression ratio (9.2 → 10.8) achieved through the placement of two spark plugs at opposite angles to each other in each of the cylinders (Fig. 3), and rapid combustion attained through dual-point sequential ignition

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timing control. This provides both improved fuel consumption properties and output.

Fig. 4 shows cross sections of a conventional cylinder head and the type adopted in this engine. In the new head, there is a single rocker arm shaft supporting the inlet rocker arm and exhaust rocker arm, and the valve angle was narrowed from  $46^\circ$  to  $30^\circ$ . The adoption of these technologies results in a more compact combustion

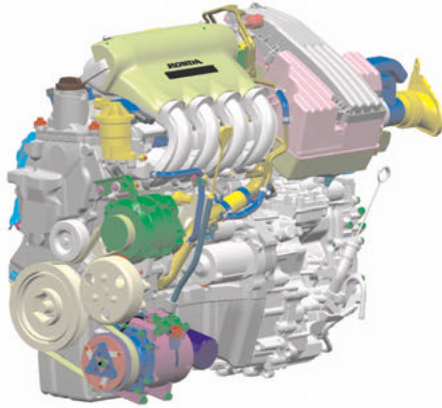


Fig. 1 View of 2002 model i-DSI engine

Table 1 Engine specification

| Engine code                     | 2002 model i-DSI<br>L13A          | 1999 model Logo<br>D13B      |
|---------------------------------|-----------------------------------|------------------------------|
| Cylinder configuration          | In-line 4-cylinder                | In-line 4-cylinder           |
| Bore $\times$ stroke (mm)       | 73 $\times$ 80                    | 75 $\times$ 76               |
| Displacement (cm <sup>3</sup> ) | 1339                              | 1343                         |
| Compression ratio               | 10.8 : 1                          | 9.2 : 1                      |
| Valve train                     | SOHC 2 Valves                     | SOHC 2 Valves                |
| Number of valves                | 2 per cylinder                    | 2 per cylinder               |
| Valve diameter (mm)             |                                   |                              |
| Inlet                           | 32                                | 30                           |
| Exhaust                         | 28                                | 26                           |
| Fuel supply system              | Programmed<br>fuel injection      | Programmed<br>fuel injection |
| Ignition system                 | Dual point<br>sequential ignition | Single point<br>ignition     |
| Max. power (kW/rpm)             | 63/5 700                          | 49/5 500                     |
| Max. torque (N·m/rpm)           | 119/2 800                         | 103/2 500                    |

chamber. Also, to promote rapid combustion, a new intake port was adopted to strengthen mixture turbulence in the combustion chamber, as well as swirl and tumble. Furthermore, in order to decrease fuel consumption, friction was reduced through a full-scale review and revision of engine part shapes, changes to materials and improvements in machining and processing methods.

Fig. 5 shows the engine performance curve. Through the changes in the specifications in Table 1, improvements in volumetric efficiency through changes in valve lift characteristics, and tuning of the intake and exhausts, etc., the new engine achieves approximately 30% higher output than the previous engine (D13B).

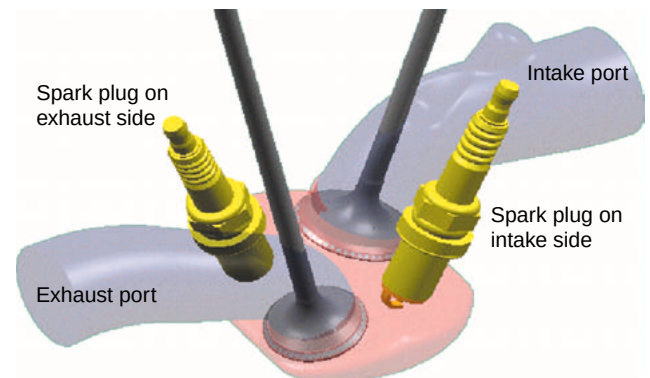


Fig. 3 Combustion chamber configuration

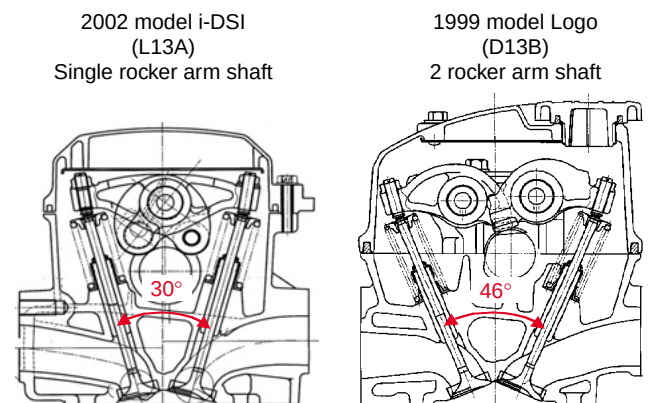


Fig. 4 Section view of cylinder head

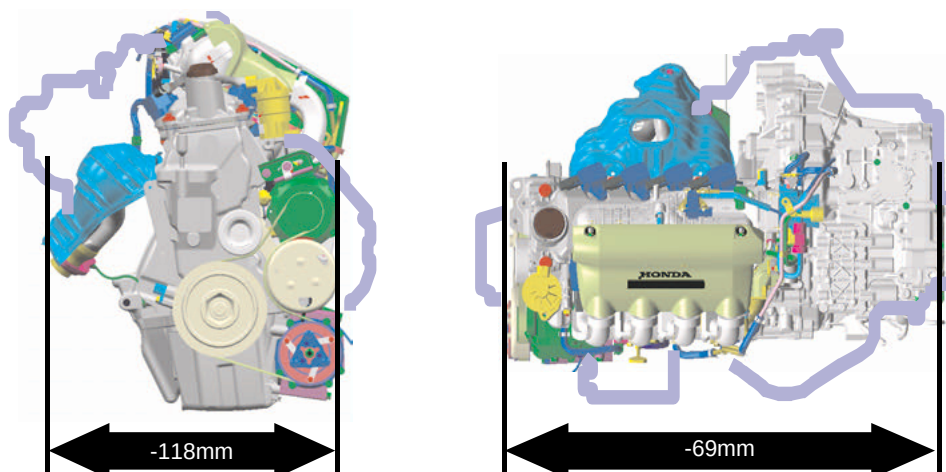


Fig. 2 Engine side and down views

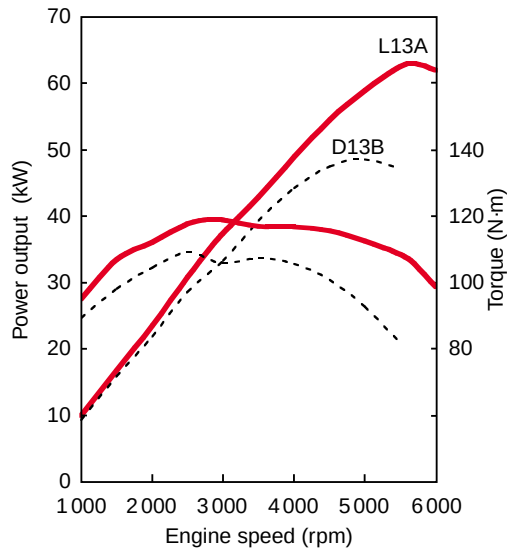


Fig. 5 Engine performance

#### 4. Combustion of i-DSI

The i-DSI combustion concept is one of rapid combustion throughout all operating ranges, using optimal control in response to engine speed and load conditions at ignition timing, using a dual-point spark method. For better homogenization of air-fuel mixture formation, the flow pattern within the combustion chamber (kinetic energy, turbulent energy) is made uniform through the intake port and combustion chamber shapes. With high precision control of the dual-point sequential ignition timing, decreased fuel consumption, knock suppression, high torque and NV (noise and vibration) performance improvements were all achieved.

##### 4.1 Modification of the Intake Port and Combustion Chamber

In order to raise the flame propagation speed, this engine's intake port is designed for higher swirl ratio ( $1.1 \rightarrow 1.36$ ) and tumble ratio ( $1.3 \rightarrow 1.58$ ) compared with a previous engine, and strengthens turbulent motion at the compression end (before ignition).

As well as the strength of in-cylinder turbulent motion, the air-fuel mixture homogeneity is an important factor in rapid combustion. Therefore, the combustion chamber is designed to attain the perfectly homogeneous air-fuel distribution. This distribution was studied by simulating the shape of the combustion chamber, with the focus on piston head shapes under the following conditions: Average air-fuel ratio  $A/F = 14.7$  Fuel injection timing: 20 degrees after TDC (top dead center) of the intake stroke. The simulation showed the horizontal sectional views at the piston of the electrode in the center of the spark plug, and the vertical sectional views at the spark plugs positions on the intake side and exhaust side.

Fig. 6 shows the computed combustion chamber air-fuel distribution at 30 degrees before TDC of the compression stroke. Apparently, it is not homogenous with the conventional design. Fuel-rich mixture exists on the intake spark plug side and around the bottom area of the combustion chamber. By adjusting the swirl and tumble ratio balance and shaping the combustion chamber, it was made possible to homogenize the turbulent energy distribution thus realizing the totally homogeneous air-fuel mixture.

Fig. 7 shows co-variation of indicated mean effect

Ne = 1500rpm  
P intake = -53kPa  
A/F (averaged) = 14.7  
End of injection timing = 20deg ATDC  
Crank angle = 30deg BTDC

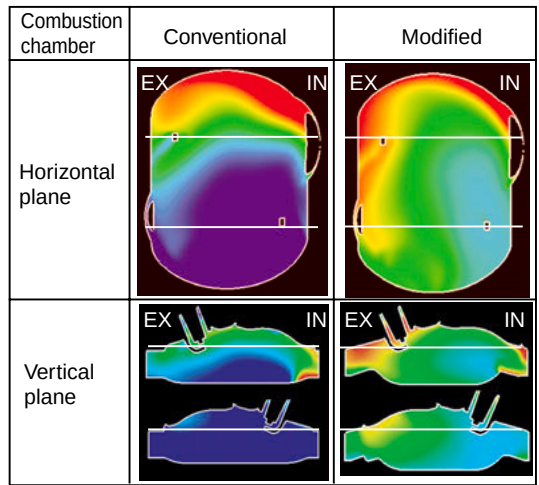


Fig. 6 Effect of combustion chamber shape modifications on in-cylinder A/F distribution

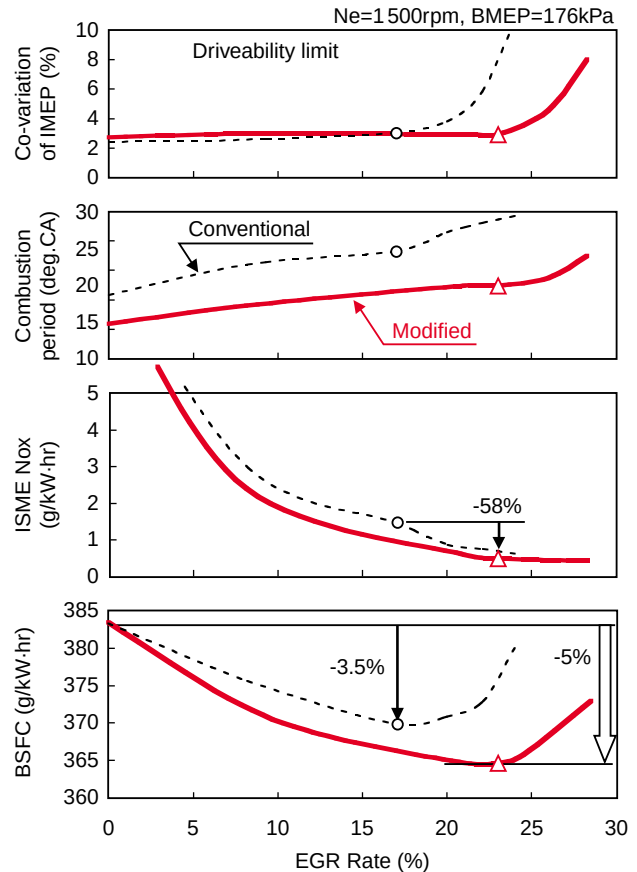


Fig. 7 Comparison of BSFC, NOx and combustion characteristics between the modified and conventional combustion chambers

pressure (IMEP), combustion period which means mass burned fraction (10%-90%), indication of specific mass emission-NOx (ISME-NOx) and BSFC in relation to the EGR (exhaust gas re-circulation) rate, both before and after the aforementioned in-cylinder air-fuel mixture formation modifications. By the rapid combustion with strengthened swirl and tumble as well as a more homogenous air-fuel

mixture, the stable combustion was achieved even with large-volume EGR rate of 23%. As a result, it is possible to reduce NO<sub>x</sub> emissions and to reduce fuel consumption by 5%.

#### 4.2 Rapid Combustion Through Sequential Ignition

Fig. 8 shows a comparison of the various losses as determined from BSFC and pressure, in terms of ignition timing for an engine speed of 1500rpm under a partial load. With intake and exhaust side dual-point simultaneous ignition control, there is a point where BSFC is decreased if the ignition timing is advanced. Furthermore, by carrying out sequential ignition timing control that advances ignition timing on the intake side, a 1.3% reduction in BSFC can be achieved. This means by promoting ignition first on the intake side, which has a comparatively low temperature, more rapid combustion is achieved than with simultaneous ignition. As a result, there is tendency for an increased cooling loss (heat loss) but there is a larger reduction in time loss. Consequently, there is an improvement in efficiency due to a reduction in overall loss.

Fig. 9 shows the results of pressure analysis at BSFC minimum ignition timing with both simultaneous ignition and sequential ignition, as well as an image of the indicator diagram. Compared with simultaneous ignition, sequential ignition shows a faster pressure rise and the maximum pressure is higher. As a result, the combustion period, in which the mass-burned fraction increases from 10% to 90%, is shortened by 1.8 degrees, and through the above-described rapid combustion, time loss is reduced. Sequential ignition has, in comparison with simultaneous ignition,

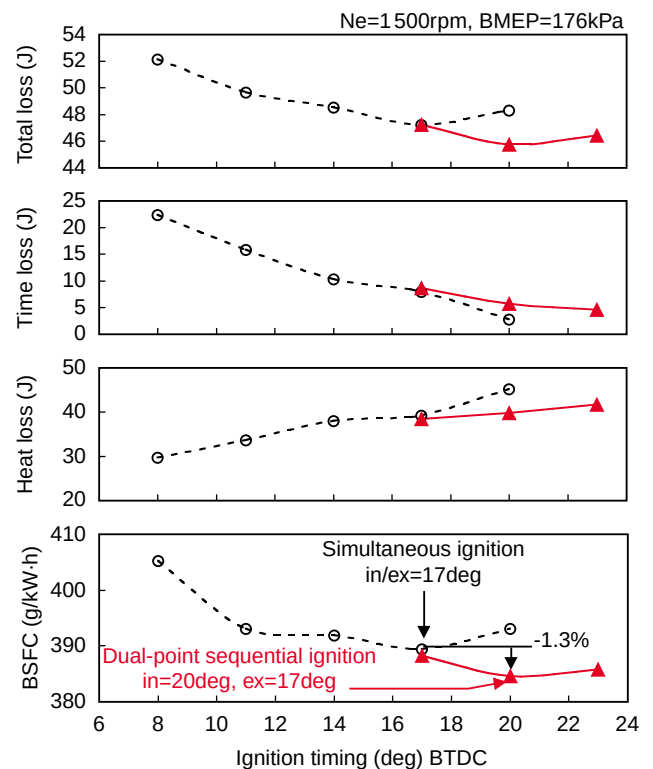


Fig. 8 Comparison of combustion characteristics

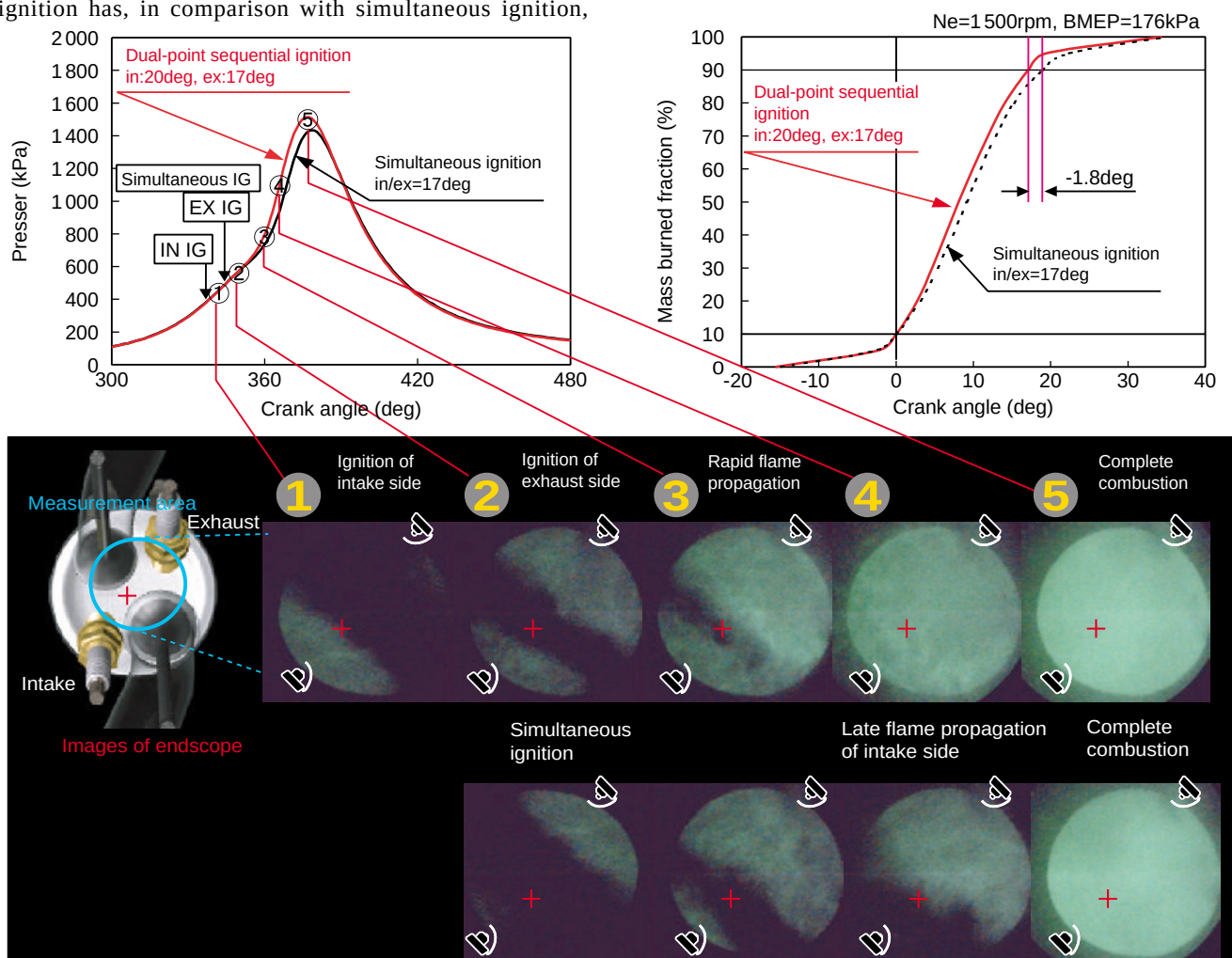


Fig. 9 Comparison of combustion characteristics

faster flame propagation, which is expressed in the image, and in the latter half of combustion, the flames on the intake and exhaust sides move to the center of the combustion chamber (in the visible range, offset to the lower left), which is an expression of the rapid combustion.

#### 4.2.1 Knock suppression

Creating a high compression ratio is a major factor in improving thermal efficiency. However, knocking is a large obstacle. A study was made of the use of dual plugs and ignition timing control in order to overcome this problem and to achieve both torque at a low speed range and a high compression ratio.

Fig. 10 shows the schematic view and an image of knock suppression through dual-point sequential ignition. The plug on the intake side is ignited first, and it is clear from the image that the flame grows from the intake side at this time. Next, if the plug on the exhaust side does not ignite, the non-combusted fuel in the air on the exhaust side end part receives this flame growth from the intake side, increases in temperature and pressure, and then self-combusts, which causes knocking. In contrast, if ignition of the plug on the exhaust side is retarded, the end gas combusts normally before knocking occurs and knocking can be suppressed.

Fig. 11 shows one example of the effectiveness of knock suppression. This figure shows knocking points and torque characteristics at an engine speed of 2000rpm and under a full load condition when ignition timing is changed. A comparison of the following three ignition methods was made: (1) ignition of intake side plug only, (2) dual-point simultaneous ignition, and (3) dual-point sequential ignition. In contrast with ignition on the intake side only, dual-point simultaneous ignition makes possible rapid combustion and advanced ignition timing due to knock suppression, and, further, a 6% improvement in torque is achieved. In addition, using sequential ignition, knocking can be suppressed and sequential ignition's set ignition timing torque is improved by 1.5% compared with that for simultaneous ignition.

Fig. 12 shows torque characteristics when simultaneous ignition and sequential ignition timing control are carried out. In the low engine speed range of 2500rpm or less, there is a 2N·m torque improvement with dual-point sequential ignition timing control.

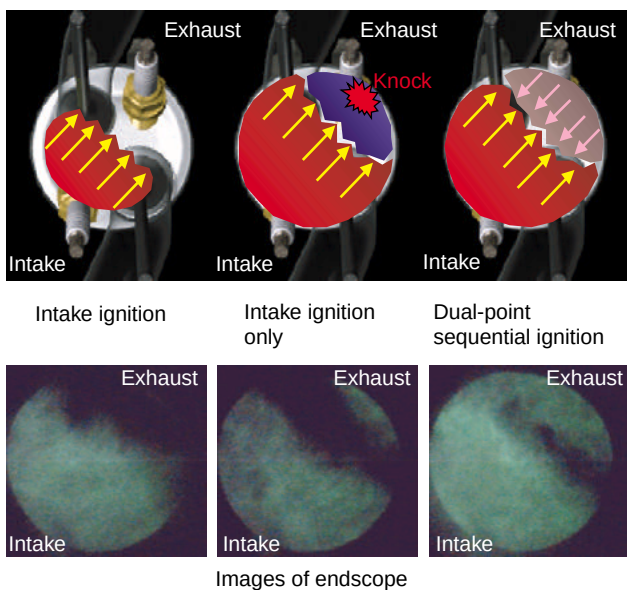


Fig. 10 Mechanisms of Improved knock suppression performance

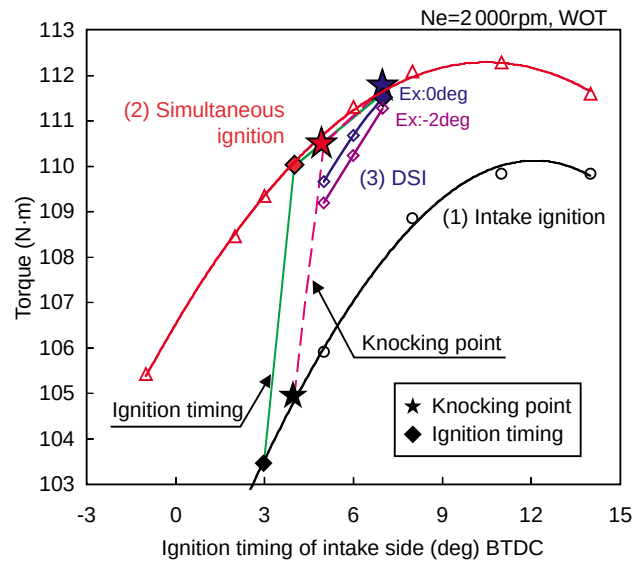


Fig. 11 Influence of ignition timing on engine torque

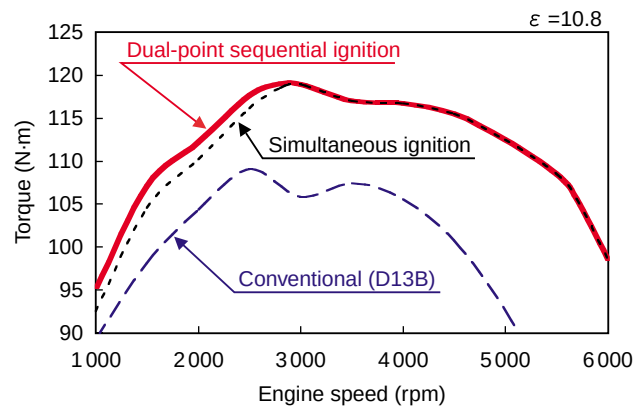


Fig. 12 Comparison of engine torque between dual-point sequential ignition and simultaneous ignition

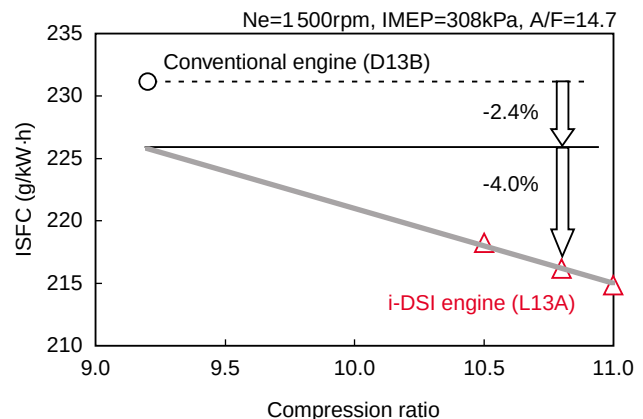


Fig. 13 Influence of compression ratio on ISFC

As a result of the knock suppression effect described above, it is possible to achieve both an improvement in torque at a low speed range, in comparison with existing engines, and the high compression ratio of 10.8. In combination with the rapid combustion gained due to the application of dual plugs and sequential ignition timing control, a 6.4% decrease in fuel consumption was achieved (Fig. 13).

#### 4.2.2 High-precision ignition timing control

Thanks to the rapid combustion gained through sequential ignition timing control, improvements in knock suppression and fuel consumption performance were achieved. However, this rapid combustion leads to aggravation of engine vibration and noise, particularly crank

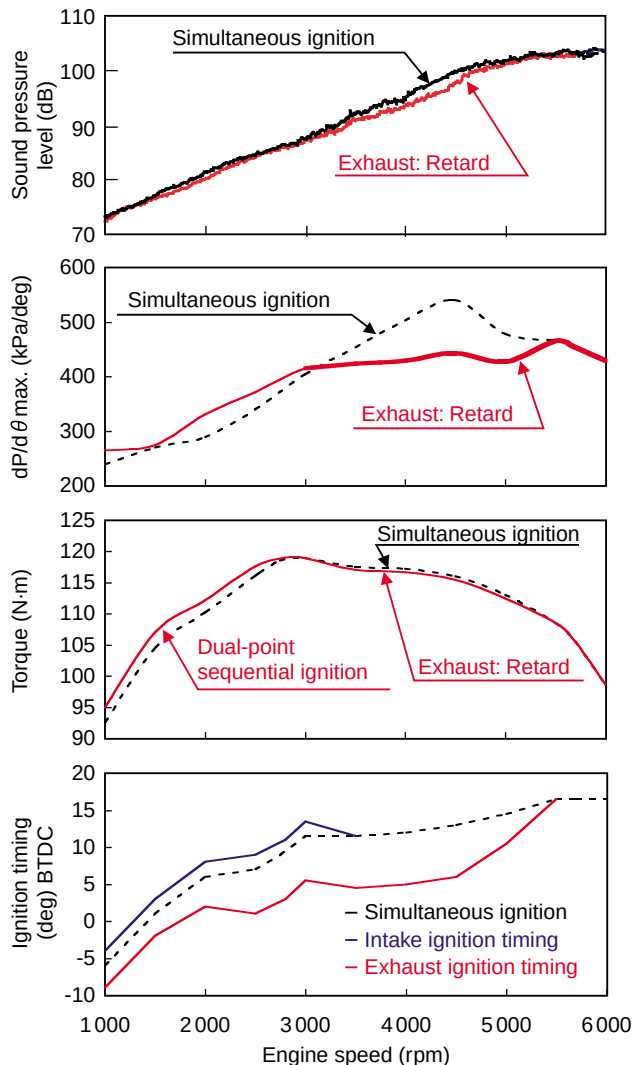


Fig. 14 Comparison of engine torque, combustion characteristic and radiation noise with control the ignition on consequential phase

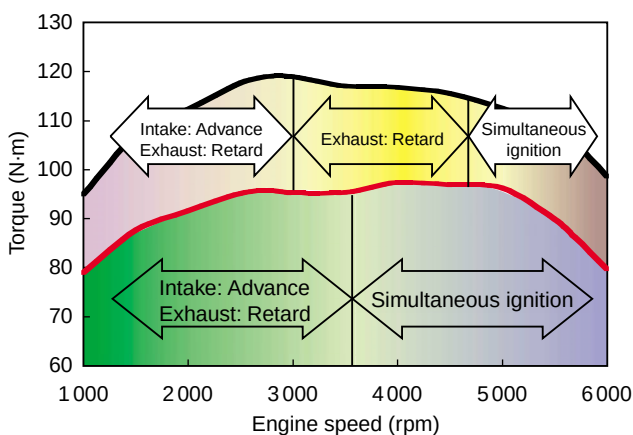


Fig. 15 Map of control the ignition on consequential phase

rattle. Crank rattle has good correlation with the maximum combustion pressure ( $P_{max}$ ), and the maximum pressure rise rate ( $dP/d\theta_{max}$ ). Thus a study was carried out into reducing noise and vibration, as well as achieving good torque characteristics by suppressing combustion pressure. Fig. 14 compares engine radiation noise, combustion pressure and torque characteristics during simultaneous ignition and sequential ignition timing control. By retarding ignition timing on the exhaust side and reducing the maximum pressure rise rate, a 1 – 2 dB reduction in the medium engine speed range is achieved. Torque also drops as combustion pressure is reduced. However, by keeping the drop percentage to a minimum at 0.4%, noise performance can also be achieved.

Fig. 15 shows an ignition timing control map. The ECU (electronic control unit) controls ignition timing according to atmospheric and engine conditions based on the three ignition timing controls: the range where the intake side plug is ignited first and the exhaust side plug has a retarded ignition, the range where the exhaust side plug only has a retarded ignition, and the simultaneous ignition range.

#### 5. Reduction of Friction

In addition to the rapid combustion outlined above, a reduction in friction is an important factor in achieving decreased fuel consumption.

In addition to the offset cylinder structure, roller follower rocker arm, ion plated piston ring, striated crank journal and connecting rod bearing metal which were developed as friction reducing technology for the ECA1 engine applied to 'Insight', other new technologies were adopted. These technologies include molybdenum disulfide ( $MoS_2$ ) shot peening of the piston skirt (first in the world), a two-mode cooling system, and a chain blade tensioner mechanism.

As for the piston skirt processing, there are the conventional molybdenum coating and the micro-dimple (MD) processing<sup>(3)</sup> adopted on the engine for 'Insight.' Here, a new friction reducing technology was developed which applies these process technologies. Fig. 16 shows the outline of piston skirt molybdenum disulfide ( $MoS_2$ ) shot peening. In this process, a very fine powder (mean diameter  $11\mu m$ , measurement method: sensing zone method) of the solid lubricant  $MoS_2$  is blown at the high pressure of 900kPa, and the heat thus generated causes the  $MoS_2$  to penetrate the piston surface layer. Fig. 16 shows an example of a view with TEM (transparent electron microscope). Through surface changes, molybdenum

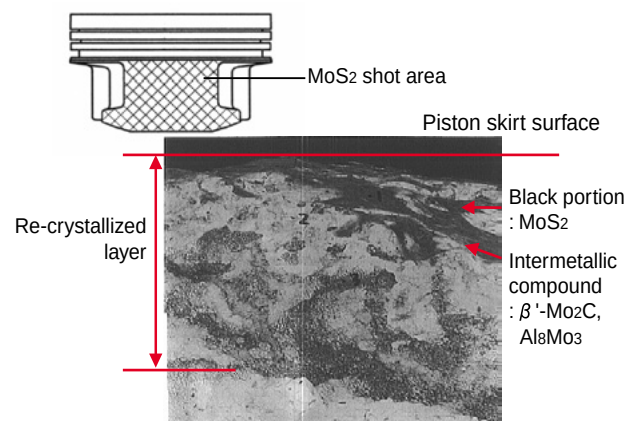


Fig. 16 Piston skirt surface treatment

disulfide gets into the base material and a  $6\mu\text{m}$  recrystallized layer is formed. In addition, an intermetallic compound, which is thought to be caused by the partial thermal expansion phenomena, can be found. Because with the conventional molybdenum coating process an adhesive component is necessary, it is difficult to raise the molybdenum purity level. In comparison, the newly developed  $\text{MoS}_2$  shot does not have an adhesive component and is a highly pure  $\text{MoS}_2$  shot. Thus, it serves to reduce friction loss and improve lubrication. Consequently, compared with the conventional molybdenum coating there is a friction reduction of 3.2%, and compared with the MD process there is a reduction of 2%. (Fig. 17)

A two-mode cooling system was adopted for the cooling method. Fig. 18 shows the coolant circulation condition when the engine is cold and after warming, according to different cooling methods. When the engine is cold, coolant flows to the cylinder head side, with the aim of increasing the block warm-up speed, in order to reduce friction through the rise in engine oil temperature. Also, after the engine is warm, the warm water that has cooled the cylinder head flows to the block side, maintaining its high temperature and thereby reducing friction. Cooling the cylinder head side first after the engine is warmed up also contributes to the suppression of knocking. Fig. 19 shows a comparison of engine oil temperatures from a cold engine through the warm-up process. Compared with conventional cooling systems, this two-mode cooling system provides a warm-up improvement of  $4.2^\circ\text{C}$  at 100 sec. after the start of

driving. Fig. 20 shows the relationship between engine oil temperature and friction. The rise in engine oil temperature during the warm-up process gives a friction reduction of approximately 3%.

As for the chain blade tensioner, the structure is such that 5 layers of blades directly push on the chain to maintain tension. Compared with hydraulic tensioners, this system is characterized by there being no unnecessary tension applied to the chain thereby resulting in a reduction in friction.

In addition to these friction-reducing technologies, other technologies have been adopted, such as valve spring load reduction, piston ring tension reduction, and low viscosity engine oil (0W-20/SL/GF3).

Total engine friction is shown in Fig. 21. A 13%

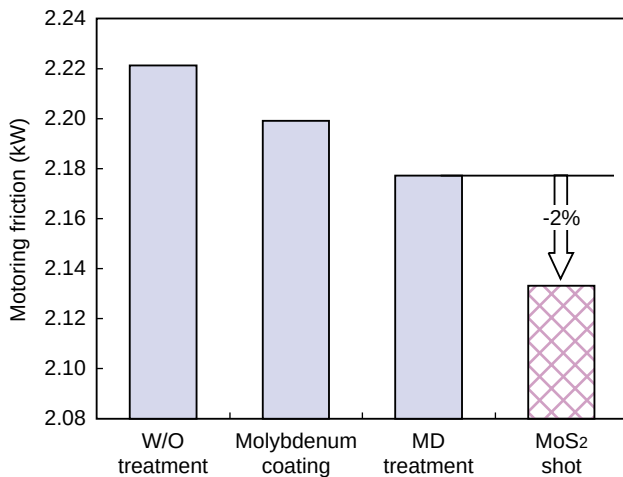


Fig. 17 Improvement of friction loss with piston skirt treatments

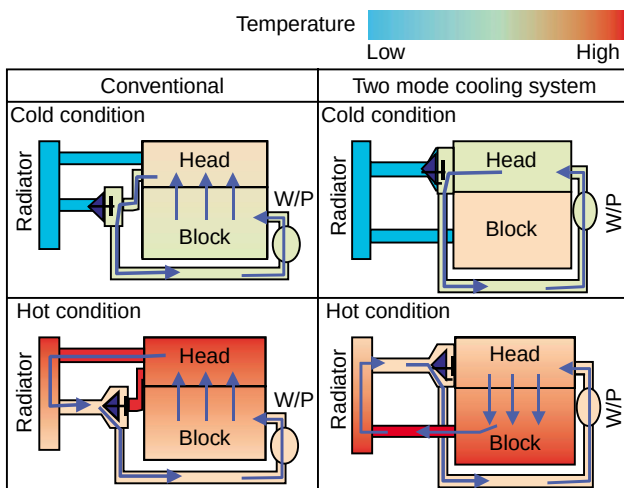


Fig. 18 Coolant of two mode cooling system

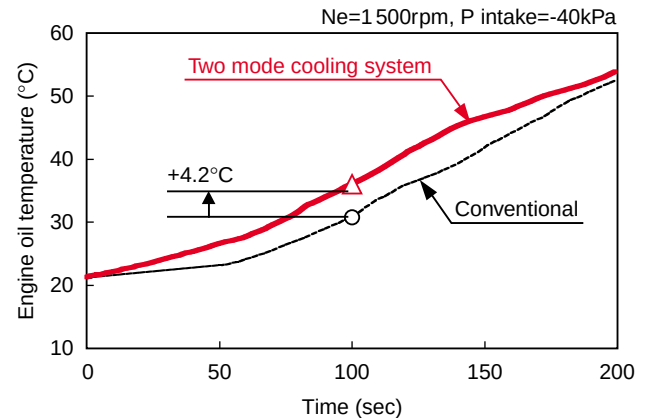


Fig. 19 Characteristics of engine oil temperature

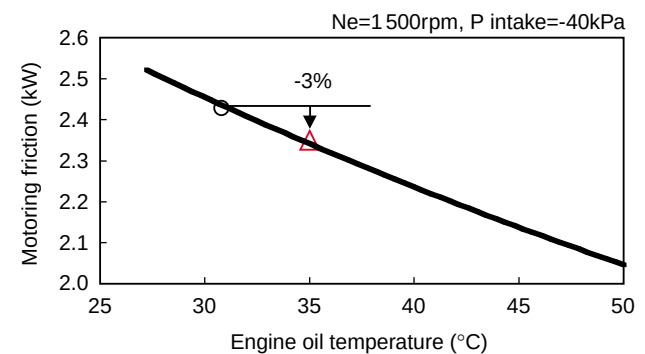


Fig. 20 Influence of engine oil temperature on motoring friction

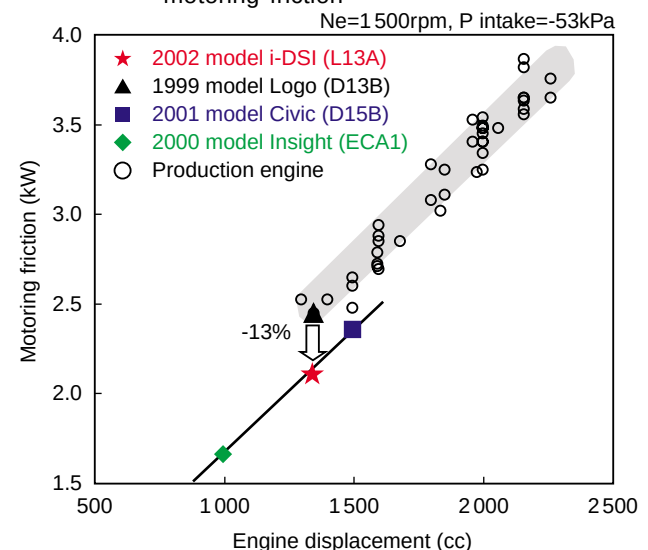


Fig. 21 Friction loss comparison

reduction in friction, as compared with previous engines, was achieved through the adoption of the above-described friction-reducing technologies.

## 6. Engine Fuel Consumption and Vehicle Fuel Economy

### 6.1 Engine Fuel Consumption

Fig. 22 shows the relationship between engine displacement and BSFC. By adopting the combustion improvement technologies of dual-point sequential ignition timing control and a higher compression ratio, as well as the friction-reducing technologies described above, this engine achieves an 11% BSFC reduction at stoichiometric combustion compared with previous engines, to thereby achieve a BSFC equivalent to existing lean burn engines. In addition, with combustion that introduces EGR (exhaust gas re-circulation), there is a further 5% BSFC reduction, so that a total BSFC reduction of 16% is provided.

### 6.2 Vehicle Fuel Economy

Fig. 23 shows the 10-15 mode fuel economy of the vehicle weight of Japanese domestic automobiles. 'Fit' has achieved a 28% improvement in fuel economy compared

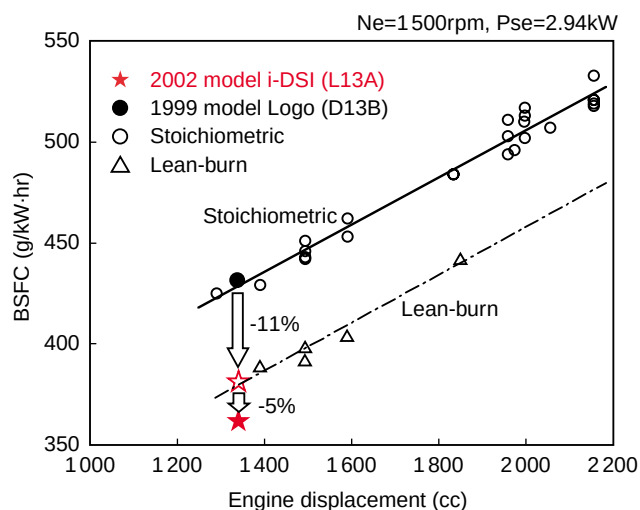


Fig. 22 Improvement of BSFC

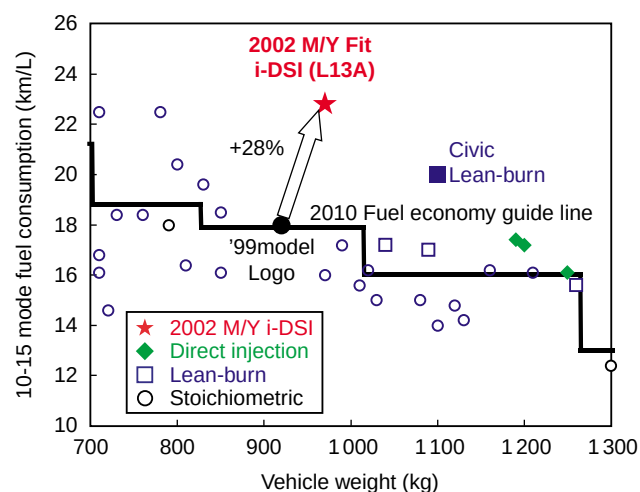


Fig. 23 Relationship between vehicle weight and 10-15 mode fuel economy in production gasoline vehicles

with the '99 model 'Logo' mounted with a previous engine. The improvement consists of the enhancement of the engine efficiency including control optimization (15%), the application of a newly developed CVT (10%), and the development of a lightweight and low friction body (3%). Achieving 23 km/L in the Japanese 10-15 mode that passes far beyond the 2010 fuel economy standard (17.9 km/L for vehicle weight of 828 to 1015 kg), the Fit deserves the category equivalent to the smallest size (Japanese K-class) vehicles.

## 7. Low Emissions Technology

### 7.1 Exhaust System

Fig. 24 shows the exhaust system. By making the exhaust manifold out of lightweight stainless steel tubes, heat mass is reduced. This, in combination with the rearward exhaust system, results in the early activation of the catalytic converter, thereby improving the catalyzing efficiency in the cold range. In the three-way catalytic converter placed under the floor, oblique introduction of exhaust gases is adopted. As the gases come into contact with the surface of the cells obliquely, the contact surface

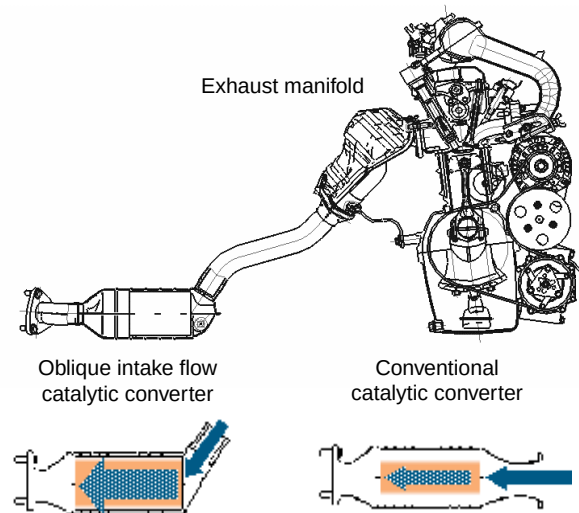


Fig. 24 Configuration of exhaust system

area is increased, and exhaust gas contact is more uniform, catalyzing efficiency is increased.

### 7.2 Multi-hole Injector

Fig. 25 shows the multi-hole injector adopted in this engine. This injector is 17% shorter in length and 58% wider than conventional injectors. In addition, in comparison with the conventional pintle type, a six-hole injection nozzle was adopted. As a result, spray atomization and the amount of gasoline adhesion to the intake port wall is reduced, so that a 5% reduction in HC emissions is achieved from a cold start in the 11 mode (Fig. 26).

### 7.3 Emissions Achievements

In addition to the emissions reducing technologies outlined above, the adoption of such measures as a reduction in feed gas through dual-point sequential ignition and high-precision electronic EGR valve control, it is possible to achieve the low fuel consumption of 23 km/L in the 10-15 mode, and a 50% reduction in emissions levels compared with the FY2000 regulations (Fig. 27).

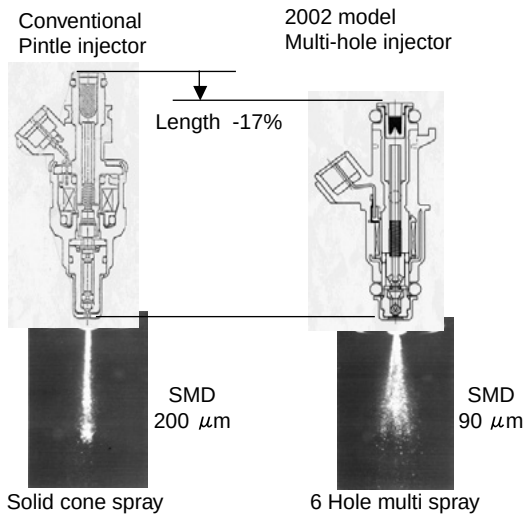


Fig. 25 Comparison of fuel injector between conventional and new one

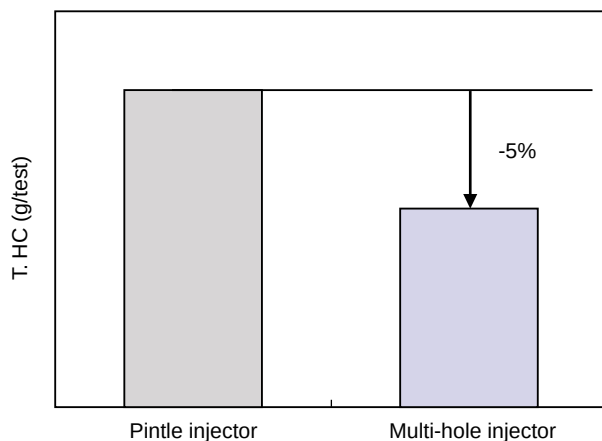


Fig. 26 Comparison of HC emissions in the 11mode between the multi-hole injector and the pintle injector

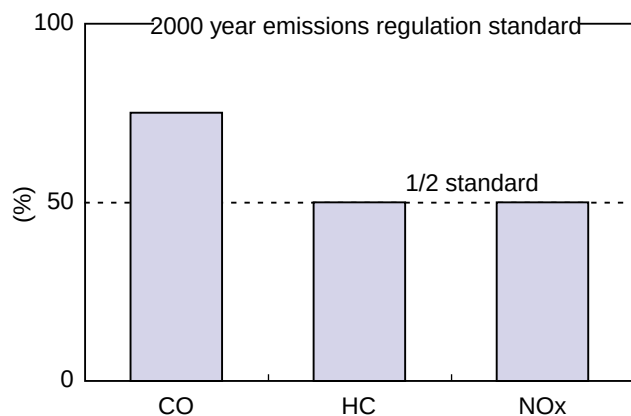


Fig. 27 10-15mode emission

## 8. Summary

Development of a lightweight compact engine with the i-DSI combustion improvement and the whole friction reduction leads to the followings.

- 1) Rapid combustion is made possible through the use of dual-point sequential ignition and uniform air-fuel mixture formation achieved with the new intake port and compact combustion chamber. It is also possible to achieve a compression ratio of 10.8 even with regular gasoline. These achievements, in combination with the new friction-reducing technologies, have resulted in the engine attaining a specific fuel consumption reduction of 16%.
- 2) Through high-precision control of dual-point sequential ignition in response to engine speed and load conditions, it is now possible to achieve good output, torque, and fuel economy as well as noise and vibration reductions.
- 3) With the adoption of the new L13A engine – with improved combustion and reduced friction– as well as the newly developed small size CVT and high efficient body technologies, a fuel economy level of 23 km/L in the 10-15 mode was achieved. In addition, with the adoption of the oblique exhaust gas intake flow three-way catalytic converter and the rearward exhaust system, a low emissions level, equivalent to a 50% reduction compared to the FY2000 regulations, was achieved (J-LEV certification).

## 9. Afterword

In order to respond to future environmental changes and customer needs, this newly developed L13A engine has a structure that can be used in derivative engines with larger or smaller capacities. It is firmly believed that this engine will be adopted as the core of our small vehicle strategy in Europe in the future, and that it will come to be widely used as our next generation power train, which has the potential to provide excellent fuel consumption properties and to meet EU2005 regulations.

It is not enough to be satisfied with the current level of refinement, but rather we plan to continue to make further enhancements, seeking to make technological improvements that will meet the demands of the coming era.

Finally, the authors would like to express their deepest appreciation and gratitude to those, both in the company and outside, who gave their cooperation in the development of this engine.

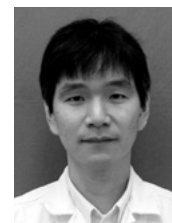
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