

# Development of DOHC VTC 0.66 L Gasoline Engine for New K-car

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## ABSTRACT

A lightweight, compact DOHC VTC 0.66-L 3-cylinder gasoline engine was developed for use in K-cars, achieving a balance of environmental performance (low fuel consumption and low emissions) and high power output. The new model engine was developed on the assumption that it would be combined with a newly developed continuously variable transmission (CVT) for k-cars. It was accordingly given a long-stroke structure and variable timing control (VTC) on the intake side as well as technology for enhancement of combustion and friction reduction. These measures achieved high power with maximum torque of 65 Nm and maximum power of 43 kW in the naturally aspirated engine together with 10% enhancement in specific fuel consumption over the previous engine model. In the turbocharged engine, use of a higher compression ratio in addition to fuel consumption enhancement technology in common with that of the naturally aspirated engine achieved a 10% enhancement in specific fuel consumption over the previous engine model as well as maximum torque at the high level of 104 Nm.

## 1. Introduction

Emissions reduction technology for protection of the environment and low fuel consumption technology for reduction of CO<sub>2</sub>, which is a cause of global warming, have become important in the development of motor vehicle engines in recent years. There has also been rising demand in Japan for k-cars because of sharply rising crude oil prices and other such factors, and raising the level of environmental performance is an urgent task. As the technologies adopted for these purposes, the engine used in the 2004 model of the LIFE<sup>(1)</sup>, a k-car, incorporates the short-stroke structure for friction reduction and the dual and sequential ignition (i-DSI)<sup>(2)</sup>, also adopted in the 2002 model Fit, to provide more rapid combustion and a higher compression ratio by having two spark plugs for each cylinder. Although the short-stroke type of i-DSI engine has an enlarged cylinder bore diameter that is not advantageous in terms of heat loss and knocking, it is advantageous in terms of fuel consumption due to the lower friction when combined with a four-speed automatic transmission that makes intensive use of relatively high speed domains. Achieving further enhancement of fuel consumption, however, will require operation at lower speeds and enhancement of brake specific fuel consumption (BSFC). To that end, the new model engine was aimed for enhancement of low-speed torque and enhancement of brake specific fuel consumption under high load. Development was therefore carried out so as to effectively achieve enhanced fuel consumption when combined with a

CVT. The following will report on the technology adopted in the new model engine.

## 2. Development Goals

In considering engine structure, the ways by which the environmental performance of the new model engine could be enhanced were studied. Figure 1 shows the engine's

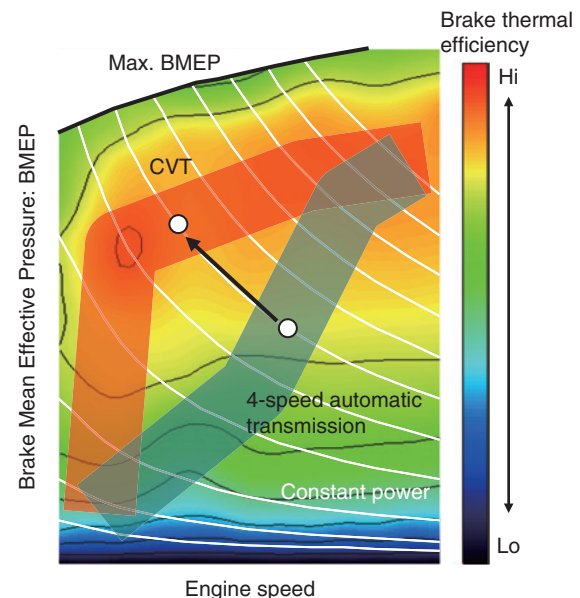


Fig. 1 Brake thermal efficiency map and typical-use domain under 10-15 mode operation of 4-speed automatic transmission and CVT

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Table 1 Engine specifications

|                                 |     | Developed model       |                       | Previous model      |                     |
|---------------------------------|-----|-----------------------|-----------------------|---------------------|---------------------|
| Air intake system               |     | Normal aspiration     | Turbo charged         | Normal aspiration   | Turbo charged       |
| Cylinder configuration          |     | In-line 3-cylinder    | In-line 3-cylinder    | In-line 3-cylinder  | In-line 3-cylinder  |
| Bore × stroke (mm)              |     | 64 × 68.2             | 64 × 68.2             | 71 × 55.4           | 71 × 55.4           |
| Displacement (cm <sup>3</sup> ) |     | 658                   | 658                   | 658                 | 658                 |
| Compression ratio               |     | 11.2                  | 9.2                   | 11.2                | 8.5                 |
| Valve train                     |     | DOHC intake VTC       | DOHC intake VTC       | SOHC i-DSI          | SOHC i-DSI          |
| Number of valves                |     | 4 per cylinder        | 4 per cylinder        | 2 per cylinder      | 2 per cylinder      |
| Valve diameter (mm)             | in. | 24.5                  | 24.5                  | 35.5                | 35.5                |
|                                 | ex. | 20.5                  | 20.5                  | 30                  | 30                  |
| Ignition system                 |     | Single-point ignition | Single-point ignition | Dual-point ignition | Dual-point ignition |
| Max. power (kW/rpm)             |     | 43/7300               | 47/6000               | 38/7000             | 47/6000             |
| Max. torque (Nm)                |     | 65/3500               | 104/2600              | 60/3600             | 93/4000             |

brake thermal efficiency map overlaid with typical-use engine speed and load domains for the four-speed automatic transmission and CVT operating in 10-15 mode. Adoption of the CVT enables lower-speed, higher-load operation with the same driving force and increases the frequency of use of domains with greater brake thermal efficiency. Consequently, adoption of the CVT alone enables enhancement of fuel consumption even with the same engine, but the new model engine was aimed to enhance performance in the low-speed domain so it could produce still greater fuel consumption enhancement effect. Figure 2 shows the brake specific fuel consumption (BSFC) of the engine measured at the same brake horsepower and speed in the previous model engine together with the pumping mean effective pressure (PMEP) and friction mean effective pressure (FMEP). As the engine speed drops, the PMEP and FMEP also drop, enhancing the BSFC. Below a certain speed, however, no further enhancement is obtained. That is due to the retarded ignition timing used for knocking avoidance, and knocking enhancement has the potential for

further enhancement of fuel consumption. Furthermore, even in cases when there is no necessity to avoid knocking, the use of lower engine speeds is constrained by the limitations on charging efficiency. It is apparent, therefore, that the maximum fuel consumption impact from low-speed operation can be obtained effectively by knocking enhancement and charging efficiency enhancement in low-speed domains, which is to say, by enhancement of low-speed torque.

Given the above, the goals for the new engine aimed at enhancement of fuel consumption performance were defined by the following items:

- (1) Achieve a balance of enhanced low-speed torque and maximum power; and
- (2) Further enhancement of brake thermal efficiency.

### 3. Engine Overview and Main Specifications

In order to achieve the performance goals for the new model engine, the following kinds of engine structure specifications and main technologies were applied.

- (1) Adopt continuous variable valve timing control (VTC) mechanism on the intake side
  - Enhance charging efficiency in the low to medium speed domains through the scavenging effect
  - Enhance thermal efficiency by internal exhaust gas recirculation (EGR)
- (2) Long-stroke structure
  - Enhance knocking by strengthening the in-cylinder flow and thermal efficiency by increasing the combustion speed
  - Enhance thermal efficiency by decreasing the surface/volume ratio (S/V ratio)
- (3) DOHC four-valve cylinder head
  - Achieve balance of small bore and maximum power
  - Adopt Tumble port
- (4) Reduction of friction in every part

Table 1 shows the main specifications compared with the previous engine model, and Fig. 3 shows an

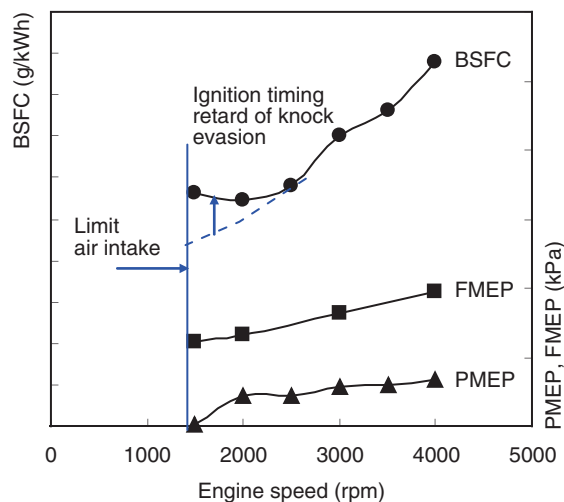


Fig. 2 Effect of engine speed on BSFC, PMEP and FMEP

external view of the engine. The new model engine was given a long-stroke structure for the purpose of enhancing combustion, but steps were also taken to make it compact, without increasing its height relative to the previous model.

Figure 4 shows a sectional view of the cylinder head. The new model engine was given roller rocker arms for reduced friction as well as hydraulic lash adjusters to make it maintenance-free, and the size of the cylinder head was reduced.

Figure 5 shows a comparison of the combustion

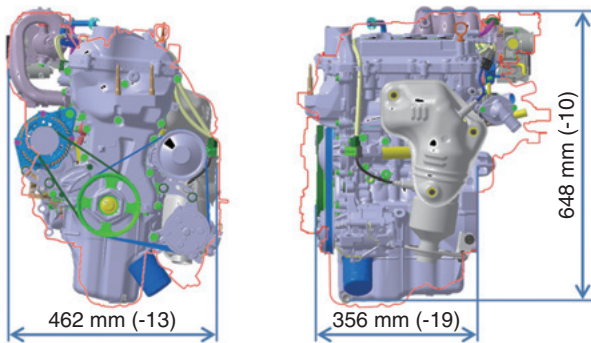


Fig. 3 View of new engine and comparison of engine size

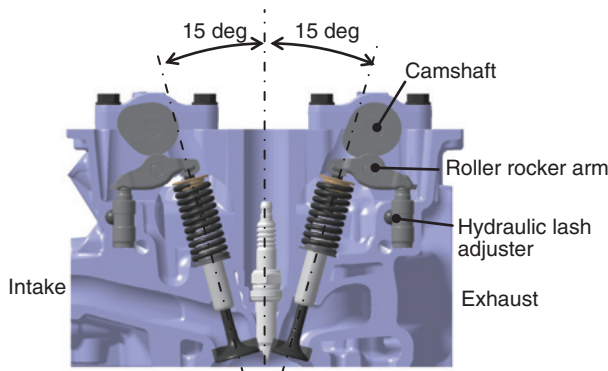


Fig. 4 Sectional view of cylinder head

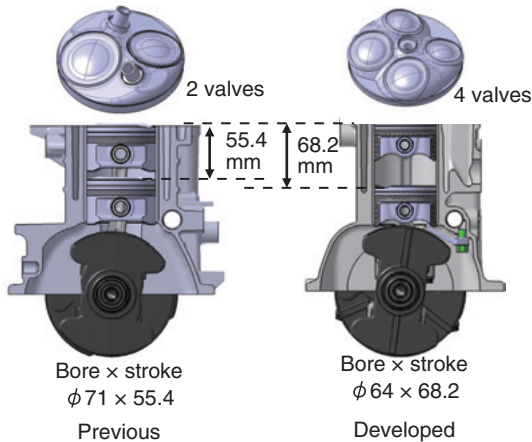


Fig. 5 Comparison of combustion chamber configuration of previous and developed engine

chamber configuration with the previous model. Where the previous model had a short stroke with a stroke-bore ratio (S/B ratio) of 0.78, the new model engine was given a longer stroke, with an S/B ratio of 1.07, to reduce the flame propagation distance and strengthen the in-cylinder flow. The limitation on valve diameter due to the smaller bore was addressed by changing from the previous two valves to four valves. Tumble ports were used for the intake ports as a way of strengthening the in-cylinder flow. Taking the conceptual approach of F1 technology for the shape of the combustion chamber, a conical shape was formed concentrically around each valve to control the flow, providing a more advantageous shape than the pent-roof of the previous model in terms of S/V ratio and flow coefficient.

## 4. Power Output Characteristics

### 4.1. Naturally Aspirated Engine

Figure 6 shows the power output characteristics at wide-open throttle in comparison with the previous model. The new model engine has VTC and optimized intake and exhaust system specifications that yield higher torque than the previous model across all speed domains. It achieves 8% torque enhancement at maximum torque and 18% at a speed of 2000 rpm, obtaining the stronger low-speed torque that was the aim. The maximum power is enhanced by 13% over the previous model, achieving a balance of low-speed torque and maximum power.

#### 4.1.1. Charging efficiency enhanced by VTC

Figure 7 shows measurements of intake and exhaust

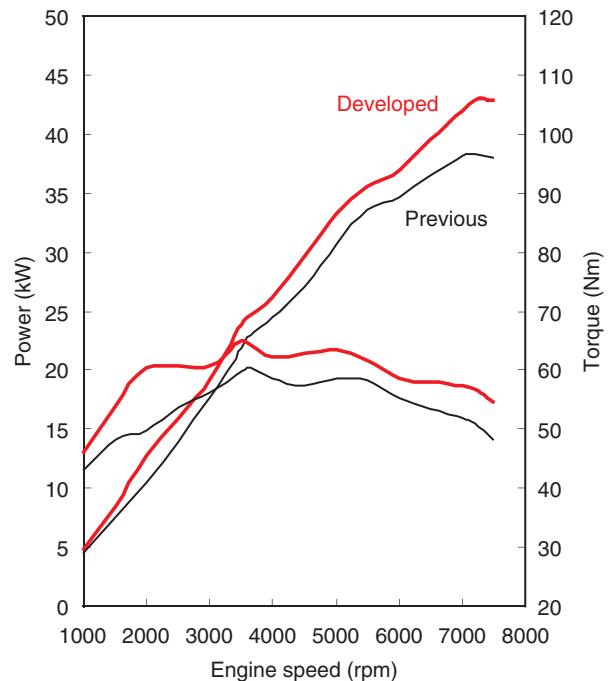


Fig. 6 Engine performance of normal aspirated engine

pulsation at the torque peak of 3500 rpm with the throttle wide open. The timing for opening the intake valves is controlled by the intake VTC timing advance action (Fig. 7A), and the timing for closing the exhaust valves (Fig. 7B) is retarded to the limit for providing stable combustion under low load, as found when idling. This extends the period when the exhaust port pressure is exceeded by the intake port pressure during overlap (Fig. 7C), enhancing the scavenging effect. Figure 8 shows charging efficiency characteristics when operating with the throttle wide open. Due in part to the effect that changing to a four-valve configuration has in high-speed domains, this engine achieves enhanced charging efficiency relative to the previous model across all speed domains.

#### 4.1.2. Knocking enhancement

In addition to strengthened in-cylinder flow due to the long-stroke structure, the adoption of piston oil jets and the change of coolant circulation method yielded knocking enhancement. Figure 9 shows the ignition timing characteristics as measured with varying charging efficiency at an engine speed of 2000 rpm in comparison with the previous model. The new model engine has been able to shift charging efficiency some 10 points more toward the high-load side than the previous model at the point where the knocking ignition timing begins to be retarded beyond

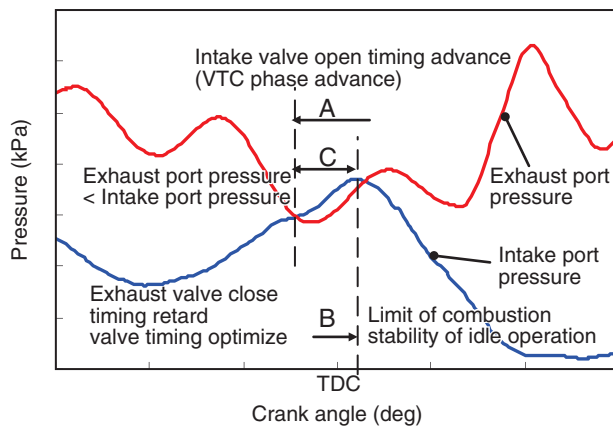


Fig. 7 Pulsation of intake port and exhaust port, and valve timing

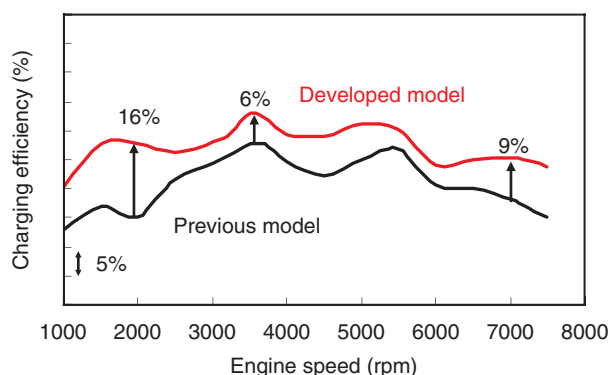


Fig. 8 Comparison of charging efficiency

the minimum advance for best torque (MBT).

Figure 10 shows the piston oil jet shape. The oil from the oil jet has been directed so as to actively cool the exhaust side behind the piston. Unlike the conrod jet in the previous engine that sprays oil intermittently, here the constant spray of oil increases the buildup of oil behind the piston, enhancing the cooling effect on the piston and enabling the knocking ignition timing to be advanced by one degree.

Figure 11 shows the coolant circulation system by comparison with the previous model. In the previous model, the total amount of coolant discharged by the water pump entered the cylinder heads, after which a portion of the coolant was distributed to the cylinder block. The coolant on the sides of the cylinder block flowed in from the cylinder-head side, which is to say from above, so that there would be a stronger flow at the bottom of the water jacket than in the vicinity of the head gasket. In the new model engine, the coolant discharged from the water pump

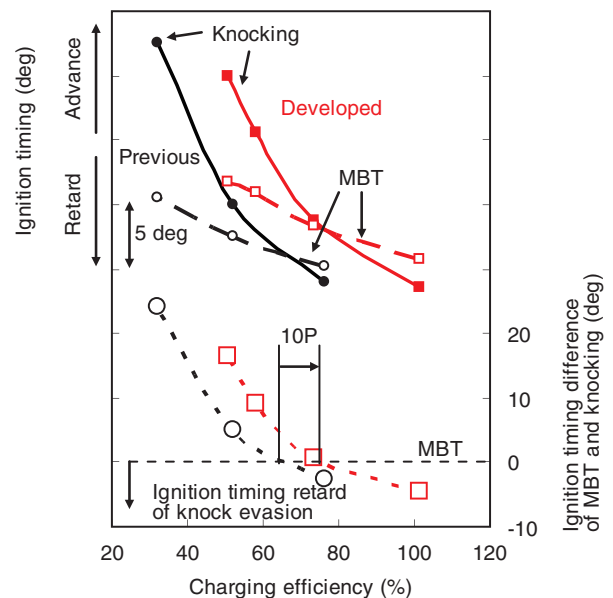


Fig. 9 Characteristics of ignition timing versus charging efficiency

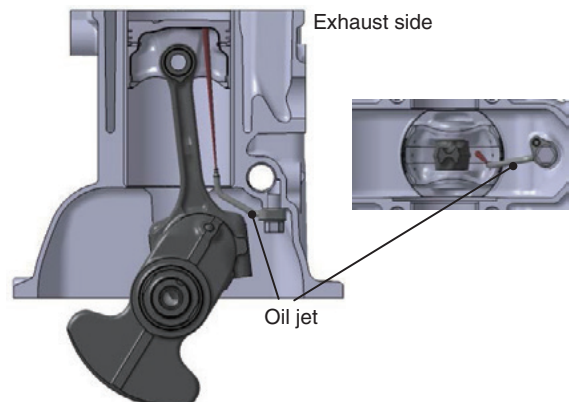


Fig. 10 Piston oil jet

is distributed simultaneously to the cylinder block and the cylinder head. The coolant flowing in the cylinder block enters the cylinder block directly from the water pump, and the coolant is thrown up to the upper areas by the ramp-shaped structure built into the water jacket.

This allows an active flow of coolant to be directed to the vicinity of the gasket surface, heightening the cooling effect on the combustion chamber and yielding knocking enhancement. Also, as shown in Fig. 12, the aluminum portion around the sleeve was made thinner, only on the upper part of the water jacket in the cylinder block, to provide active cooling of the combustion chamber.

#### 4.2. Turbocharged Engine

Figure 13 shows the power output characteristics at wide-open throttle in comparison with the previous model. The new model engine was equipped with intake VTC, which increased the compression ratio from the previous 8.5 to 9.2 while achieving a 12% enhancement at maximum torque. This further enabled enhancement of the torque start-up characteristics.

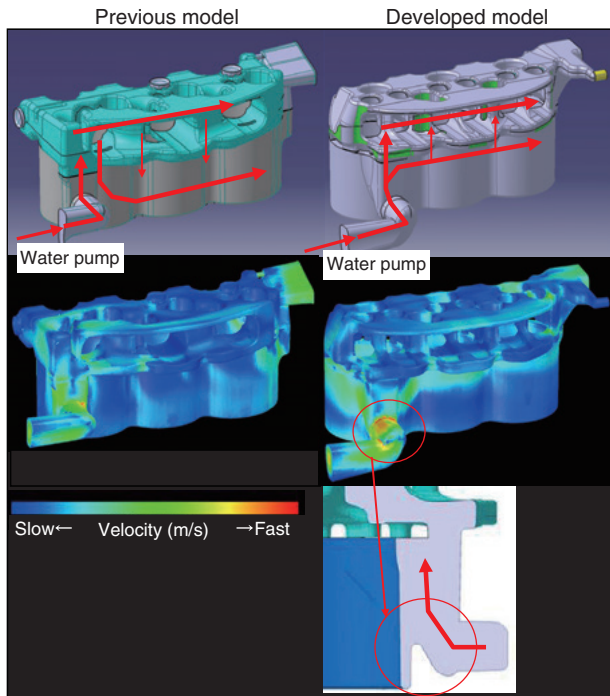


Fig. 11 Cooling circulation system

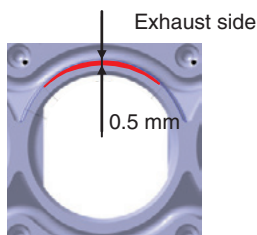


Fig. 12 Enhanced shape of water jacket

#### 4.2.1. VTC knocking enhancement effect

Figure 14 shows the relationship of boost pressure and ignition timing to charging efficiency at an engine speed of 3000 rpm with and without VTC. As with a naturally aspirated engine, the adoption of VTC increases the scavenging effect, enabling a reduction of about 20 kPa in boost pressure at the same charging efficiency. Reducing the boost pressure results in lower intake temperature and yields knocking enhancement together with reduction in the amount of residual gas. This enabled an increase in the compression ratio from 8.5 in the previous model to 9.2, enabling achievement of a balance between high torque and fuel consumption performance.

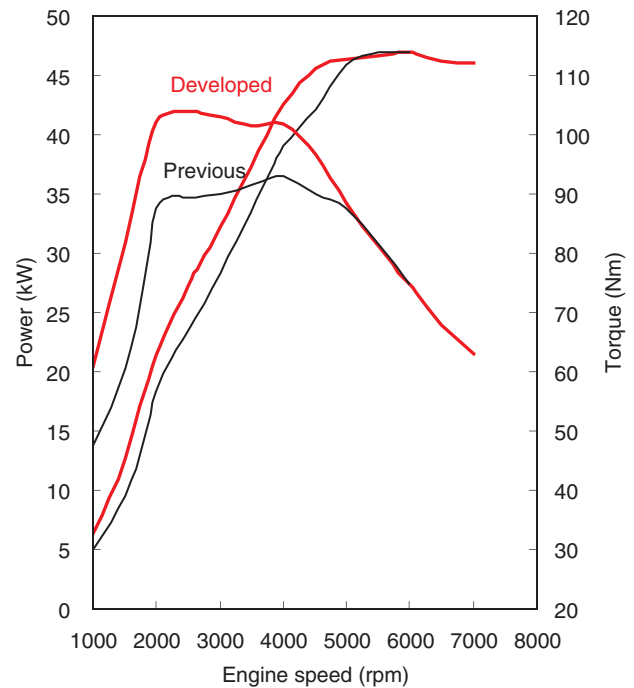


Fig. 13 Turbocharged engine performance

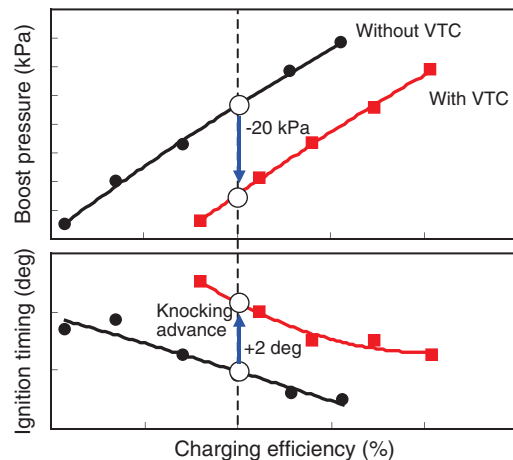


Fig. 14 Comparison of boost pressure and ignition timing between engines with and without VTC

#### 4.2.2. Boost pressure start characteristics enhanced by VTC

Enhancement of the scavenging effect in low-speed domains due to the adoption of VTC also contributes to enhancement of transient boost pressure start characteristics. Figure 15 shows transient characteristics when acceleration of a completely built vehicle from a standing start with wide-open throttle is simulated on the bench. Due to VTC, the time to reach target boost pressure was shortened about 18% and the engine speed rises more quickly, contributing to enhancement of standing start performance in completely built vehicles.

#### 4.2.3. Measures addressing exhaust soot

The greatest issue for adoption of VTC in a turbocharged engine was the increase in exhaust soot caused because the advanced intake valve timing increased the overlap. Figure 16 shows smoke and other particulate matter density when measures to be described below are taken to address soot and other particulate matter and when a previous model engine without VTC is operating at wide-open throttle power, both with VTC angle advanced to increase charging efficiency. When the VTC angle is advanced in order to increase the scavenging efficiency, there is an increase in the amount of blow-by from fuel injected into the intake port and the amount of soot and other particulate matter increases. The increase in soot and other particulate matter can be avoided by limiting the VTC advance angle, but this will also lower the charging efficiency, so it is desirable to keep the limitation on the advance angle to a minimum. This was addressed by

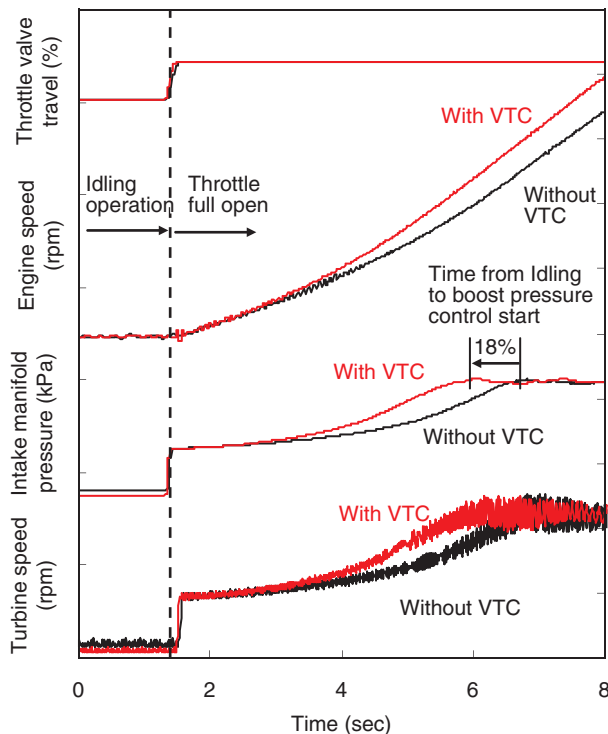


Fig. 15 Transient performance comparison with and without VTC

increasing the pressure of fuel injection from 294 kPa to 343 kPa and shortening the injection time while also optimizing the injection timing. This reduced the amount of fuel injected before overlap, which reduced the amount of blow-by without placing any limit on the VTC advance angle, and achieved a balance between reduction in the amount of exhaust soot and increase in charging efficiency.

### 5. Brake Specific Fuel Consumption Performance

Due to VTC, both the naturally aspirated version and the turbocharged version of the new model engine achieved enhancements over the previous model engines of 4% in internal EGR and 6% in combustion enhancement and friction reduction, for a total enhancement of 10% in specific fuel consumption. Figure 17 shows BSFC characteristics at 2000 rpm in the previous model and new model of naturally aspirated engine.

#### 5.1. Enhancement of Indicated Thermal Efficiency

Figure 18 shows the cylinder pressure diagram for the previous model engine and for the new model engine with the VTC angle not advanced, both at 2000 rpm and an indicated mean effective pressure (IMEP) of 710 kPa. The

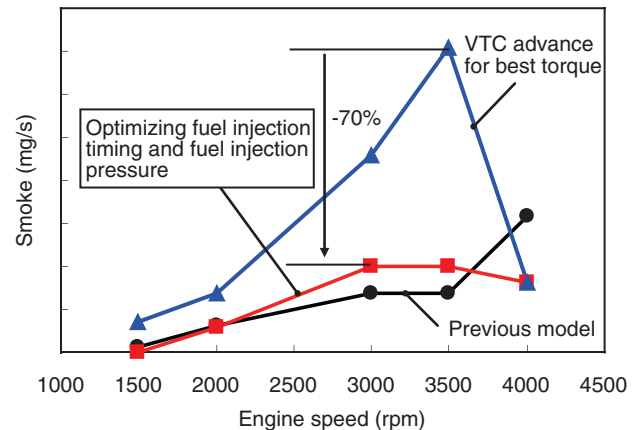


Fig. 16 Comparison of amount of smoke

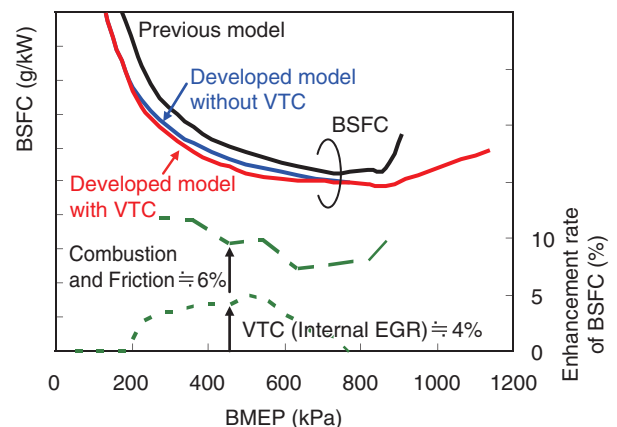


Fig. 17 Comparison of characteristics of BSFC

increased stroke length relative to the previous model yielded a lower S/V ratio, which together with other factors enabled a 14% reduction in heat loss; the strengthened in-cylinder flow together with other factors enabled a 5% reduction in time loss; and the optimized exhaust valve timing enabled a 70% decrease in exhaust loss. The VTC also has an internal EGR effect that yielded enhancement of brake specific fuel consumption relative to the previous model engine.

## 5.2. Friction Reduction Technology

Relative to the previous model engine with short stroke and SOHC, the new model engine involves numerous factors that increase friction. These include the adoption of a long stroke, which increases piston sliding resistance; of DOHC, which increases valve assembly friction; and of hydraulic VTC and the piston oil jet, which increases the lubricating oil requirement and the oil pressure. The adoption of the following technologies, however, enabled a 13% reduction in mechanical friction and a 3.5% enhancement of brake specific fuel consumption relative to the previous model. Figure 19 shows an analysis of

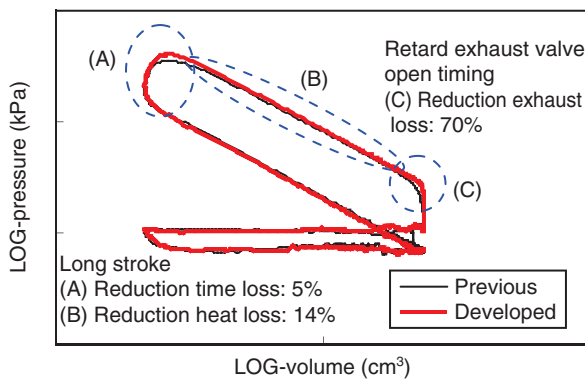


Fig. 18 Comparison of cylinder pressure

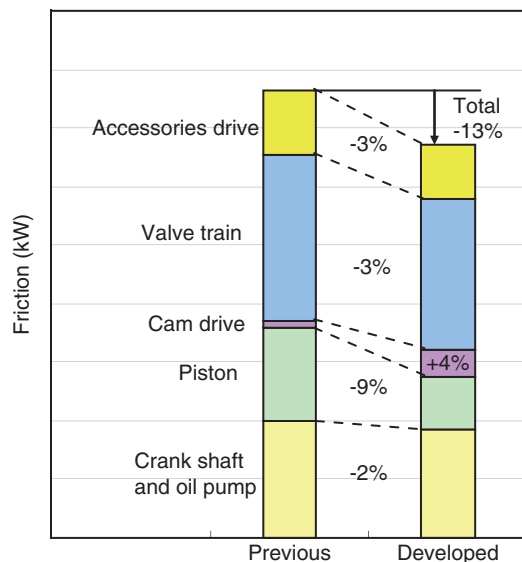


Fig. 19 Mechanical friction of previous and developed model

mechanical friction at 2000 rpm in comparison with the previous model.

### (1) Crank shaft, oil pump drive

- High-efficiency, rotor-type, two-stage relief oil pump
- Crankshaft journal bearings with eccentric grooves on the bottom side and with molybdenum disulfide coating
- Adopt oil seal with low binding force and Teflon coating
- Reduce crank journal surface roughness
- Reduce viscosity of engine oil
- Minimize oil leakage in each part

### (2) Piston sliding resistance

- Piston skirt with alternating-pattern molybdenum disulfide coating
- Optimize piston clearance
- Reduce piston ring tension

### (3) Cam drive

- Reduce timing chain width
- Adopt cam journal holder (thinner cam journal axis)
- Reduce cam journal surface roughness

### (4) Valve train

- Reduce load on valve springs

### (5) Auxiliary drive

- Optimize auxiliary belt layout

The high-efficiency two-stage relief oil pump will be described as an example of mechanical friction reduction technology. In the new model engine, the shape of the rotor teeth in the oil pump was enhanced to increase the rate of volumetric change. This was done to raise the oil discharge rate per rotor turn and reduce the rotor diameter by 2%. Adoption of the two-stage relief structure lowered the oil pump driving force. Figure 20 shows the two-stage relief structure of the oil pump. Figure 21 shows the engine oil pressure and friction with two-stage and one-stage relief oil pumps. The pressure relief has two stages, and the first stage is configured to assure the minimum necessary oil pressure at low speeds. This enables a 36% lowering of the oil pressure, yielding a 9% reduction in friction relative to the one-stage relief oil pump.

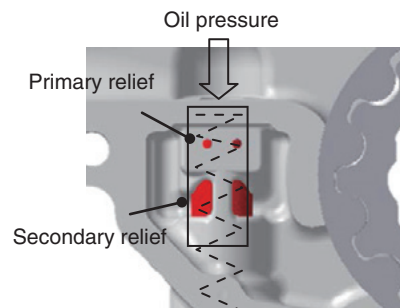


Fig. 20 2-stage relief oil pump

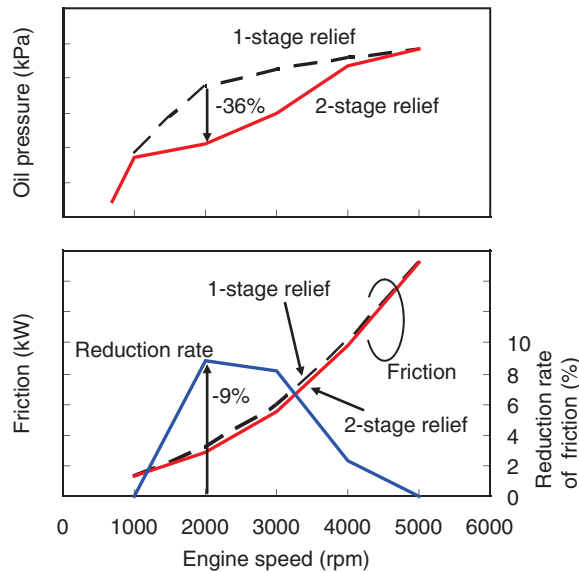


Fig. 21 Effect of 2-stage relief oil pump for oil pressure and friction

## 6. Ignition Timing Control

### 6.1. Calculation of Ignition Timing

As discussed in the preceding, VTC is being used to increase charging efficiency and enhance specific fuel consumption. The combination of VTC with a three-cylinder engine readily yields a scavenging effect and internal EGR effect from the VTC. On the other hand, however, there is little influence from exhaust interference, and internal EGR under the same intake pressure is subject to considerable influence from atmospheric pressure and varies accordingly. Where the previous control system calculated the ignition timing from the intake pressure measured by a pressure sensor and from the engine speed, therefore, it could not track the variations in internal EGR and could not calculate a correct control value. The present engine adopted a control system that calculates the proper ignition timing by using the output from an atmospheric pressure sensor built into the electronic control unit (ECU) to correct for the intake pressure, thereby cancelling out the variations in internal EGR. Therefore, the internal EGR remains always constant with respect to the intake pressure corrected for atmospheric pressure, even if the atmospheric pressure varies, and the fresh air volume is also subject to a similar relationship. Consequently, even if the atmospheric pressure changes when the VTC angle is being advanced, the system will be able to calculate the correct ignition timing. Figure 22 shows how the internal EGR rate with respect to the intake pressure at varying atmospheric pressures and the internal EGR rate with respect to the intake pressure corrected for atmospheric pressure are different.

### 6.2. Optimizing the Retardation of Ignition Timing by means of Knocking Detection

The knocking control system applied from the 2011 model CIVIC uses information on multiple frequencies by employing a non-resonant knocking sensor and short-time Fourier transform (STFT) processing. By comparison with the previous method, which only determined whether or not knocking had occurred by the magnitude of the knocking sensor output, the new method uses frequency and time component pattern matching to enable identification of knocking intensity<sup>(3)</sup>. Therefore, as shown in Fig. 23, the previous method could not identify the intensity of knocking, so retard was always at a constant value. The new method, however, can provide retard amounts appropriate to the intensity. This new control method has also been applied in the present engine, and it has produced enhancement of specific fuel consumption by reducing the retard amount during trace knocking, which occurs frequently in completely built vehicles.

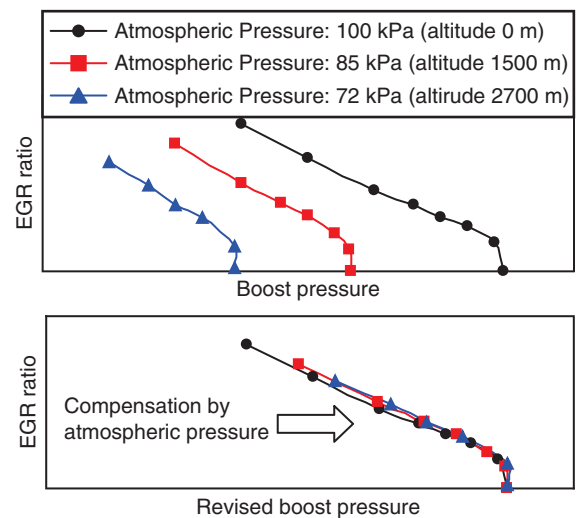


Fig. 22 Compensation by atmospheric pressure of internal EGR rate

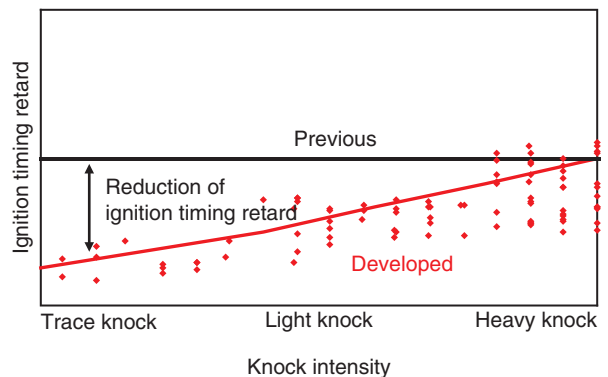


Fig. 23 Ignition timing retard depending on knock intensity

## 7. Conclusion

An engine was developed that achieves a balance of low fuel consumption and high power through the adoption of intake VTC technology, combustion enhancement technology, and friction reduction technology as well as the revision of ignition timing control. The effects of this engine are as follows:

- (1) In comparison with the previous model engine, it achieved a 13% enhancement of maximum power and 8% enhancement of maximum torque in the naturally aspirated engine and obtained a 12% enhancement of maximum torque in the turbocharged engine; and
- (2) Brake specific fuel consumption was reduced 10% in both the naturally aspirated and turbocharged engines.

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