

Development of New 2.0L Lean-burn Engine

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ABSTRACT

A new lean-burn engine was developed as a 4-cylinder power plant for Honda's next generation 2L-class vehicle. New mechanisms, such as i-VTEC which combines variable valve timing and lift control and continuously variable camshaft phase control, are applied to this power plant. As a result, a 113kW maximum output and a 186Nm maximum torque were achieved while at the same time a 14.2km/L fuel economy in the 10-15 mode and a 50% reduction over the Japanese 2000 Emissions standards were realized. Further, improved assembly conditions were facilitated by decreasing the weight of the engine by 10kg and the length of the engine by 54mm.

1. Introduction

Progress in reducing the fuel consumption and emissions of vehicle engines is an important issue in addressing global concerns about the environment. The requirements of vehicle owners are diversifying, and the demand for high performance and improved noise and vibration performance from engines installed in a medium-class chassis is also increasing.

With this in mind, we developed our next generation 2L engine, aiming specifically at a balance between lower negative impact to the environment and achievement of high performance. The high-output, lightweight and compact DOHC-VTEC engine⁽¹⁾ that was developed for the S2000 was joined with the continuously variable camshaft phase system (VTC) to make a low fuel consumption and emission volume compatible with high torque and power output. We present in this paper an outline of the main components, technological features and structure of the engine.

2. Aims of the Development

It was planned that this engine would be fitted in various chassis types as the main new-generation engine. Therefore, we established the following developmental aims:

- (1) Maximization of efficiency to achieve the best level of fuel economy
- (2) An emission level complying with the 50% reduction in emission volume contained in the 2000 Japanese Emissions Regulations (J-LEV), with the possibility of complying with even stricter regulations
- (3) Improved power output in the most frequently used speed range, i.e., medium to low speeds

- (4) Lightweight and compact design making it suitable for use in small cars

We sought to achieve these aims in a balanced system that would be supplemented by conducting further development of the transmission system.

3. Outline of the Main Features and Specifications

Table 1 compares the specifications of the new engine (K20A) with those of the previous model (F20B), while Fig. 1 presents a graph of the new engine's output characteristics, and Fig. 2 shows a cross-sectional view.

Improved output was achieved by using a high

Table 1 Engine specification

Engine type	K20A New type	F20B Previous type
Cylinder configuration	In-line 4-cylinder	In-line 4-cylinder
Bore × Stroke (mm)	86 × 86	85 × 88
Displacement (cm ³)	1998	1997
Compression ratio	9.8:1	9.0:1
Valve train	DOHC i-VTEC	SOHC VTEC
Valve diameter (mm)		
Inlet	35	34
Exhaust	30	29
Fuel supply system	Programed fuel injection	Programed fuel injection
Ignition system	Programed ignition	Programed ignition
Recommended gas	Regular unleaded	Regular unleaded
Maximum output (kW/rpm)	113/6500	110/6200
Maximum torque (N·m/rpm)	186/4000	186/5000
10-15mode fuel consumption (km/L)	14.2	12.4
Exhaust emission	J-LEV	J-LEV

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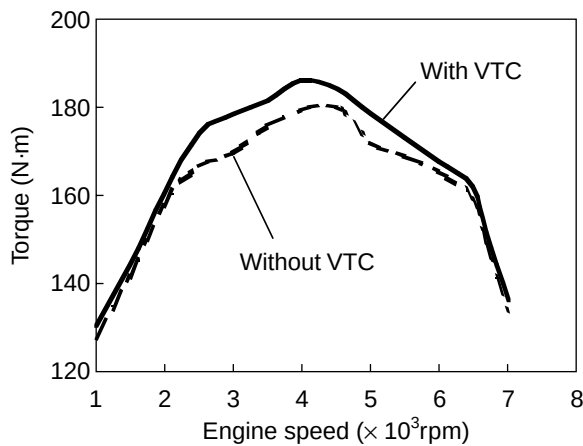


Fig.1 K20A engine torque

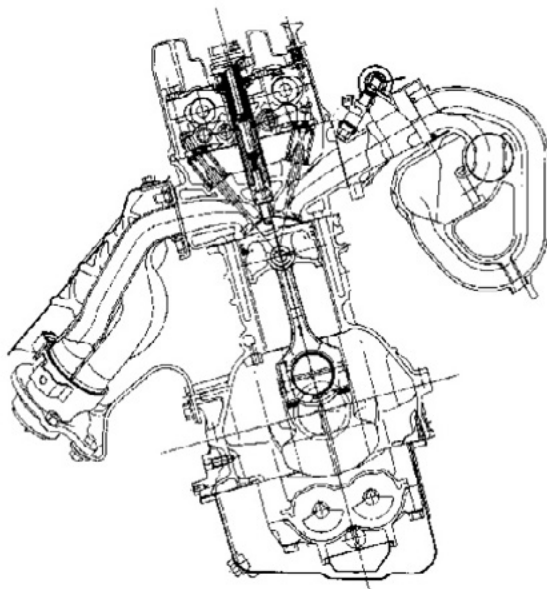


Fig.2 Section view of the K20A engine

compression ratio, larger-diameter intake valves, roller cam followers on the rocker arms and a special surface treatment of the piston skirts. The new i-VTEC valve system combines the proven VTEC system (Variable Valve Timing and Valve Lift Electronic Control) with the continuously variable camshaft phase control system (VTC, Variable Valve Timing Control). VTC was adopted by Honda for the first time, to provide improved performance across the entire driving range. The former VTEC system adjusted the camshaft operating angle and valve lift through two or three stages at low and high engine speeds to provide both low- and medium-speed torque and high power output. By making the valve timing continuously variable in response to the

load with VTC, the valve lift is always at the optimum setting, enabling us to achieve an improvement in volumetric efficiency.

We adopted a silent chain to drive the valve train, enabling the total engine length to be reduced and maintenance-free operation to be achieved. We also utilized a silent chain drive for the balancer shafts which reduce the secondary vibration characteristics of a four cylinder engine and suppress booming noise, and positioned these shafts inside the oil pan. We used a long tubular resin intake manifold to improve the medium- and low-speed torque, and made it compatible with high-speed power output by switching between different lengths with a rotary valve.

We adopted a cross-flow head with front intake and rear exhaust manifolds, making it possible to continue using a dual exhaust system, which reduces exhaust interference, and to place the catalyst close to the engine. In this way, the catalyst is activated almost as soon as the engine is started, enabling improvements in both emissions and output to be attained.

The adoption of this new configuration allowed a reduction in the total engine length of 54mm and in engine weight of 10kg in comparison with the F20B.

4. i-VTEC

Fig. 3 shows the configuration of the i-VTEC system, and Fig. 4 shows the operation map. To achieve a balance between high power output, good fuel economy and low emissions in the K20A engine, we adopted i-VTEC, which controls the valve timing to provide the optimum amount of overlap and operating angle of the cams in response to driving conditions. This system brings together VTEC, in which the rocker arm linked by the hydraulic switching device varies the cam profile, and VTC, in which hydraulic actuation allows continuous variation of the camshaft phase.

Switching between VTEC's low- and high-speed camshaft operation is set at an engine speed of about 2000 rpm. When the low-speed camshaft setting is in operation, one intake valve is almost completely inactive, increasing the airflow speed and leading to an improvement in volumetric efficiency. The high-speed camshaft setting produces higher power output through high valve lift and a wide operating angle. Furthermore, VTC varies the angle for closing the intake valve in the high-load area to achieve the highest volumetric efficiency from the respective camshaft settings. In the low-load area, it controls the amount of valve-opening overlap, to give priority to fuel economy and emission reduction. VTC further reduces the amount of valve overlap to minimize rpm fluctuation in the idling range.

Fig. 5 shows an outline of the VTC configuration. The signal from the ECU governs the oil control valve and provides oil pressure to the actuator. In response to control instructions, the oil pressure acts on the oil chamber for advance or retarding, the vanes change their position to advance or retard, and the camshaft phase is adjusted. As already explained, the optimum control across the entire driving range offered by the combination of VTEC and VTC provides an engine with high power output and good fuel economy.

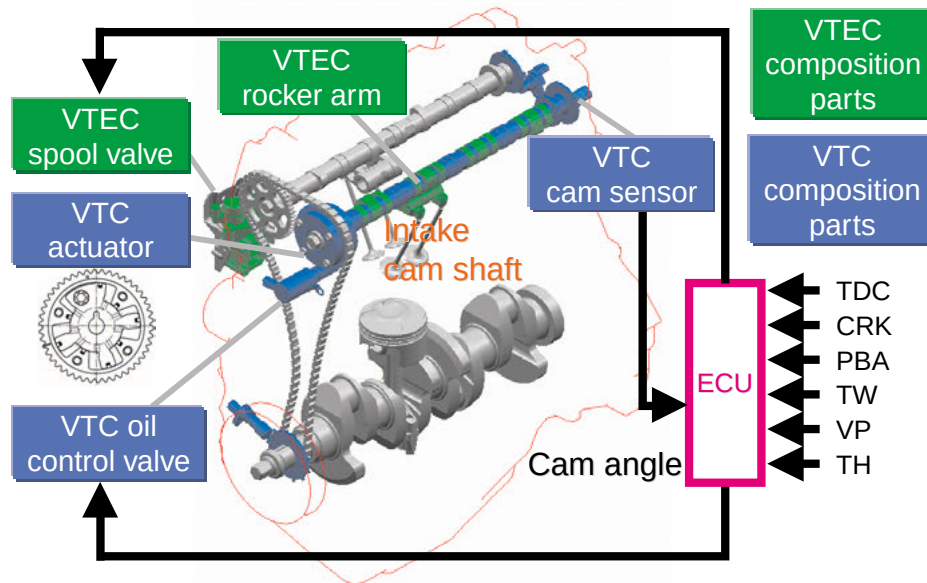


Fig.3 Construction of i-VTEC systems

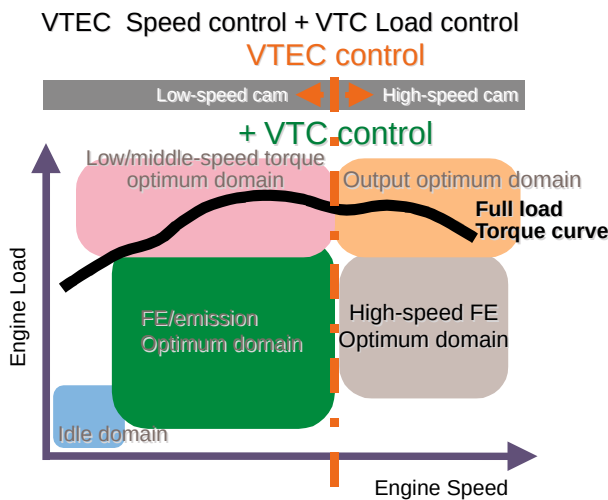


Fig.4 Outline of i-VTEC operation map

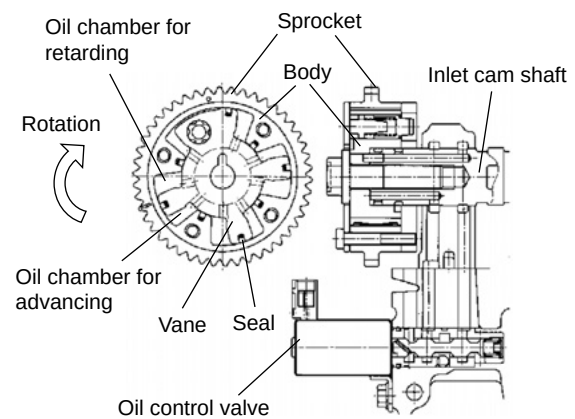


Fig.5 VTC unit and control system

5. Fuel Economy

We achieved a substantial improvement in fuel economy with the K20A by adopting the lean-burn principle, and conducting a thorough review to reduce engine friction. Next, we will discuss the technology applied for this achievement.

5.1. Improved Combustion

5.1.1. Cylinder Head

To improve the lean limit of lean-burn, we adopted a shape for the intake port which induces strong swirl flow. The VTEC system leaves one valve mostly inactive (1mm lift) in the low-speed range. The combination of this minute lift of one valve and the port shape induces strong swirl flow in the combustion chamber. This strong swirl flow facilitates combustion and improves fuel economy. Intake resistance is generally high with a port which induces swirl flow, which limits improvements to the power output. However, our

careful design of the port shape has brought these factors into balance.

Fig. 6 shows the shape of the head port, while Fig. 7 demonstrates the relationship between the mean flow coefficient and the swirl ratio. It can be seen that the intake port of the K20A engine has a higher power output potential than that of the previous design. Looking at the balance between the lean limit and mean flow coefficient with the two-valve system, the optimum point is established with a swirl ratio of 3.4 and a mean flow coefficient of 0.288.

By pre-assembling the supports for the two camshafts, we could make the engine as compact as the SOHC unit, and greatly reduce the amount of machining required.

Furthermore, in addition to improving the bowl-in shape of the piston crown and the squish shape of the combustion chamber, we adopted wide-gap spark plugs with thin iridium center electrodes, thus achieving rapid combustion while improving the lean limit.

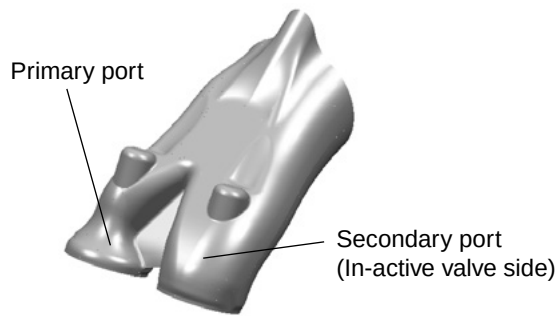


Fig.6 Intake port configuration of K20A

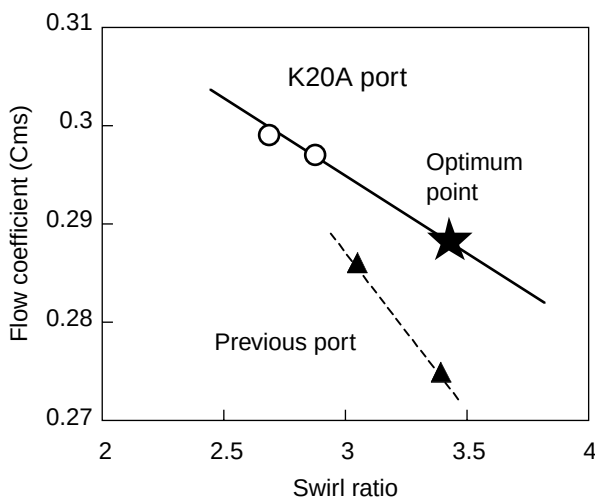


Fig.7 Relationship of swirl ratio and flow coefficient

5.1.2. Optimization of the Valve Timing

In addition to the established benefit of fast combustion with one valve mostly inactive, i-VTEC improves the fuel economy by reducing the pumping loss and raising the lean limit by controlling the amount of valve overlap and turbulence in the cylinder via variable valve timing. We established the i-VTEC system for the K20A engine to maximize these benefits to the fuel economy.

The i-VTEC principle under lean-burn conditions was established by taking into account the fast combustion that can be achieved by making one valve mostly inactive, while enhanced combustion results from the changes in pumping loss and turbulence in the cylinders due to the VTC system. Fig. 8 shows the co-variance of IMEP with the intake valve timing change during lean-mixture combustion. The co-variance of IMEP is an index of the stability of combustion. The co-variance of IMEP is at its lowest when the intake valve open timing is in the region of 20 - 35 degrees after TDC, and hence combustion is most stable in this region. This is the result of the delayed opening of the intake valve after the piston has begun its downward stroke causing the mixture to flow at a relatively high speed therefore increasing turbulence in the cylinder, and of the improved combustion achieved by reducing the residual gas through a smaller valve overlap.

Fig. 9 shows the variation of the indicated specific fuel consumption (ISFC) and co-variance of IMEP with the air-fuel ratio. The effects of the intake port swirl, improved lean limit due to changes in the shape of the combustion chamber and the crown shape of the pistons, and the use of a high compression ratio with optimized variable valve timing enabled an air-fuel ratio of 23 to be set. Therefore the ISFC was improved by 10% over the F20B engine. Further, we solved the issue of NO_x emissions in lean-burn engines during lean-mixture combustion by adopting a lean NO_x catalyst.

There are certain driving conditions under which lean-burn is not applied, and the mixture is regulated to a stoichiometric air-fuel ratio. As a means to improve the fuel economy with a stoichiometric air-fuel ratio, EGR, which recirculates exhaust gases, is effective. The i-VTEC system

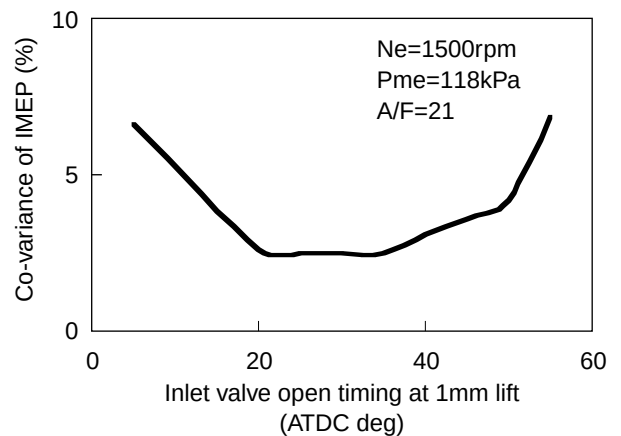


Fig.8 Influence of inlet valve open timing for lean mixture combustion stability

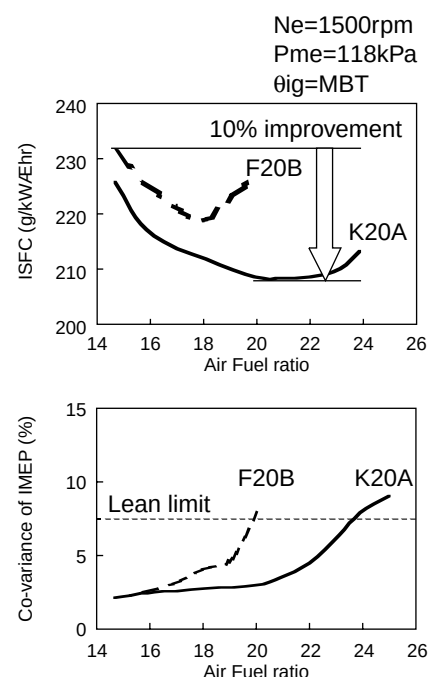


Fig.9 ISFC improvement and co-variance of IMEP

employs internal EGR to reintroduce exhaust gases into the intake manifold by varying the valve overlap.

Fig. 10 shows the relationship between the timing for intake valve opening and the brake specific fuel consumption (BSFC). Fig. 11 compares the fuel economy improvement of this system with that of the external EGR method utilized in the previous engine. Exhaust gas is recirculated into the intake runner when the valve timing is advanced and the amount of overlap is increased; therefore, internal EGR is increased and the fuel economy is improved by reductions in the pumping loss and thermal loss and by the improvement in ratio of specific heat of the working medium. However, if the volume of EGR is too high, combustion becomes unstable and misfires occur. Since i-VTEC combines a high swirl port with one valve being almost completely inactive, combustion is stable even when the EGR volume is increased, enabling fuel economy to be improved by 5%. Furthermore, when we compare internal and external EGR for the same EGR volume, we can see that the rate of fuel economy improvement with internal EGR is higher. This is

because the temperature of the recirculated gas is high, and the intake valve closing angle approaches bottom dead center with internal EGR, thus increasing the effective compression ratio. VTC establishes valve timing which minimizes the fuel consumption at all engine speeds and loads.

By setting the degree of overlap to the minimum in the idling range, there is no loss of market appeal due to rough combustion. Further, since it was possible to reduce the idling speed by 50 rpm, which has a significant effect on fuel economy, the idling mode fuel economy was improved by 2%.

5.2. Friction Reduction Techniques

We made improvements in many areas relating to the reduction of mechanical losses. Table 2 presents an outline of the techniques that were applied to friction reduction. Together with decreasing sliding friction by improving the oil film retention on the rubbing surfaces, we reexamined the valve spring load. In Fig. 12, the friction under partial load is compared with that of mass production models with a reduction of more than 7% being achieved. We explain next how this was achieved.

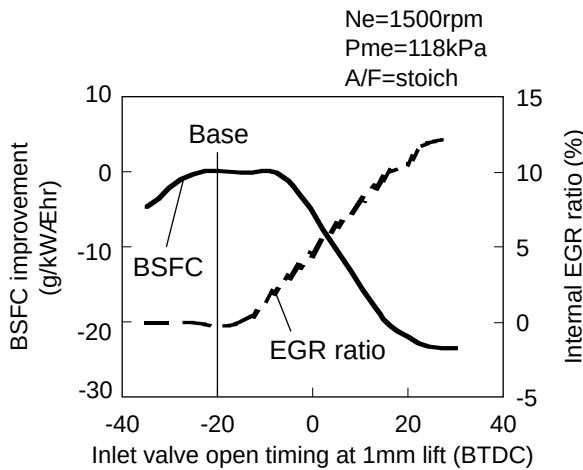


Fig.10 Influence of inlet valve open timing for BSFC improvement

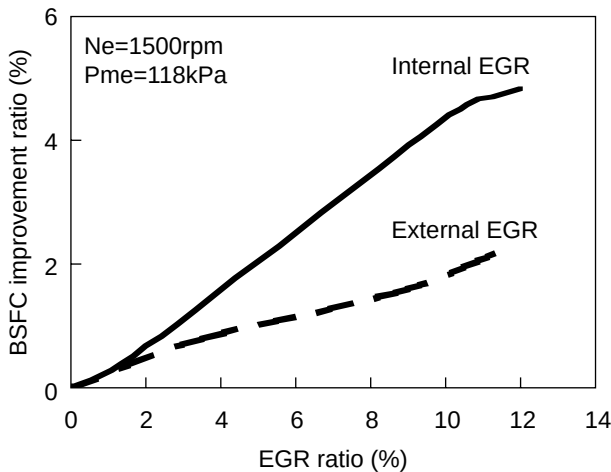


Fig.11 Comparison of BSFC improvement with internal and external EGR

Table 2 Friction reduction technology

Parts	Applied technologies friction reduction
Rocker arms	Roller follower
Valve springs	Spring full lift load reduction
Camshaft	Camshaft bearing clearance optimized
Piston	Skirt contact surface product reduction Skirt surface micro dimple machining Skirt surface molybdenum coating
Piston rings	Tension reduction
Oil pump	Rotor miniaturization Relief setting oil pressure down
Crank journal bearing	Surface single-cut turning machining
Balancer journal bearing	Surface single-cut turning machining
Cam chain tensioner	Tensioner spring load optimized
Engine oil	Low viscosity oil (0W-20)

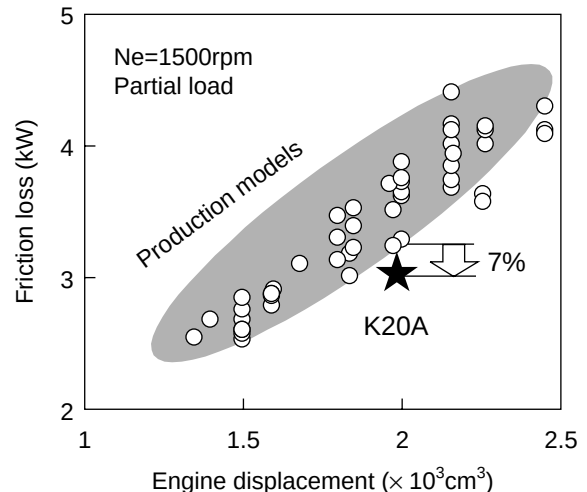


Fig.12 Friction loss and displacement

5.2.1. Valve Train

To reduce friction, we used roller followers throughout the entire valve train. For the intake VTEC rocker arm we adopted a coaxial configuration of three rockers and a roller follower, and for the exhaust we used an aluminum die cast structure featuring two valves in one unit, in order to reduce both weight and costs.

The intake rocker arm is manufactured by lost wax precision casting, making possible a lightweight structure with a high capability to follow high engine speeds. Fig. 13 shows the structure of the rocker arms.

5.2.2. Pistons

To improve fuel economy, we greatly reduced the weight of the pistons and applied friction reduction techniques. As shown in Fig. 14, we shortened the piston skirt and compression height and reduced the width of the skirt, leading to a reduction in weight of 10% compared to the previous engine. To avoid any loss of durability caused by this weight reduction, we evaluated the stress at each point and the pressure distribution on the surface of the skirt by a CAE analysis, and used the results to optimize the structural design. We increased the oil film retention on the skirt surface by a Micro-dimple treatment⁽²⁾ and molybdenum coating. The side force pressure is reduced by the lighter pistons and oil film shearing is reduced by shortening the width of the skirt. Therefore we achieved a friction reduction of about 25% for the pistons.

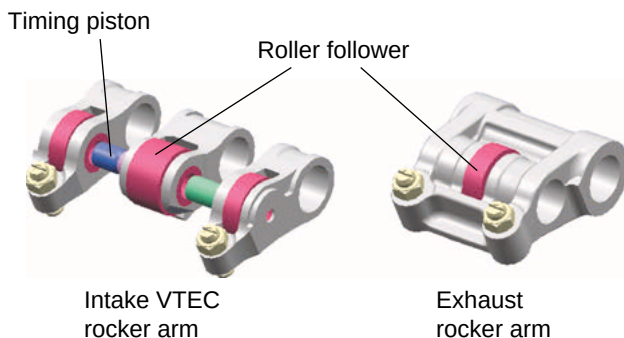


Fig.13 Rocker arm



Fig.14 Piston configuration

6. Power and Torque Output

6.1. Intake System

The lightweight intake system utilizes an intake manifold made of resin. Furthermore, in order to balance the improvement in medium-to-low-speed torque with a high output, we applied a variable-length manifold controlled by a switching valve. We used a rotary type switching valve rather than the previously applied butterfly valve as it has lower flow resistance. Fig. 15 shows a cross-section of the intake manifold. We used resin for the intake manifold rather than the former aluminum alloy to suppress heat transfer from the engine and to reduce the temperature of the air intake, which increased the volumetric efficiency and raised the output. A computer simulation was used to vary the conditions and to establish the basic specifications for the intake manifold length. Fig. 16 shows the effect of varying

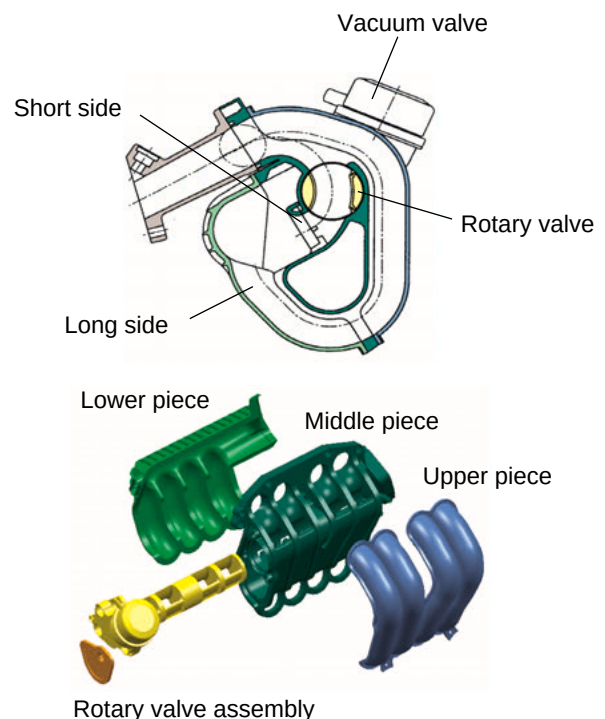


Fig.15 Dual mode resin intake manifold

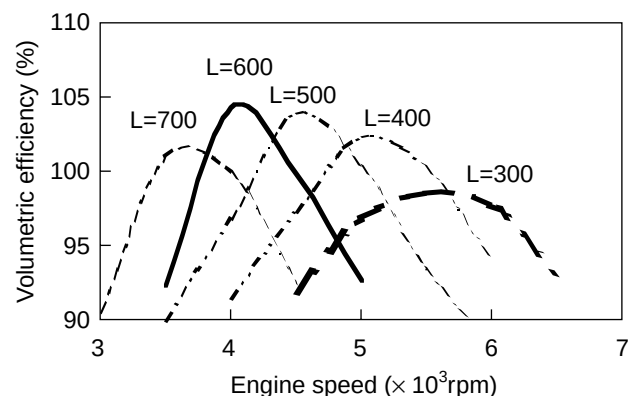


Fig.16 Influence of intake manifold on volumetric efficiency

the pipe length on the volumetric efficiency. The volumetric efficiency of the former model's intake runner length of about 300mm is adequate at high revolutions, but in the medium-to-low-speed range it drops and a so-called torque dip occurs. To avoid this drop, we aimed for a high volumetric efficiency increase throughout the driving range by switching between two tube lengths. We established tube lengths of 540mm and 270mm to achieve the required volumetric efficiency and installation space. VTC establishes the valve timing which will optimize the torque in response to engine speed, and we achieved with this system an improvement of about 13% in the medium-to-low-speed torque compared to the F20B. Fig. 17 demonstrates the torque improvement resulting from the i-VTEC system.

6.2. Exhaust System

A dual system for the exhaust manifold and exhaust pipe is used to reduce exhaust interference and improve engine output. In addition to a low-heat-capacity stainless steel pipe applied to ensure quick heating of the catalyst, the split cross-section produced at the center of the pipe by a partition plate in the confluence area has reduced the heat-radiating area and balanced the output with low emissions. We furthermore achieved a weight reduction of 1.3kg compared to the FC 4-1 exhaust manifold used with the F20B engine.

Fig. 18 shows the construction of the exhaust system, and Fig. 19 demonstrates the improved engine output with the dual inner pipe type exhaust pipe.

6.3. Emissions Performance

We achieved low emissions with the K20A engine by combining a rear exhaust, low-heat-mass exhaust system and three-way catalyst, and a lean NOx catalyst. Fig. 20 shows the temperature rising characteristics of the three-way catalyst directly after starting the engine.

Compared with the F20B, the time taken to reach catalyst activation temperature is reduced, thus reducing emissions at engine start-up.

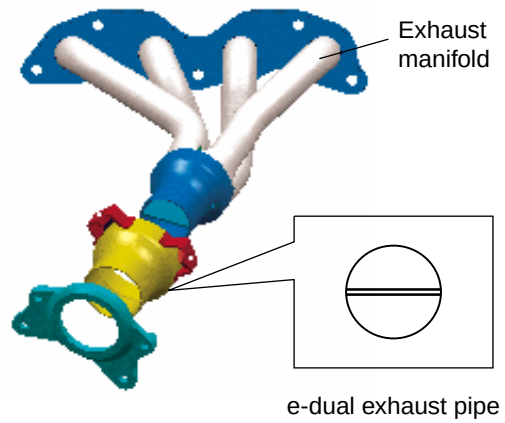


Fig.18 Configuration of e-dual exhaust system

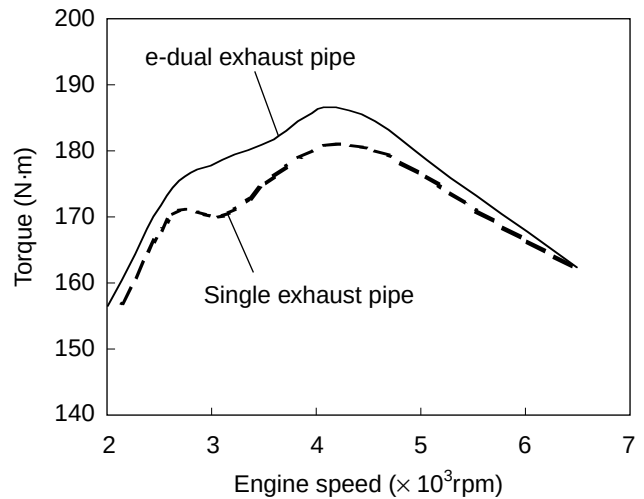


Fig.19 Effect of e-dual exhaust

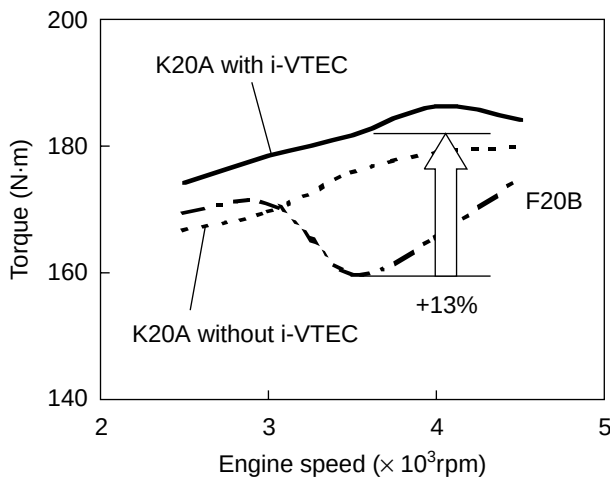


Fig.17 i-VTEC torque improvement effect

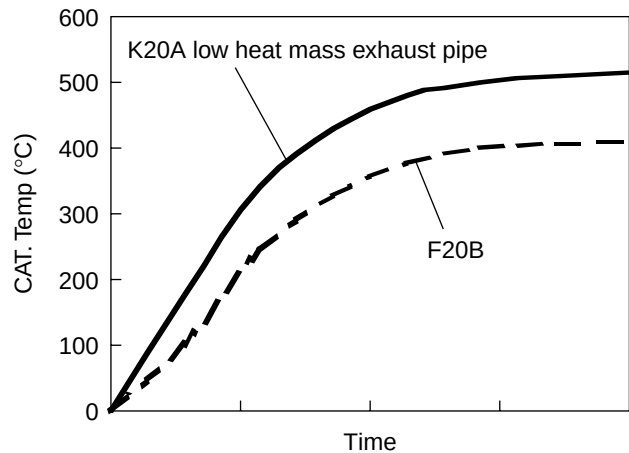


Fig.20 Comparison of catalyst temperature after engine start

7. Engine Construction

The K20A has an improved engine construction with better output performance and fuel economy, as well as improved noise and vibration (NV) properties, decreased weight and size. We discuss next the engine construction used to realize these benefits.

7.1 Improvement in Noise and Vibration (NV)

7.1.1. Cylinder Block Structure

The lower block structure has integrated FC bearing caps in order to reduce the crankshaft hitting noise and the power plant resonance which exacerbate engine noise during acceleration and at high engine speed. Crankshaft hitting noise was reduced by suppressing transmitted vibration by increasing the rigidity of the bearing journals. As for the problem of power plant resonance, we improved the connection rigidity between the engine and transmission, which is weakest point in its flexural rigidity, by setting the connection flange to the transmission on a lower block that has higher rigidity than the previous deep skirt. Fig. 21 shows the cylinder block construction utilizing this lower block. Fig. 22 compares the effectiveness of the improved flexural rigidity in the K20A lower block structure with the former deep skirt and separate bearing caps, as expressed by the vibration response data at the front and rear of the transmission.

7.1.2. Low-mounted Balancers and Oil Pump Structure

To eliminate the second order vibration that is characteristic of an in-line four-cylinder engine and to reduce booming noise in the chassis, we utilized secondary balancers. The trochoid-shaped oil pump is mounted coaxially with the balancer shafts to concentrate their functions and simplify the construction. Fig. 23 shows the configuration of the balancer mechanism with the built-in oil pump. The rear balancer shaft is driven by a chain from the crankshaft, and the front shaft is driven in the opposite direction by helical gears. The split construction of the block and balancer assembly allowed us to optimize the external

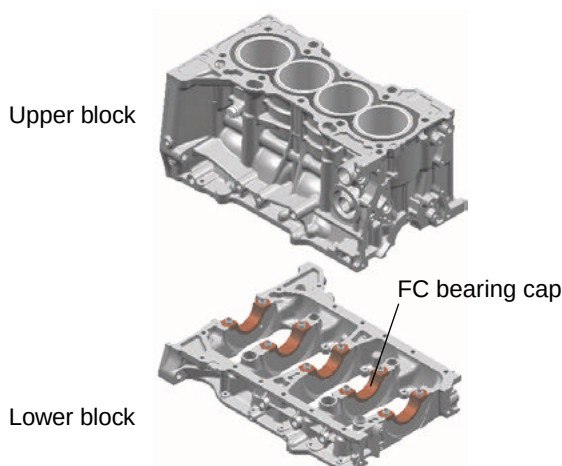


Fig.21 Cylinder block

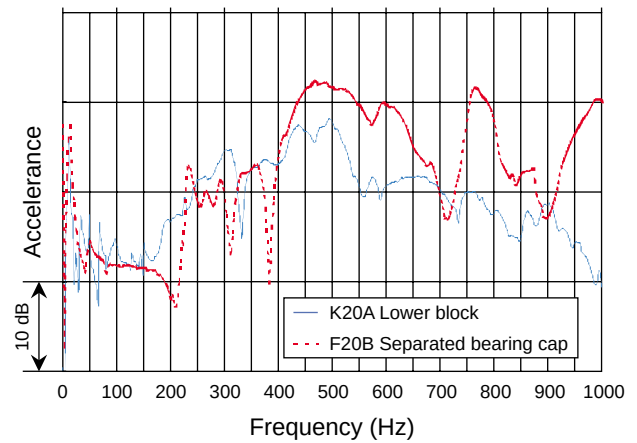


Fig.22 FRF of power plant (Lateral bending direction)

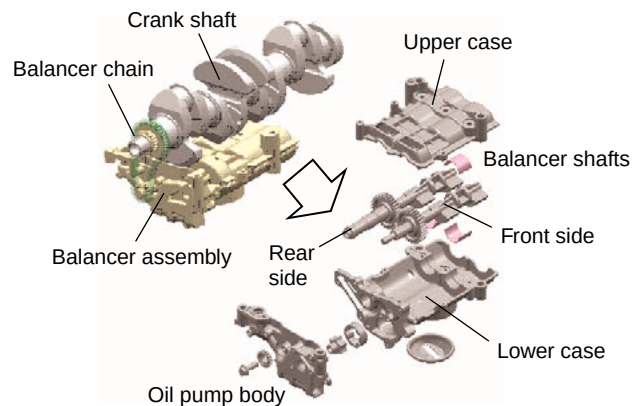


Fig.23 Balancer system

diameter of the balance weights and journals, and to achieve a compact construction with the two journals per balancer shaft. The oil pump rotor is driven by the front balancer shaft at twice the engine speed. Because of this, we were able to halve the external diameter of the rotor and achieve higher efficiency in comparison with the former directly connected crankshaft construction. We were further able to reduce the total length of the engine by 25mm by using a chain drive and locating the balancers below the engine block.

7.2. Lightweight, Compact Design

7.2.1. Variable-Length Resin Intake Manifold

Resin is used instead of the former aluminum alloy for the intake manifold, including the switching valve, to achieve a weight reduction of 6.0kg.

7.2.2. Camshaft Drive System

A silent chain is used to drive the camshafts and balancers, and this has enabled the engine length to be reduced by 13mm in comparison with the former engine employing a timing belt drive. Improvements in the system's durability and reliability have eliminated the necessity for periodic maintenance. Fig. 24 shows the layout of the camshaft drive chain. By adopting an FCD hollow

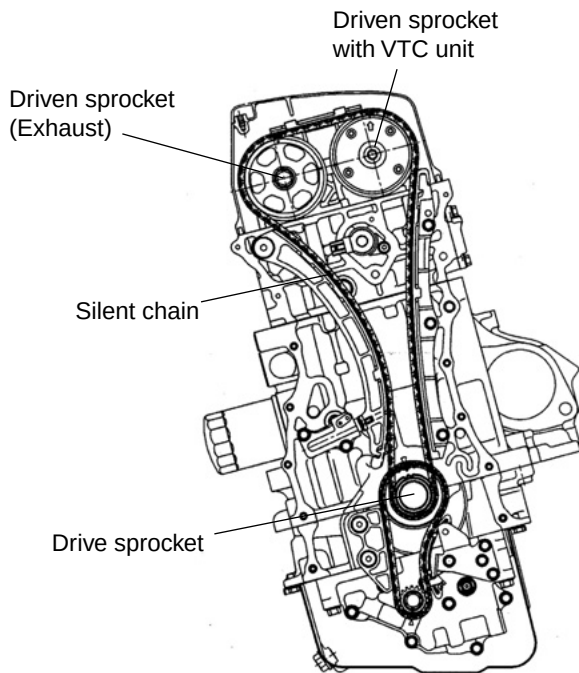


Fig.24 Camshaft chain layout

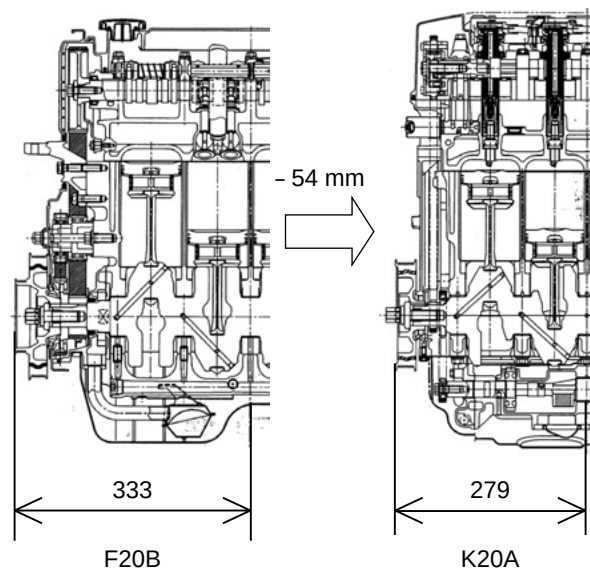


Fig.25 Comparison of engine length

construction for the camshaft, we achieved a reduction in weight of 0.7kg compared to the former camshafts used with the DOHC engine. We further simplified the construction of the cylinder head by utilizing the hollow camshafts as channels to supply oil to the camshaft journals. We were able to reduce manufacturing costs by utilizing a production process known as Quick Cast with the FCD hollow camshafts for the first time. In this process, sand molds are not required. Rather, the material is directly cast into a die.

7.2.3. Drive System for the Accessories

A serpentine belt drive is used to drive the accessories.

By driving all the accessories with a single multi-groove belt, we achieved a reduction in engine length of 16mm compared to the former two-belt system. As Fig. 25 shows, the change to a camshaft chain drive, lowering the balancers and oil pump and the adoption of a serpentine accessories belt has allowed a total reduction in engine length of 54mm compared to the F20B engine.

A torsion spring auto-tensioner is used to make the accessories belt maintenance-free. Furthermore, the change to a camshaft chain drive allowed us to drive the water pump with the accessories belt, rather than with the former timing belt drive.

Table 3 K20A main technology

Subject	Main technology
<u>Good fuel economy</u>	• i-VTEC • High swirl intake port • Friction reduction technology (see Table 2)
<u>Low emission</u>	• Back exhaust layout • Lean NOx catalyst
<u>High performance</u>	• High compression ratio • e-dual pipe exhaust manifold • Dual mode resin intake manifold
<u>Light weight and compact</u>	• Formation of serpentine accessories belt • Camshaft and balancer chain drive
<u>Silent (NV improvement)</u>	• High rigidity cylinder block • Balancer in oil pan

8. Conclusion

Table 3 summarizes the aims and main techniques used in the development of the K20A engine.

It was possible to realize higher standards for the next-generation 2L engine by improving the output, fuel economy, and emissions utilizing i-VTEC as the key feature. At the same time, the lightweight and compact design and quiet operation make the K20A suitable for installation in various types of chassis.

9. Outlook

The K20A engine has a construction that allows for derivatives such as increased displacement and sport versions as responses to environmental changes and the diversifying needs of users. We are proud of our achievement of this system, however, we plan to put forth great efforts toward enhancing this technology in response to changing demands.

We sincerely thank all those people, both inside and outside Honda R&D, who supported our efforts.

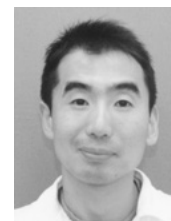
References

- (1) Y. Akimoto, et al.: Development of a High-power and low-emission Engine for the HONDA S2000, Honda R&D Technical Review, Vol. 11 No. 1, p. 29-38 (1999) (Written in Japanese)
- (2) H. Ogihara: "Surface profile modified for engine friction reduction" (Lecture in Japanese), Society of Fine Particle Bonberding Technology of Japan, (1999)

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