

Technology for Realization of Gasoline Engine Brake Thermal Efficiency of 45%

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ABSTRACT

Seeking to achieve a brake thermal efficiency of 45% in gasoline engines, the development project discussed in this paper conducted research to increase the compression ratio in order to boost theoretical thermal efficiency and to realize an EGR technology making it possible to control knocking and reduce cooling loss.

At the same time, the basic framework of the engine was optimized in order to help realize these combustion targets. Based on the use of a long stroke to realize a stroke/bore ratio of 1.5, which is higher than that of mass-production engines, thermal efficiency was increased by reducing time loss and cooling loss through the selection of an optimal combustion chamber shape. With this engine framework as the basis, a variety of measures were applied, including the selection of an intake port shape that would enhance in-cylinder flow, the adjustment of the effective compression ratio by means of intake valve timing, control of in-cylinder fuel distribution, and high-energy sparking. These measures made it possible to realize Minimum advance for Best Torque operation at a compression ratio of 17 and stable operation at an external EGR rate of 35%, helping to enable the achievement of a maximum brake thermal efficiency of 45% in an actual single-cylinder engine.

1. Introduction

Today, the realization of increased automotive fuel efficiency and the reduction of exhaust emissions are demanded in order to reduce CO₂ emissions that contribute to global warming and protect the earth's environment, and as a result regulation values are becoming increasingly stringent year by year. Automotive electrification, including the use of electric vehicles (EV), is increasing in response to these demands; at present, the use of conventional vehicles operating exclusively by means of internal combustion engines, or hybrid vehicles which combine electric motors with internal combustion engines, can be considered the main trends in this area. Increasing the thermal efficiency of internal combustion engines will therefore be an important technological issue into the future. The maximum thermal efficiency of current gasoline internal combustion engines is 39%, but extrapolating from changes in fuel efficiency regulation values, a level of brake thermal efficiency of 45% will be necessary in several years. The realization of an increase

in the level of brake thermal efficiency to 45% or higher can therefore be considered a significant issue to be addressed in gasoline internal combustion engines in the near future. The development discussed in this paper elucidated the technological elements necessary for the realization of a 45% level of brake thermal efficiency in gasoline engines, and verified its findings in an actual vehicle. Lean-burn operation⁽¹⁾ can be considered as a means of realizing a maximum brake thermal efficiency of 45%, but the project under discussion assumed stoichiometric operation from considerations of the level of complexity of exhaust gas treatment technologies. This paper provides an overview of the research.

2. Development Goals

2.1. Setting of Development Targets

The following targets were set at the commencement of the development process:

- (1) Realization of a maximum brake thermal efficiency of 45%; and

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- (2) Realization of the above target via stoichiometric operation using 91-octane regular gasoline.

2.2. Development Orientation

The necessary elements for the realization of increased thermal efficiency were identified, and the details of the main measures to be applied were determined. Figure 1 shows factors affecting the realization of increased thermal efficiency. The following orientations for the development were set based on a factor analysis:

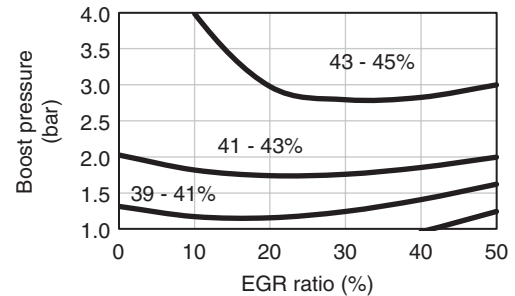
- (1) High compression ratio for increased stoichiometric thermal efficiency;
- (2) High-EGR operation in order to reduce cooling loss and control knocking and realize Minimum advance for Best Torque (MBT) operation at a high compression ratio; and
- (3) Realization of increased thermal efficiency by means of optimization of basic engine framework.

2.3. Setting of Conditions for Achievement of Development Targets

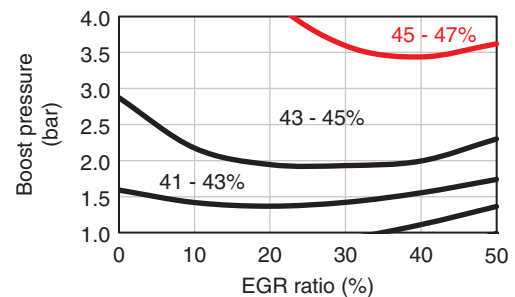
First, the tendencies of cooling loss, time loss, and change in friction in relation to engine load and EGR rate were determined from mass-production engine data. Brake thermal efficiency under MBT conditions was predicted by extrapolation from these results with boost pressure, EGR rate, and compression ratio as parameters. The operating condition for these calculations was an engine speed of 2000 rpm. The results of the calculations are shown in Fig. 2. These results indicate that MBT operation at a compression ratio of 15 or higher and a high boost pressure of 2.5 bar or higher is necessary for the realization of a brake thermal efficiency of 45%. Because the increased demand on the supercharger as the boost pressure rises also increases the level of technological challenge involved, it was decided to realize the target level of thermal efficiency at the minimum boost pressure, as shown by the circle in Fig. 2 (c). Based on these calculation results, the following development guidelines were set in order to realize a 45% level of brake

thermal efficiency:

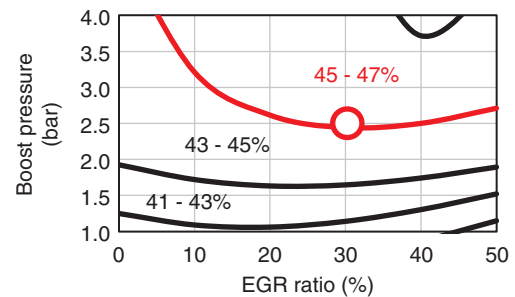
- (1) MBT operation at a compression ratio of 17.0 and a boost pressure of 2.5 bar or higher; and
- (2) Stable operation at an external EGR rate of 30% or higher.



(a) Compression ratio: 13.0



(b) Compression ratio: 15.0



(c) Compression ratio: 17.0

Fig. 2 Calculation of maximum brake thermal efficiency

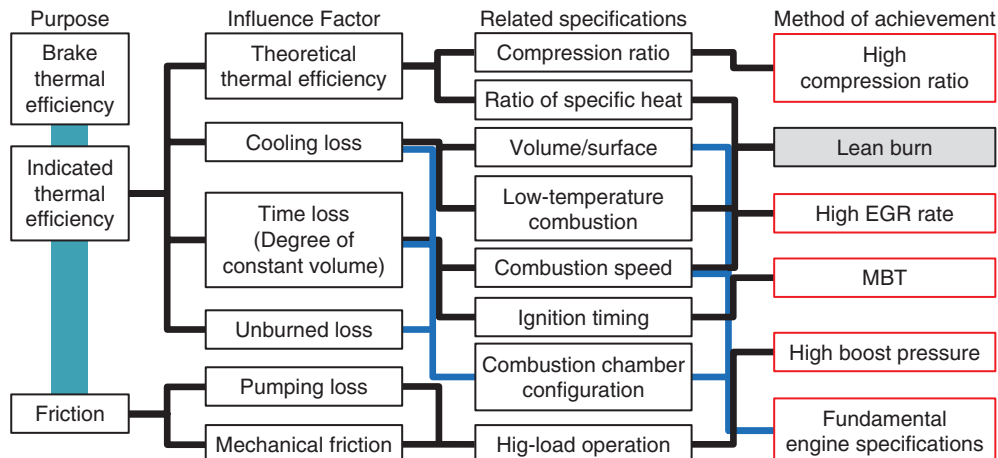


Fig. 1 Analysis of factors influencing brake thermal efficiency

3. Realization of Increased Compression Ratio

The reduction of combustion speed due to the flattening of the combustion chamber shape and the retardation of ignition timing due to knocking are the main factors that represent a challenge in increasing thermal efficiency through the use of a high compression ratio. In addition to the high EGR rate that will be discussed in Chapter 4, the present research responded to these factors by means of the following methods:

- (1) Limitation of flattening of combustion chamber shape through use of a long stroke; and
- (2) Adjustment of the effective compression ratio through delayed closure of the intake valve.

3.1. Use of Long Stroke

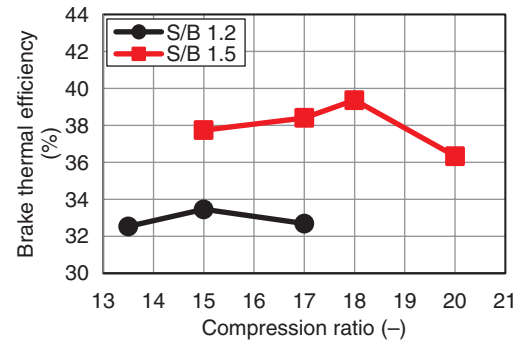
3.1.1. Effect in controlling reduction of combustion speed at high compression ratio

The use of a high compression ratio is an effective method of increasing theoretical thermal efficiency, but presents the tradeoff of reducing combustion speed due to the flattening of the combustion chamber shape. This issue is thought to result from the fact that the flattening of the combustion chamber increases the area of contact between the flame and the walls of the chamber, impeding flame propagation. The positioning of a cavity close to the spark plugs to help prevent any interference with initial flame formation has been proposed as a method of responding to this issue⁽²⁾. However, while this method is effective at the initial stage of combustion, it does not offer sufficient response to the reduction of combustion speed in the second half of the combustion process. In addition, the method necessarily involves the engineering of large concavities and convexities in the piston heads, creating a shape that is disadvantageous from the perspective of cooling loss. It was therefore determined that the use of a long stroke would be employed as a response in the research under discussion. The use of a long stroke makes it possible to increase the height of the combustion chamber without changing the compression ratio, making it possible to control the flattening of the combustion chamber and form piston head shapes with minimal concavities and convexities. In order to verify the effectiveness of the use of a long stroke at a high compression ratio, single-cylinder engines with stroke/bore ratios (S/B) of 1.2, corresponding to mass-production engines, and 1.5 were manufactured, and changes in thermal efficiency against the compression ratio were compared. Table 1 shows the test engine specifications, and Fig. 3 shows results for thermal efficiency at an engine speed of 2000 rpm, an Indicated Mean Effective Pressure (IMEP) of 520 kPa, and an EGR rate of 20% in MBT operation. Combustion speed was calculated using 3D models for each compression ratio and S/B, and the shape that

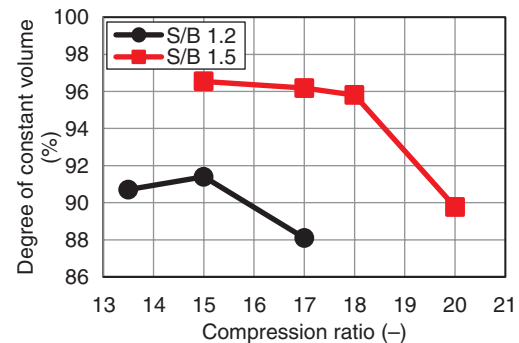
Table 1 Test engine specifications

Engine type	DOHC 4-valve single cylinder	
Bore (mm)	ϕ 81	
Stroke (mm)	96.9	121.5
S/B	1.2	1.5
Compression ratio	13.5, 15.0, 17.0, 18.0, 20.0	
Displacement (cm ³)	499	626
Intake port	Filling port	
Air supply	Supercharged	
EGR system	Low-pressure loop-cooled EGR	
Fuel supply	Port injection	

produced the highest combustion speed was employed for the piston heads. Figures for brake thermal efficiency in the single-cylinder engines were calculated using mechanical friction corresponding to friction in an L4 engine and the calculated value of pumping loss for the supercharger. At an S/B of 1.2, thermal efficiency increases until a compression ratio of 15, but declines at a compression ratio of 17. By contrast, at an S/B of 1.5, thermal efficiency increases until a compression ratio of 18. The main factor producing the decline in thermal efficiency at high compression ratios is the decline in the



(a) Brake thermal efficiency



(b) Degree of constant volume

Fig. 3 Comparison of influence of compression ratio on brake thermal efficiency and degree of constant volume for different S/B

degree of constant volume. Figure 4 shows measurement results for the Mass Burn Fraction (MBF) at compression ratios from 15 to 20 for an S/B of 1.5, an engine speed of 2000 rpm, an IMEP of 520 kPa, MBT, and an EGR rate of 20%. At a compression ratio of 20, there is a sudden decline in the speed of combustion. The effect is particularly marked in the second half of the combustion process. This is thought to be due to the fact that there is a local increase in the Surface/Volume ratio (S/V) close to the ends of the bore, as shown in the diagram in Fig. 5, and this impedes the propagation of the flame. The use of a higher S/B makes it possible to increase the height of the combustion chamber without changing the compression ratio, helping to enable the use of a high compression ratio while avoiding this issue.

Figure 6 shows the mean effective combustion chamber height, H , against S/B and compression ratio. H was determined by dividing the combustion chamber volume, V_1 , at Top Dead Center (TDC) by the bore cross-sectional area, A . The empty circles in Fig. 6 show the engine specifications displaying the highest compression ratios at which there was no increase in unburned HC or decline in combustion speed. When the average combustion chamber height, H , falls below 7 mm, there is a sudden

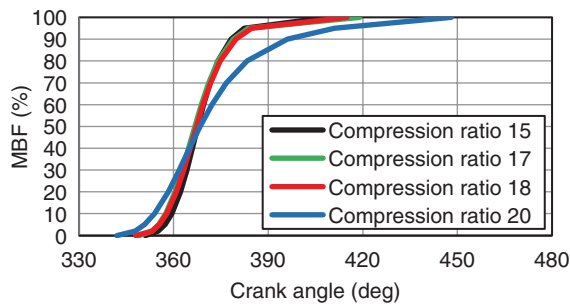


Fig. 4 Comparison of MBF for different compression ratios at S/B 1.5, engine speed 2000 rpm, IMEP 520 kPa, ignition timing MBT, and EGR ratio 20%

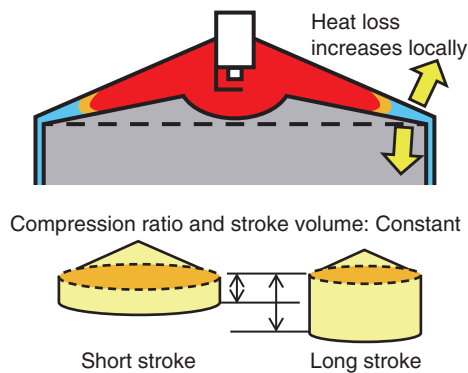


Fig. 5 Diagram of increase in local heat loss in flat combustion chamber and comparison of combustion chamber height for different S/B

decline in combustion speed and increase in unburned HC. Based on these results, it was judged that an S/B of 1.4 or higher would be necessary to obtain good combustion at a compression ratio of 17 and an average combustion chamber height of 7 mm or higher.

3.1.2. Effect in reducing cooling loss

The use of a long stroke was not only important to the realization of a compression ratio of 17, but could also be expected to have the effect of reducing cooling loss as a result of a reduction in $S/V^{(3)}$. Engines of the type shown in Table 1 were modified and thermal efficiency was measured at S/B of 1.2 to 2.0 at a bore diameter of $\phi 81$ and a compression ratio of 13.5 in order to determine the effect of S/B on thermal efficiency. Figure 7 shows results for an engine speed of 2000 rpm and an IMEP of 720 kPa. Brake thermal efficiency tends to increase from S/B of 1.2 to 1.5 and decline at S/B higher than 1.5. This is because the increase in Friction Mean Effective Pressure (FMEP) and Pumping Mean Effective Pressure (PMEP) due to the increased piston speed at S/B of above 1.5 exceeds the effect of reduction of cooling loss. The reduction in the effect of reduction of cooling loss is considered to be due to an increase in the heat transfer coefficient of the surfaces of the cylinder walls resulting from the enhanced in-cylinder flow produced by increased piston speed. The highest value of thermal efficiency occurs at an S/B of 1.5 with a balance between cooling loss and friction.

Up to this point, studies had been conducted for a fixed bore of $\phi 81$. A single-cylinder engine with a bore of $\phi 73$ was now manufactured, and tests were conducted under the same conditions as employed for the engine with a bore of $\phi 81$ in order to determine the optimum S/B when the bore was changed against a fixed displacement. Based on

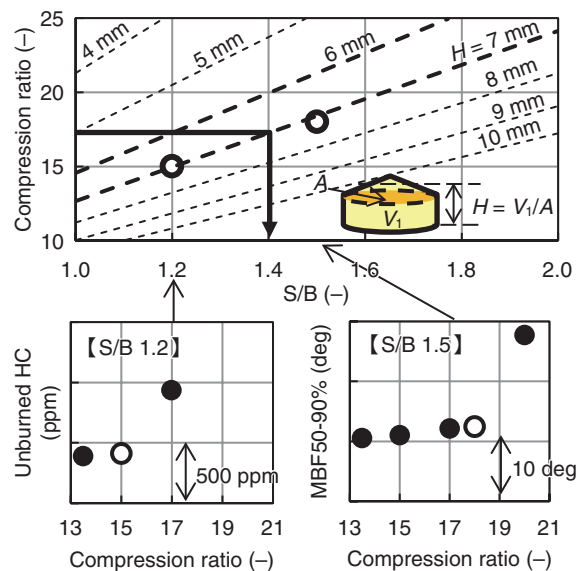


Fig. 6 Average combustion chamber height at TDC for different S/B and compression ratios

the test results for the $\phi 81$ and $\phi 73$ engines, the trend of change in combustion speed due to change in cooling loss in response to S/V and enhancement of in-cylinder flow in response to S/B was calculated, and FMEP was calculated with consideration of changes in bore and stroke, making it possible to estimate brake thermal efficiency in relation to stroke volume and S/B corresponding to conditions in an L4 engine. Figure 8 shows results for the estimation of maximum brake thermal efficiency at an engine speed of 2000 rpm and an EGR rate of 20%. The circled areas in Fig. 8 show the specifications that were actually tested. The estimation results show that even with changes in bore and displacement, the tendency for thermal efficiency to reach its highest level at around S/B of 1.5 remains unchanged.

Based on the above, 1.5 was selected as the optimum S/B that would both provide the necessary S/B for a compression ratio of 17, as discussed in the previous chapter, and increase thermal efficiency to its maximum level.

3.2. Atkinson Cycle with Delayed Closure of Intake Valve

Knocking is another significant issue associated with the use of a high compression ratio. This issue was responded to in the research under discussion by delaying the closure of the intake valve without changing the mechanical compression ratio (equivalent to the expansion

ratio) in order to reduce the effective compression ratio. Using the S/B 1.5 engine shown in Table 1, a comparison was conducted of thermal efficiency when the timing of closure of the intake valve was changed at a mechanical compression ratio of 17 and thermal efficiency during Otto cycle operation at a compression ratio of 17. Figure 9 shows the results of measurements of brake thermal efficiency for each Brake Mean Effective Pressure (BMEP)

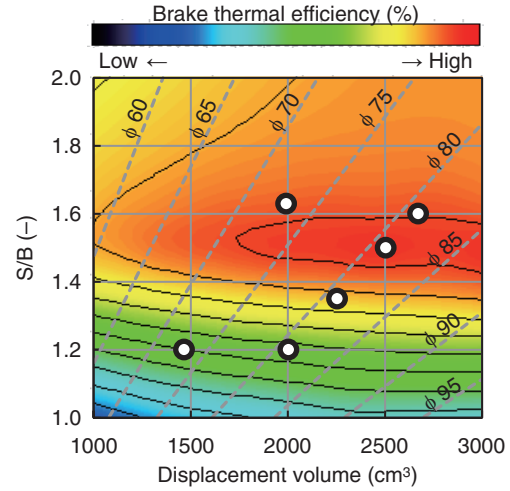


Fig. 8 Engine displacement and S/B in relation to maximum brake thermal efficiency

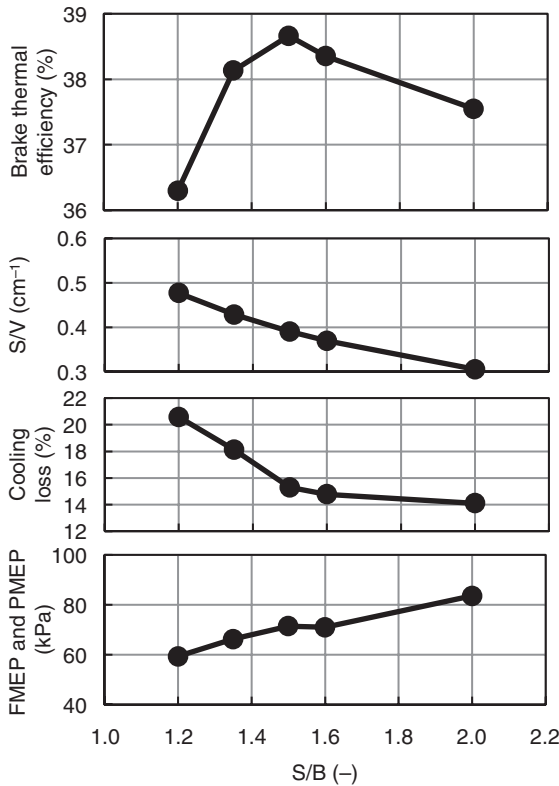


Fig. 7 Effect of S/B on thermal efficiency, cooling loss and friction

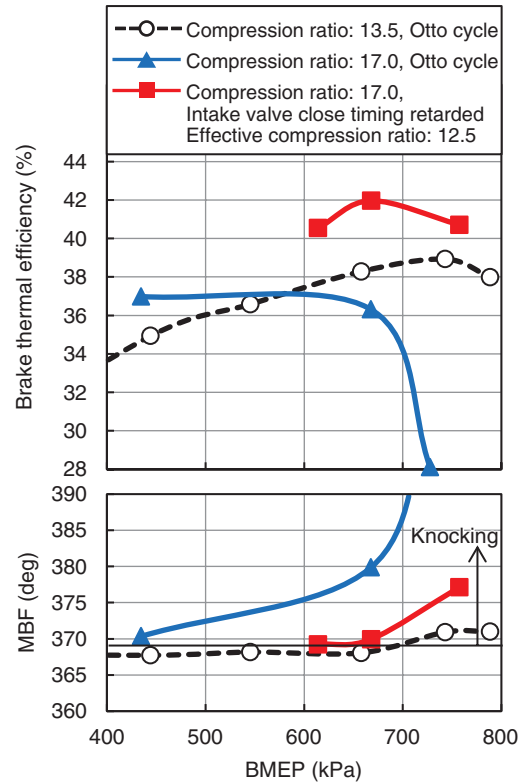


Fig. 9 Effect of effective compression ratio on brake thermal efficiency and knocking

figure at an engine speed of 2000 rpm and an EGR rate of 20%. The reduction of the effective compression ratio made it possible to increase the maximum thermal efficiency by helping to enable MBT operation up to a load close to that of an Otto cycle with a compression ratio of 13.5 and an expansion ratio of 17.

Figure 10 shows theoretical thermal efficiency when the effective compression ratio is changed at an expansion ratio of 17 and theoretical thermal efficiency for Otto cycle operation. The test results show that the theoretical thermal efficiency at an expansion ratio of 17 and an effective compression ratio of 12.5 is equivalent to the theoretical thermal efficiency at a compression ratio of 16.5 during Otto cycle operation, demonstrating that it was possible to realize a level of theoretical thermal efficiency equivalent to that for a compression ratio of 16.5 while achieving a level of knocking equivalent to that for a compression ratio of 13.5.

3.3. Direct Injection Technology

In realizing the use of a high compression ratio, the use of the latent heat of vaporization of the fuel in order to ameliorate knocking was taken into consideration, and direct injection was adopted for the final specifications, which will be discussed in Chapter 5. Figure 11 shows knocking, unburned loss, Indicated Specific Fuel Consumption (ISFC), and the amount of fuel adhering to the pistons and the cylinder sleeves (as calculated using a 3D model) in relation to injection timing at an engine speed of 2000 rpm and an IMEP of 810 kPa. The issue of using direct injection was the fact that injection of the fuel at a timing at which the intake flow speed was high would promote atomization of the fuel and advance knocking ignition timing, but the use of a fuel injection timing that would enable maximum control of knocking would increase unburned fuel loss and ultimately make it impossible to increase thermal efficiency. The main factor in unburned fuel loss is adhesion of fuel to the

walls of the combustion chamber. At an injection timing that ameliorated knocking, fuel adhered to the cylinder sleeves. In order to reduce the amount of fuel adhering to the cylinder sleeves, the fuel spray was optimized to travel downwards, and the fuel injection timing was set close to the timing that would maximize the effect of amelioration of knocking and minimize the amount of fuel adhering to the walls of the combustion chamber, in order to realize the effect of direct injection to the greatest possible extent.

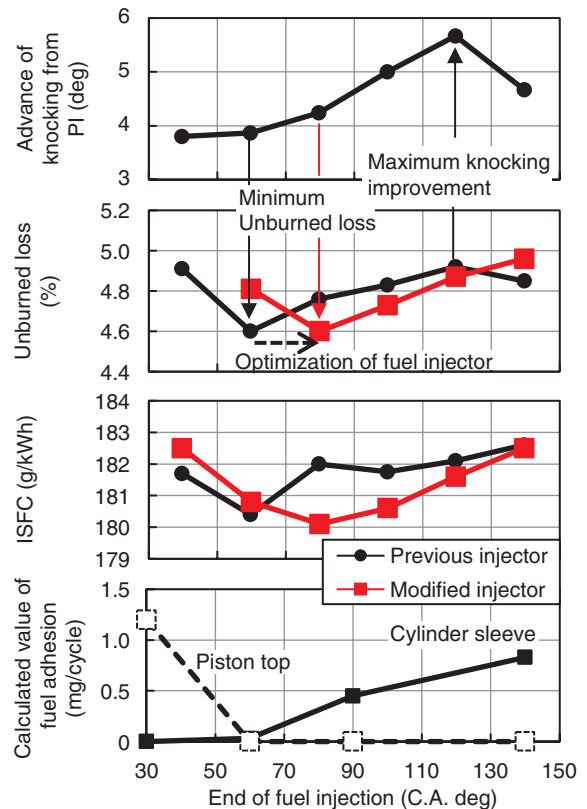


Fig. 11 Effect of fuel injection timing on knocking, unburned loss and ISFC, and calculated value of fuel adhesion

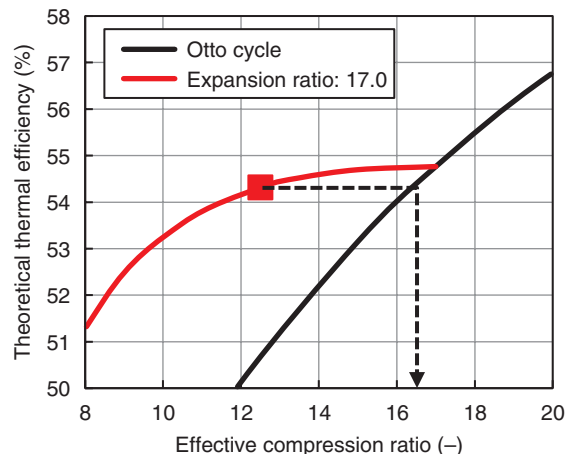


Fig. 10 Comparison of theoretical thermal efficiency

4. High-EGR Operation

Stable operation at an external EGR rate of 30% or higher was set as a target to help enable the reduction of cooling loss and the control of knocking, and the realization of MBT operation at a high compression ratio. The main issues presented by high-EGR operation are reduction of the speed of flame propagation and decline in ignition performance. The reduction in the speed of flame propagation was responded to by enhancement of in-cylinder flow through selection of the intake port shape. The decline in ignition performance

was responded to by selecting a combustion chamber shape that would help to eliminate factors impeding initial flame growth and adjust the vortical center of the in-cylinder flow, in addition to the use of high-energy sparking. If either of the respective measures to respond to the speed of flame propagation and sparking performance was lacking, it would not be possible to realize stable operation while increasing the EGR rate. The application of both measures was important.

4.1. Enhancement of In-cylinder Flow

Methods of increasing the tumble ratio are widely used in order to increase in-cylinder turbulence energy to the level necessary to help ensure the desired flame propagation speed in a high-EGR environment. Restriction of the size of the intake port is the most widely used method of increasing the tumble ratio, but this method basically results in a trade-off with the flow coefficient. Reduction of the flow coefficient increases the load on the supercharger, increasing pumping loss and resulting in a decline in thermal efficiency. In addition, even if the tumble ratio is increased, in-cylinder turbulence energy will not necessarily increase. Figure 12 shows calculated values for tumble ratio and turbulence energy for different intake port shapes. The results indicate that turbulence energy varies significantly even at the same tumble ratio. It is therefore important to control the increase in the tumble ratio to the minimum while efficiently producing turbulence energy.

The in-cylinder non-steady-state flow speed distribution was calculated for Specifications A and B in Fig. 12. Figure 13 shows 5 m/s isosurfaces during the intake stroke. If the isosurfaces are considered the axes of the vortical centers, while the axis of the vortical center for Specification A, for which turbulence energy is high, is comparatively stable, the axis of the vortical center for Specification B undulates significantly during

the intake stroke. This undulation of the axis of the vortical center is considered to be energy loss, producing a decline in turbulence energy. The direction of the main intake flow from the intake valve was focused on as a factor in this undulation. For Specification A, the main intake flow moves towards the outside of the bore, while for Specification B, the main intake flow moves towards the inside of the bore. The flow balance was defined as shown in Fig. 14⁽⁴⁾ in order to quantitatively express the directions of the main flows. When the flow balance is positive, the main flow moves in the direction of the outside of the bore, and when it is negative, the main flow moves in the direction of the inside of the bore.

Figure 15 shows values calculated for turbulence energy when the flow balance was changed while the tumble ratio remained the same. Turbulence energy was highest for flow balances from 0 to 2. Intake port shapes that would generate flow balances within this range were calculated using 3D models. An intake port shape that generated a tumble ratio 1.7 times higher than the mass-production engine and a flow balance between 0 and 2 was reflected in the final specifications.

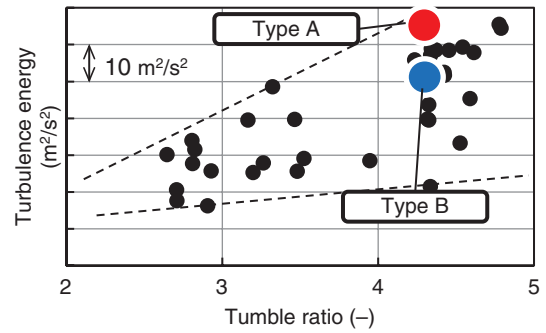


Fig. 12 Calculation results for tumble ratio and turbulence energy for different intake port shapes

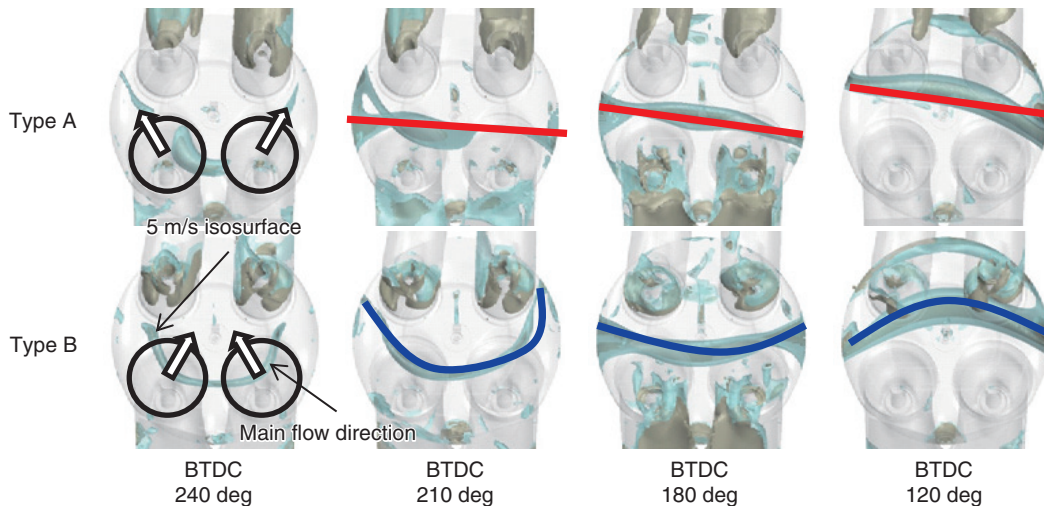


Fig. 13 Comparison of vortical center axes resulting from direction of intake flow

4.2. Measures to Respond to Reduced Ignitability

In a high-EGR environment, the initial flame tends not to grow due to the decline in the laminar burning velocity, and the enhancement of the in-cylinder flow in order to reduce the combustion period extinguishes the spark-plug spark and reduces ignitability. This section will discuss the use of a combustion chamber shape to promote the growth of the initial flame and the enhancement of spark energy in order to help prevent extinguishing the flame by in-cylinder flow.

4.2.1. Realizing ignitability in relation to combustion chamber shape

The initial flame that is produced in a high-EGR environment makes contact with the pistons, cylinder heads or other walls, and this can result in heat loss and the extinguishing of the flame or reduction of the speed of flame propagation. In order to help prevent this, it is important to attempt to ensure sufficient volume in the vicinity of the

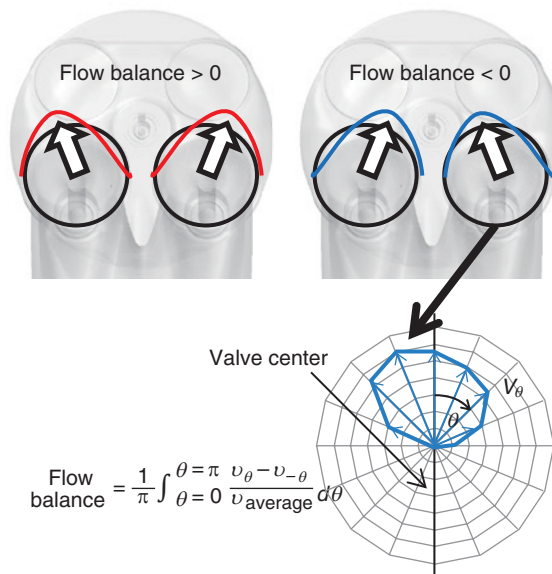


Fig. 14 Definition of flow balance

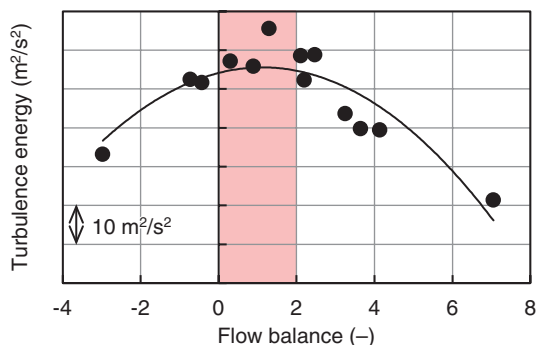


Fig. 15 Turbulence energy for different flow balances

plugs. Figure 16 shows test results for the EGR rate that represents the limit for combustion stability at an engine speed of 2000 rpm and an IMEP of 810 kPa when the volume in the vicinity of the plugs is changed by means of the shape of the piston heads. The ratio between the volume V_1 , when a virtual sphere with the plug electrodes at TDC as its center first makes contact with a piston or a cylinder head; and V , the volume of the entire combustion chamber, (V_1/V), was used to express the distance between the spark plugs and the pistons.

The EGR rate that represents the limit for combustion stability increases as V_1/V increases, becoming relatively constant at ratios of 2% and above. When V_1/V is low, the initial flame makes contact with the piston heads or the walls of the combustion chamber at an early stage, and heat loss results in a decline in the speed of combustion. Alternatively, the flame is extinguished and the limit EGR rate for combustion stability declines. When the initial flame grows to a certain extent, the stability of main combustion rather than ignitability is the dominant factor, and this domain determines the limit EGR rate for stable combustion. Attempting to ensure ignitability and not hindering the growth of the initial flame by realizing a V_1/V of 2% or above is important in realizing high-EGR operation. In order to realize a high compression ratio like 17 and a V_1/V of 2% or above at an S/B of 1.2, as in a mass-production engine, it is necessary to use a convex shape for the piston heads, which is an undesirable shape from the perspectives of heat loss and flame propagation. By contrast, realizing an S/B of 1.5 offers a margin in the direction of the height of the combustion chamber, making it possible to use a shallow-dish shape for the pistons while realizing a V_1/V of 2%. This makes it possible to balance ignitability with good combustion. The use of a long stroke is also effective from this perspective. Figure 17 shows the combustion chamber shapes for S/B of 1.2 and 1.5 at a compression ratio of 17 and a V_1/V of 2%.

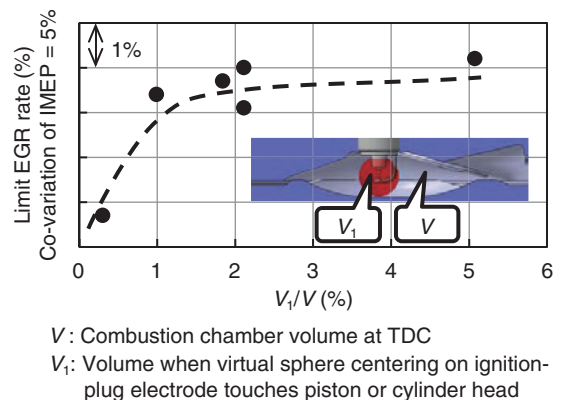


Fig. 16 Effect of V_1/V on EGR rate (Co-variation of IMEP: 5%)

4.2.2. Adjustment of position of vortical center of in-cylinder flow

The position of the vortical center of the in-cylinder flow is an important factor in realizing good ignition performance and forming the initial flame. If there is significant deviation between the positions of the spark plug electrodes and the vortical center of the in-cylinder flow, the in-cylinder flow tends to extinguish the spark-plug spark, resulting in a decline in ignition performance. In addition, even if the initial flame formation is successful, the flame will be carried by the flow and diverted to one side of the combustion chamber, extending the combustion period. It is necessary to align the positions of the spark plugs and the vortical center in order to help avoid these issues. A shallow-dish shape was employed for the piston heads in order to adjust the position of the vortical center. Figure 18 shows calculation results for in-cylinder flow for the flat piston shape and the shallow-dish piston shape. At close to spark timing, the vortical center is forced towards the bore center by the shallow-dish shape, aligning the positions of the spark plugs and the vortical center.

In addition to these modifications, further changes including the reduction of the piston-top land height and the squish height were made to the combustion chamber shape in order to reduce unburned HC and increase combustion speed, and these were reflected in the final specifications⁽⁵⁾, to be discussed in Chapter 5.

4.2.3. High-energy sparking

Even when sufficient volume had been realized in the vicinity of the spark plugs and the position of the vortical center had been optimized, the enhancement of spark

energy was an important factor in igniting the air-fuel mixture and forming the initial flame in the high-EGR environment. In the research under discussion, the spark was enhanced by means of multiple discharge in order to realize high-energy sparking. Up to this point, the single-cylinder engine had been modified in order to produce a high flow rate generating a tumble ratio 1.7 times higher than that of a mass-production engine and fitted with a combustion chamber shape that did not hinder flame propagation. Tests were now conducted in order to determine how the limit EGR rate for stable combustion would change with change in the spark energy employed. Energy per unit time was held constant, and the period of supply of energy was changed in order to adjust the amount of energy supplied to the spark plugs. Figure 19 shows the results. The limit EGR rate for stable combustion increases with the amount of energy supplied, becoming relatively constant at 1500 mJ and above. It is considered that in the domain at 1500 mJ and above, rather than an issue of ignitability, the in-cylinder flow is insufficient and the main combustion period varies, increasing co-variation of IMEP and determining the limit EGR rate. In order to further increase the limit EGR rate, it was necessary to further enhance the in-cylinder flow. Because power consumption increases with increase in the amount of energy supplied, becoming a factor in reducing efficiency, a setting of 450 mJ was ultimately adopted in order to help enable ignition at the target EGR rate of 30% and above and minimize any effect on fuel consumption.

The measures discussed above to increase the flow rate, modify the shape of the combustion chamber, and realize high-energy sparking in order to help enable high-EGR operation were applied in a single-cylinder engine, and tests were conducted in order to verify their effect. Figure 20 shows test results for an engine speed of 2000 rpm and an IMEP of 810 kPa. The results demonstrate that even at an external EGR rate of 30% and above, operation is stable, and the effect of the high EGR rate in reducing cooling loss and controlling knocking has increased brake thermal efficiency.

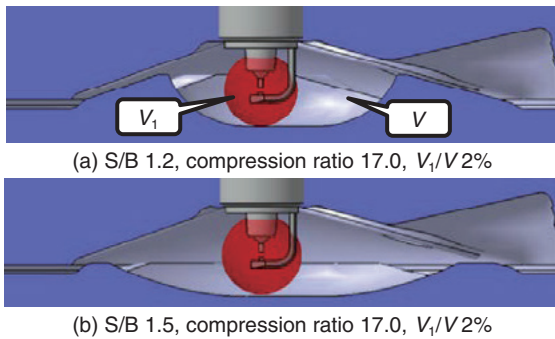


Fig. 17 Comparison of combustion chamber configuration for S/B 1.2 and 1.5

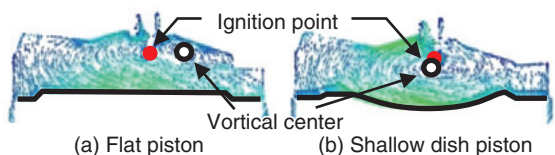


Fig. 18 Influence of piston shape on vortical center

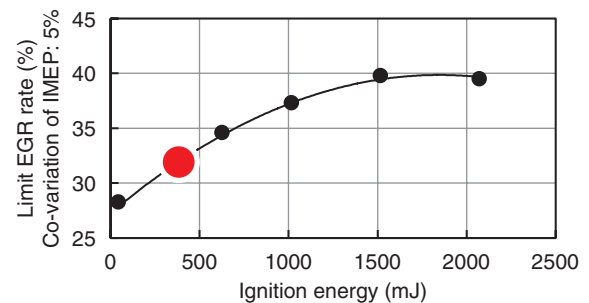


Fig. 19 Influence of ignition energy on EGR rate (Co-variation of IMEP: 5%)

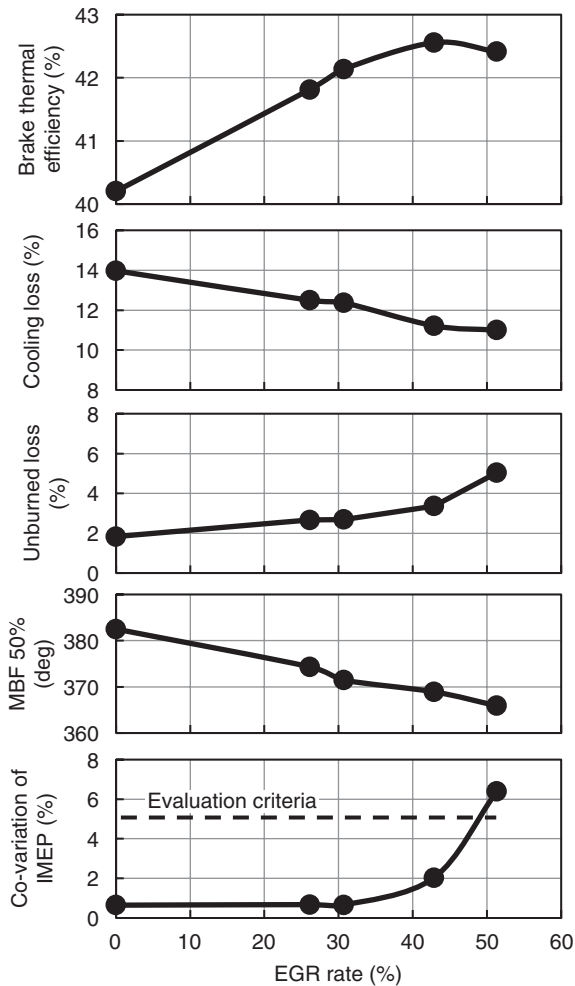


Fig. 20 Change in thermal efficiency due to increased EGR rate

5. Achieved Thermal Efficiency

A single-cylinder engine incorporating the basic framework, technologies for the control of knocking, technologies to increase in-cylinder flow, and technologies to increase ignitability discussed above was manufactured, and the achieved values were tested. Table 2 shows the final specifications of the engine, and Fig. 21 shows results for thermal efficiency. The increase in the potential of the basic framework realized through the use of a long stroke, the increased theoretical thermal efficiency realized through the use of a high compression ratio, the enhanced in-cylinder flow realized through the modification of the shapes of the intake port and the combustion chamber, and the application of high-energy sparking and direct injection have resulted in the achievement of a brake thermal efficiency of 45.2%.

Table 2 Final test engine specifications

Engine type	DOHC 4-valve single cylinder
Bore (mm) × Stroke (mm)	φ81 × 121.6 S/B 1.5
Compression ratio (–)	12.5
Expansion ratio (–)	17.0
Displacement (cm ³)	627
Intake port	Tumble port
Air supply	Supercharged
EGR system	Low-pressure loop-cooled EGR
Fuel supply	DI
Piston shape	Shallow dish
Ignition energy (mJ)	450

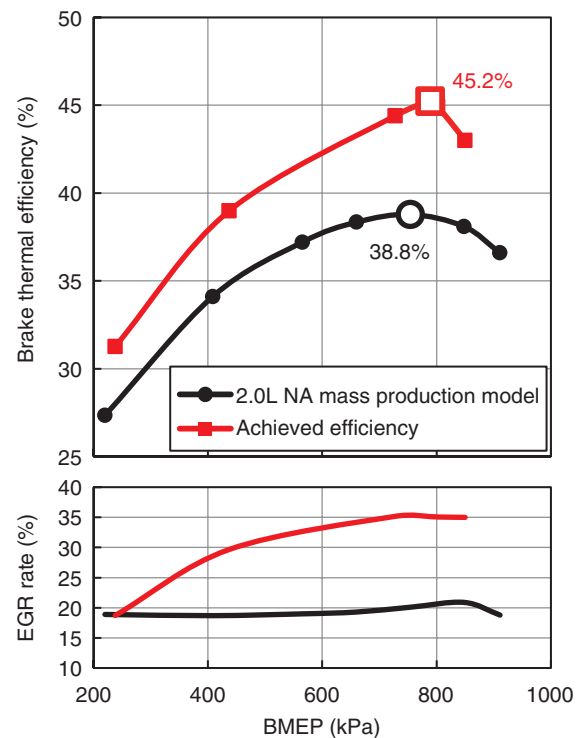


Fig. 21 Achieved brake thermal efficiency at engine speed of 2000 rpm

6. Conclusion

The research discussed in this paper realized MBT operation using 91-octane regular gasoline at a stoichiometric air-fuel ratio with a mechanical compression ratio of 17 and an EGR rate of 35%, increasing brake thermal efficiency to 45.2%.

A combustion chamber shape exploiting the freedom in shape design offered by a long stroke (S/B 1.5) was selected and the timing of the closure of the intake valve was delayed, making it possible to advance knocking spark timing through the use of an Atkinson cycle. This

helped to enable the realization of MBT operation up to a load equivalent to that of an Otto cycle engine with a compression ratio of 13.5 while controlling reduction in the speed of combustion even at a high compression ratio (a mechanical compression ratio of 17).

The selection of an intake port shape that balanced intake resistance with the enhancement of turbulence energy during high-EGR operation, the use of a combustion shape that helped to ensure sufficient volume in the vicinity of the spark plugs and optimized the flow, and the application of 450 mJ high-energy sparking have made it possible to realize stable operation even at an EGR rate of 30% and above.

7. Afterword

The achieved thermal efficiency indicated in Chapter 6 was realized in a single-cylinder engine. Numerous issues remain to be resolved in order to help enable the technologies discussed in this paper to be applied in multi-cylinder automotive engines.

The main issues are the increase in engine height due to the use of a long stroke and measures to respond to condensed water and increase EGR responsiveness in order to realize a Low-Pressure Loop-cooled EGR (LPL-EGR) system for high-EGR operation. In addition, in order to respond to the reduction in exhaust energy due to the use of a high expansion ratio and high-EGR operation, it will be necessary to develop a boost system that balances the supercharger efficiency necessary for increased thermal efficiency with the flow-rate range necessary for increased maximum power.

It will be important to address these issues at an early stage in order to apply the technologies discussed in this paper to automotive gasoline engines. The authors believe that this will be an important focus for future development initiatives.

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