

Optimization of Combustion Chamber Shape in High-efficiency Stoichiometric Combustion Gasoline Engine

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ABSTRACT

The combustion chamber shape of an engine with high compression ratio, long stroke, high-quantity cooled EGR, and retarded intake-valve-closing specifications was optimized to increase the EGR limit and minimize unburned loss and time loss with the aim of enhancing the brake thermal efficiency of a stoichiometric gasoline engine. A piston shallow-dish shape was adopted to optimize the tumble center and the piston top surface distance from the spark plug, and the combustion chamber peripheral shape was optimized to minimize unburned loss. The resulting enhancement of the external EGR rate limit and increase in combustion speed reduced both time loss and unburned loss, and achieved a brake thermal efficiency of 45.2%.

1. Introduction

Recently, interest is increasing in technology for achieving high-efficiency engines with good fuel economy and emissions characteristics from the viewpoint of global environmental issues and reduction of fuel costs.

In order to increase efficiency it is necessary to enhance theoretical thermal efficiency by increasing the compression ratio. Issues for increasing the compression ratio include a delayed ignition timing angle due to thermal loss and knocking, slower combustion speed due to a flatter combustion chamber shape, and reduced thermal efficiency due to increased unburned loss. Introduction of cooled EGR is an effective means of reducing this thermal loss and reducing knocking⁽¹⁾. However, introduction of high-capacity EGR has the issue that the combustion limit and unburned loss increase. In addition, reduction of the effective compression ratio by retarding the intake valve-closing timing (Atkinson cycle) is an effective measure against knocking, but unburned loss due to reduced flow becomes an issue⁽²⁾. Use of the latent heat of vaporization due to direct injection is an effective measure against knocking produced by a higher compression ratio, but there is the issue that the knocking enhancement effect is reduced by the adherence of fuel to the combustion chamber wall surface⁽³⁾.

An effective method of addressing these issues accompanying an increase in the compression ratio is to increase the length of the engine stroke. Use of a long stroke enables reduction of the flattening of the combustion chamber at the top dead center (TDC). In addition, the surface/volume ratio (S/V ratio) also decreases, enabling reduction of thermal loss. Furthermore, use of a long stroke increases the piston speed, which also enhances the combustion limit by strengthening the flow^{(4), (5)}.

As described above, the introduction of cooled EGR and use of a long-stroke direct injection engine are effective measures when increasing the compression ratio with the aim of enhancing the brake thermal efficiency of the engine. Potential methods for further increasing engine efficiency in this manner include increasing the EGR limit, reducing unburned loss, and reducing adherence of the direct injection spray to the combustion chamber wall surface.

A long-stroke engine has some degree of freedom in the combustion chamber shape at the same compression ratio. This study positioned enhancement of combustion by the combustion chamber shape as an important factor for pursuing brake thermal efficiency. Investigations were conducted with the aim of presenting a combustion chamber shape that enhances combustion by making use of the degree of freedom of the combustion chamber obtained

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by increasing the stroke length. The aim of this study was to enhance combustion in terms of the following items by optimizing the combustion chamber shape to enhance the brake thermal efficiency at the stoichiometric air-fuel ratio of a direct injection Atkinson engine with a high compression ratio and a long stroke.

- (1) Enhancement of the EGR limit by increasing ignitability
- (2) Reduction of time loss by shortening the combustion period
- (3) Reduction of unburned loss by changing the combustion chamber wall surface

This paper describes enhancement of the combustion chamber shape, adaptation of the flow and the direct injection spray shape to this new combustion chamber shape, and the resulting achievement of a brake thermal efficiency of 45.2% at the stoichiometric air-fuel ratio in a direct injection Atkinson engine with a high compression ratio and a long stroke.

2. Study Method

2.1. Single-cylinder Engine

Table 1 shows the specifications of the base engine used to optimize the combustion chamber shape.

The brake thermal efficiency (η_e) was obtained from the following equation using the mechanical friction ($FMEP$) and pumping loss ($PMEP$) of a four-cylinder engine relative to the indicated thermal efficiency (η_i) and indicated mean effective pressure ($IMEP$) of a single-cylinder engine.

$$\eta_e = \frac{IMEP - FMEP - PMEP}{IMEP} \times \eta_i$$

$BMEP$ in this paper refers to the brake mean effective pressure. In addition, the EGR limit in this paper is the value at which the $IMEP$ cycle fluctuation exceeds 3%.

2.2. Visualization Engine

Table 2 shows the specifications of the visualization engine.

Visualization was performed by recording the naturally emitted light with a high-speed camera via a side window (Fig. 1). All visualization in this study was performed at an engine speed of 2000 rpm.

2.3. CFD

The flow inside the cylinder was evaluated by 3D-CFD of the engine described in Table 1. Turbulent kinetic energy (TKE) was used as an index of the flow inside the cylinder. Evaluation of TKE inside the engine combustion chamber mainly used 3D-CFD software (VECTIS) made by Ricardo Software. The standard k- ϵ was used as the turbulence model. Calculation of the boundary conditions used 1D-CFD software and the engine simulation code WAVE made by Ricardo Software.

All CFD in this study was performed at engine speed = 2000 rpm.

Table 1 Specifications of test engine

Bore (mm) × Stroke (mm)	φ 81 × 121.6 S/B 1.5
Compression ratio (–)	12.5
Expansion ratio (–)	17.0
Engine type	DOHC 4-valve single cylinder
Number of cylinders	1
Displacement (cm ³)	627
Intake port	Tumble port
Air supply	Supercharged
Fuel supply	PI/DI
Piston shape	Shallow dish
Ignition energy (mJ)	450
Gasoline	RON91

Table 2 Specifications of visualization engine

Bore (mm) × Stroke (mm)	φ 81 × 121.6 S/B 1.5
Compression ratio (–)	11.0
Expansion ratio (–)	15
Engine type	DOHC 4-valve single cylinder
Number of cylinders	1
Displacement (cm ³)	627
Intake port	Tumble port
Air supply	Supercharged
Fuel supply	PI
Ignition energy (mJ)	450
Gasoline	RON91

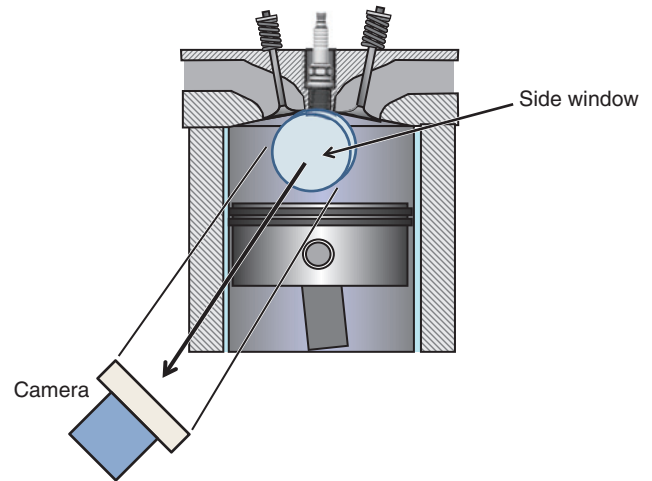


Fig. 1 Method of visualizing combustion

3. Results and Examination

This chapter describes the investigation of methods for achieving each goal and the confirmation results for the combustion chamber shape used to achieve the three goals described in Chapter 1.

3.1. Enhancement of EGR Limit by Increasing Ignitability

In order to enhance the combustion limit when introducing high-capacity EGR, it is necessary to enhance

ignitability. The initial flame behavior is an important factor for enhancing ignitability. Therefore, the factors influencing the initial flame behavior were identified, combustion chamber shapes that enhance this behavior were investigated, and the effects were confirmed in an actual engine.

3.1.1. Combustion chamber shapes that enhance initial flame behavior

Tests were performed using a visualization engine to understand the current initial flame behavior.

Figure 2 shows the results of visualizing the initial flame with a high-speed camera via the side window. The left side in the figure is the IN side (hereafter, the left side is the intake port in all combustion chamber figures), and the light-gray area shows the flame inside the cylinder at ATDC -12 deg. The center of gravity of this light-gray area is labeled “D.” The flame in the figure approximates a circle of equal radius centering on D. The radius of this circle is labeled “R.”

The dots in Fig. 3 plot the flame center of gravity positions at the timing when $R = 10$ mm for each cycle. In addition, the star indicates the average position of the flame center of gravity positions for 100 cycles. Assuming the flame is globe-shaped, the timing when $R = 10$ mm is defined as the heat generation start point, and is thought to reflect the initial combustion.

These results show that the initial flame center of gravity position varies within the combustion chamber. In particular, there is insufficient space in the combustion chamber height direction, so there are concerns that flame growth may be impeded by contact with the piston top

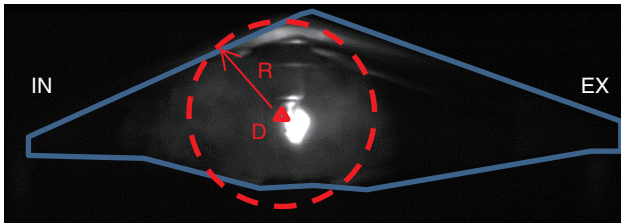


Fig. 2 Initial flame image by visualization

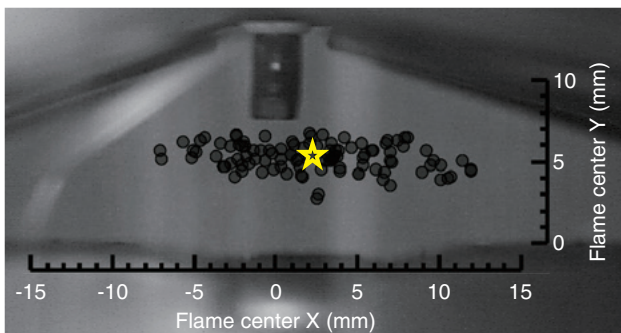


Fig. 3 Cycle variation of initial flame position

surface and the head.

The visualization engine test results indicate that potential measures for avoiding contact between the initial flame and the combustion chamber wall surface include reduction of contact between the initial flame and the piston at the cycle average, and reduction of cycle variation in the flame center of gravity position. Specific methods for achieving each of these are described below.

The influence of piston shape on the mass-combustion rate at the timing when the flame has grown to contact the piston after ignition was investigated by calculations. Here, the flame was assumed to grow around the center of the ignition point, and the mass-combustion rate used the ratio of the flame volume to the combustion chamber volume at the radius when the flame contacts the piston top surface, at the piston TDC position.

The piston shape was investigated with the assumption that the mass-combustion rate at the time of contact increases together with the piston top surface distance from the bottom of the spark plug (plug). A flat piston shape was used as the base shape (A in Fig. 4), and investigations were performed for a shape joined by a large curve (B in Fig. 4) and a shape further carved out only below the plug (C in Fig. 4).

Figure 4 shows the calculation results of the mass-combustion rate at the time of contact between the initial flame and the piston according to the piston shape. All results in the figure are the results at the piston TDC position. The ignition point was set to the midpoint between the head and the piston top surface.

In order to enhance ignitability, it is thought that the flame surface should contact the piston after the flame has grown sufficiently. This investigation found that contact between the flame surface and the piston top surface can be delayed by securing a sufficient piston top surface distance from the bottom of the plug.

When using the base piston shape (A), the flame contacts the piston at the timing when the mass-combustion

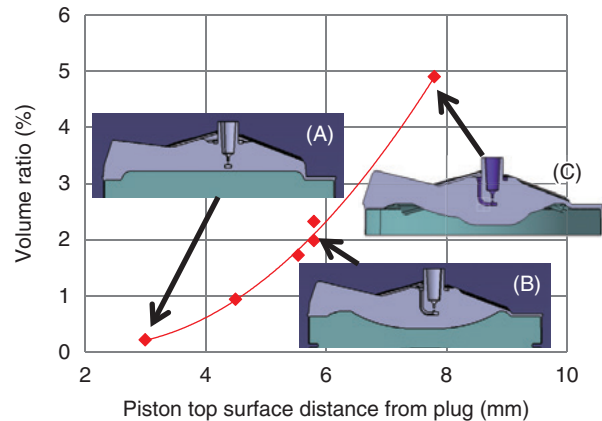


Fig. 4 Calculation results of contact timing between flame and piston top surface

rate is less than 1%. In contrast, the enhanced piston shape (C) provided prospects for delaying the mass-combustion rate at the time of contact by up to 5%.

The mass-combustion rate at the time of contact required during high EGR combustion was confirmed by tests.

Next, regarding a combustion chamber shape that stabilizes initial flame behavior, CFD and visualization tests were performed to organize the approach to stabilizing initial flame behavior.

This engine uses a flow port with strong tumble to support slower combustion speeds due to the introduction of high-capacity EGR. Therefore, bulk flow of approximately 30 m/s is present at the ignition point even during ignition (BTDC 40 deg) (Fig. 5).

Figure 6 shows the results of visualizing the discharge arc shape inside the cylinder during strong flow. The bulk flow indicated in the figure is thought to be present at the ignition point, and this figure shows that the discharge arc is shifted by this bulk flow.

This bulk flow is thought to shift the initial flame from the ignition point. The cycle variation of the initial flame center of gravity shown in Fig. 3 is thought to be a result of fluctuation in the bulk flow near the ignition point for each cycle. To reduce the cycle variation of the initial flame position and obtain stable ignition, it is considered necessary to reduce the bulk flow variation near the ignition point. The bulk flow near the ignition point needs to be reduced while maintaining sufficient flow, and it is thought that this can be effectively achieved by bringing the vortex center of the in-cylinder flow closer to the ignition point.

A combustion chamber shape that brings the vortex center closer to the ignition point was investigated.

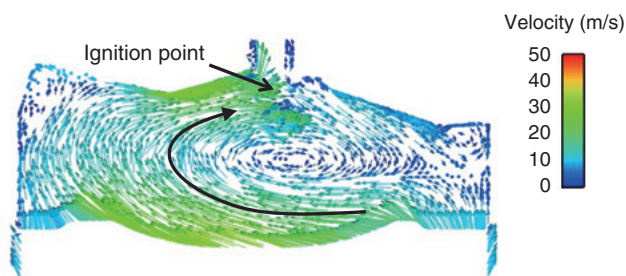


Fig. 5 Bulk flow from tumble

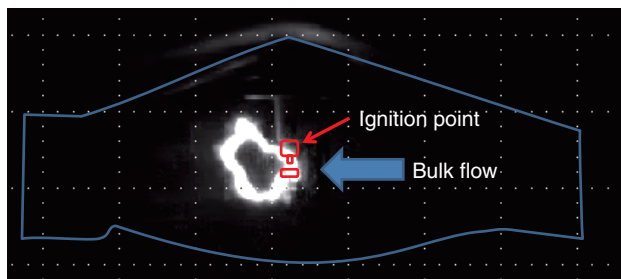


Fig. 6 Bending of discharge arc by bulk flow

Figure 7 shows the difference in the behavior of the tumble vortex center for different piston top surface shapes as calculated by CFD. The base piston shape has a flat piston top surface, so the vortex center is offset from the ignition point. In contrast, for the piston with a shallow-dish top surface shape, the vortex center tends to move towards the center of the shallow dish near the TDC.

It is thought that locating the shallow-dish center directly below the plug enables stabilization of the initial flame behavior even in the condition with high in-cylinder flow.

The effects of a shallow-dish piston shape to control the vortex center were confirmed by tests described hereafter.

In order to enhance the combustion limit when introducing high-capacity EGR, it is thought necessary to delay the timing at which the initial flame contacts the piston by increasing the piston top surface distance from the plug.

In addition, it is also thought necessary to reduce the flow speed near the plug and stabilize the initial flame behavior by giving the piston top surface a shallow-dish shape with the center directly below the plug to lead the tumble vortex center close to the plug.

The following section describes confirmation of the combustion enhancement effects of a piston shape based on the above concepts in an actual engine.

3.1.2. Confirmation of effects by engine tests

The effects of piston shape on enhancement of ignitability were confirmed from the above-noted viewpoints. Figure 8 shows the tested piston specifications, including the distance from the top surface of each piston to the ignition point, and the mass-combustion rate at the calculated time of contact between the flame and the piston top surface shown in Fig. 4.

Piston (a) in Fig. 8 is the base piston. Pistons (b)

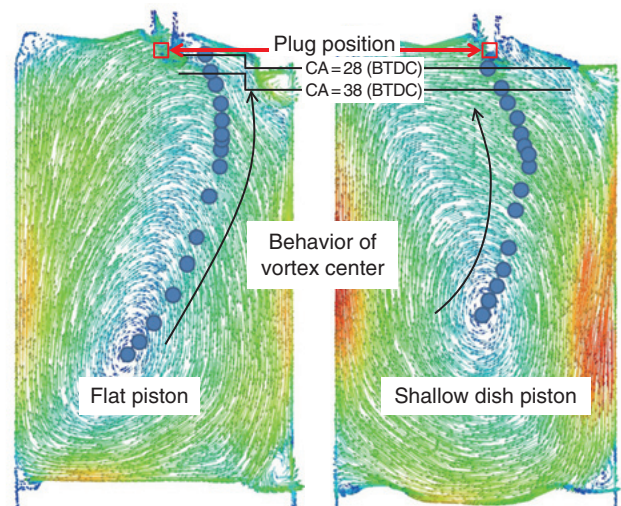


Fig. 7 Behavior of vortex center position by piston shape

through (e) varied the piston top surface distance from the plug with a shallow-dish shape, and piston (f) used a shape carved out below the plug to further increase the piston top surface distance from the plug.

Figure 9 shows the EGR limit relative to the piston top surface distance from the plug. The test was performed under conditions of engine speed = 2000 rpm and IMEP = 810 kPa.

The red line in the figure shows the results of confirming the EGR limit enhancement effect of the pistons shown in Fig. 8. In addition, the black line shows the effect when the plug height is changed with the base piston (flat top surface).

This shows that when the piston has a shallow-dish shape, the EGR limit depends on the piston top surface distance from the plug. However, when using the flat shape, the EGR limit is not enhanced even if the piston top surface distance from the plug is increased by the plug position. In addition, the EGR limit decreased for a piston shape with

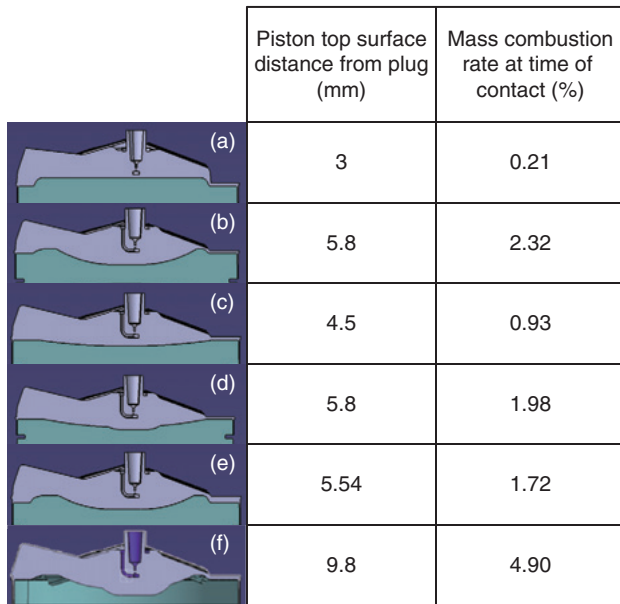


Fig. 8 Test piston shape list

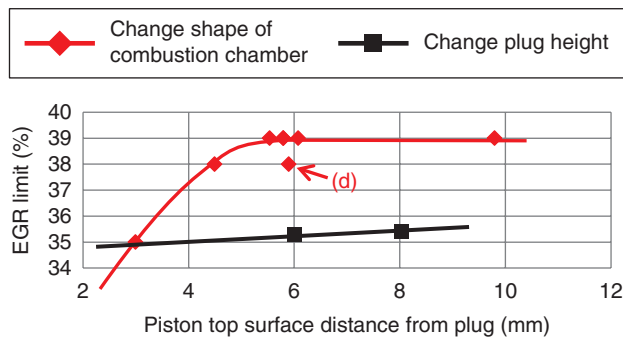


Fig. 9 Test results of EGR limit due to piston top surface distance from plug

the part below the plug carved out to increase the piston top surface distance from the plug, such as piston shape (d) in Fig. 8.

In order to increase the EGR limit by enhancing ignitability, it is necessary to stabilize the initial combustion behavior by using a shallow-dish piston to lead the vortex center close to the plug, and also to secure sufficient distance between the piston top surface and the plug. It was found that a piston top surface distance from the plug of 6 mm is needed to enhance ignitability. The mass-combustion rate at the time of contact in this case is approximately 2%.

3.2. Reduction of Time Loss by Shortening the Combustion Period

The drop in thermal efficiency due to an expanded combustion period is an issue for engines with a high compression ratio and high-capacity EGR introduction.

Therefore, the factors comprising the combustion period were extracted, combustion chamber shapes that shorten the combustion period were investigated, and the effects were confirmed in an actual engine.

3.2.1. Investigation of combustion chamber shape to shorten the combustion period

Figure 10 shows the factors comprising the combustion period.

In order to shorten the combustion period, it is necessary to increase the flame propagation speed and reduce the flame propagation distance.

Of the factors shown in Fig. 10, assuming that the engine framework is not changed, the bore diameter cannot be changed, so the flame propagation distance cannot be reduced. Furthermore, if the intake air is the same, the laminar burning velocity will also be the same.

Therefore, combustion chamber shapes that increase the turbulence strength and combustion chamber shapes that realize flame propagation from the center of the combustion chamber while reducing the combustion chamber size were investigated by calculations and visualization tests.

As shown in Fig. 10, strengthening the flow is an important factor for shortening the combustion period. To obtain sufficient TKE at the compression end inside the combustion chamber, it is necessary to maintain the tumble

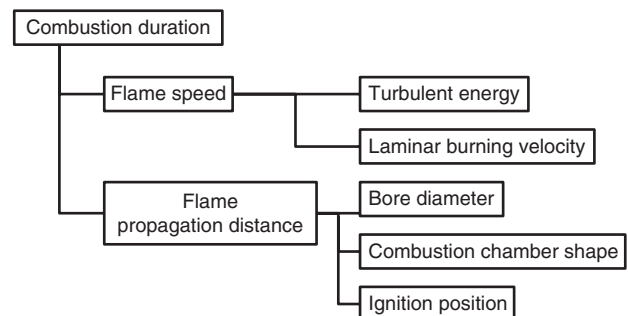


Fig. 10 Tree diagram for combustion period shortening

created by the intake stroke and to form a squish flow in the combustion chamber end-part.

Figure 11 shows the TKE calculation results for the shallow-dish shape. These results show that use of a shallow-dish shape as the piston top surface shape effectively maintains the turbulence strength.

Another approach to increasing the turbulence strength during the compression stroke is to use squish flow. Squish flow creates flow in the combustion chamber end-part during the second half of the compression stroke. This is thought to increase the combustion speed during the second half of combustion and reduce unburned loss.

Figure 12 shows the TKE calculation results by diagonal squish length. The shape with a squish clearance of 14 mm enabled achievement of strong TKE near TDC after a crank angle of -20 deg.

However, the combustion chamber requires a complex shape to realize a squish shape, so there are concerns over an increase in unburned loss and thermal loss.

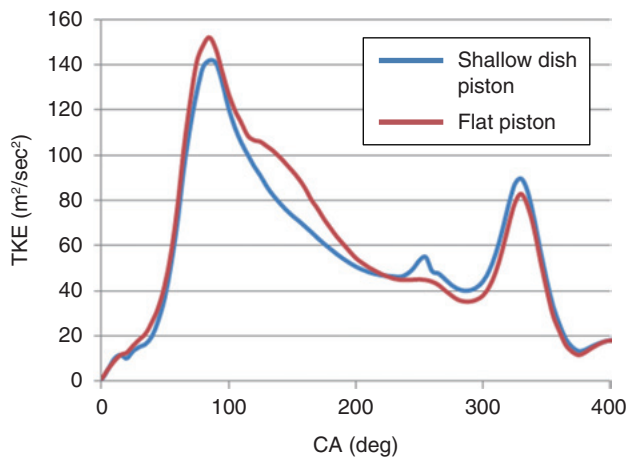


Fig. 11 TKE by piston shape

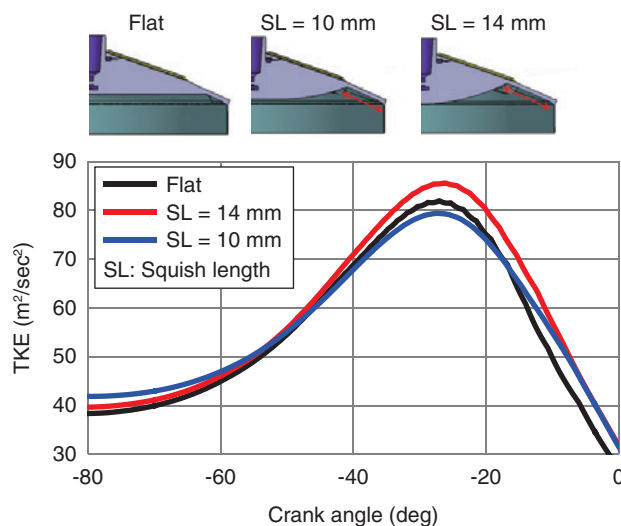


Fig. 12 TKE by squish length

Figure 13 shows the results of visualizing the initial flame with a high-speed camera via the side window. The circles in the figure indicate the flame center of gravity D , and the color of each circle indicates the period from a mass-combustion rate of 50% to 90%. As shown in the figure, the period of the second half of combustion is shorter for cycles in which the initial flame is near the center of the combustion chamber, but the second half of combustion tends to become longer in cycles where the initial flame is shifted toward the intake valve side. The reason for this is thought to be that when combustion starts from one end of the combustion chamber, the flame propagation distance to the opposite end increases. To minimize the flame propagation distance, combustion needs to propagate from the center of the combustion chamber.

The need for a shallow-dish piston shape with the center directly below the plug was described in section 3.1.1. It is thought that locating the vortex center near the plug enables stabilization of the initial flame behavior near the plug and at the same time reduces the flame propagation distance and shortens the combustion period. Therefore, a piston top surface shape that has a shallow-dish shape with the center directly below the piston is also better from the viewpoint of shortening the combustion period.

Figure 14 shows the flame radius and burned volume calculation results. These calculations assume a globe of radius R with the center near the plug to be the flame, and treat the volume V , which excludes the parts cut out from this globe by the combustion chamber wall surface, as the burned area. The burned area reaches the combustion chamber volume at the time when the flame radius is the bore radius of 40.5 mm. The piston shapes used the base shape and the shallow-dish shape shown in Table 3.

Assuming the flame propagation speed to be constant, the flame radius R has a proportional relationship with the period from the start of combustion. The shallow-dish shape piston tends to have a larger burned volume V at the same flame radius R compared to the flat piston, which indicates that the combustion chamber shape of the shallow-dish piston achieves the same mass-combustion rate at an earlier timing. This leads to a shorter combustion period by reduction of the flame propagation distance as shown in Fig. 10.

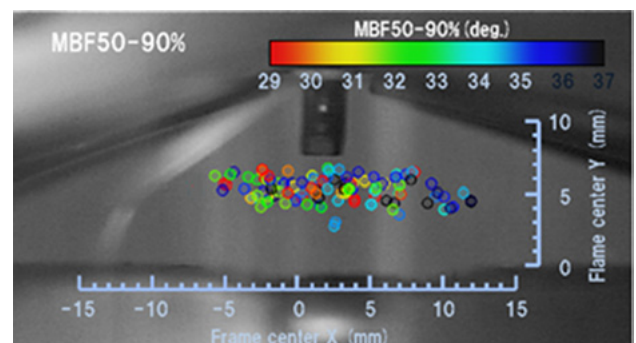


Fig. 13 Combustion period by initial flame position

Table 3 Final piston shape

Piston shape	Swallow dish (wide)
Piston top surface distance from plug (mm)	6
Top land height (mm)	3.5
Squish height (mm)	0.8
Squish	No squish

A compact combustion chamber shape with the volume collected near the plug is thought to be necessary to shorten the combustion period, as expressed by Fig. 14.

Summarizing the above, the following items are considered necessary.

- The piston top surface should use a shallow-dish shape to enhance the flow maintainability in order to shorten the combustion period.
- The piston top shape should use a shallow-dish shape with the center directly below the plug to promote flame growth from the center of the combustion chamber.
- The combustion chamber volume should be collected near the plug to reduce the combustion chamber size and minimize the combustion distance.

The following section describes the results of confirming the combustion enhancement effects of piston shapes based on the above concepts in an actual engine.

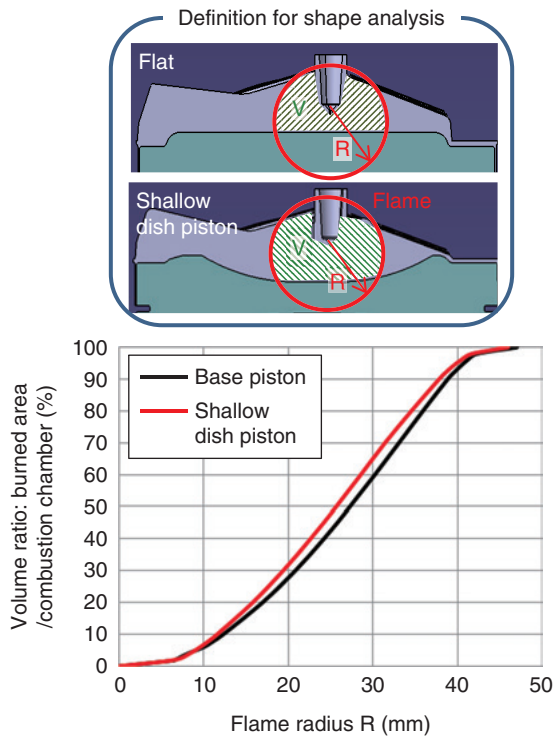


Fig. 14 Difference in burned volume at same flame radius by piston shape

3.2.2. Confirmation of effects by engine tests

Figure 15 shows the effects of strengthening the flow by diagonal squish length on unburned loss and time loss. An engine test was performed using the three types of piston shapes shown in Fig. 12. The test was performed under conditions of engine speed = 2000 rpm and IMEP = 810 kPa.

This test used the high-load condition of IMEP = 810 kPa, so the ignition timing angle is delayed by knocking. The introduction of cooled EGR alleviates knocking and also reduces time loss. However, at the external EGR rate conditions, high EGR lengthens the combustion period, which increases time loss. This same tendency was seen for all piston shapes.

Expanding the squish area did not notably enhance the combustion period. On the other hand, unburned loss increased. The reason for this is thought to be that the flame could not propagate to the squish area, so the air-fuel mixture in the squish area could not combust and became dead volume. This indicated that diagonal squish provided no efficiency enhancement effect.

Next, the results of confirming the effects of using a shallow-dish piston to shorten the combustion period are described as follows. Figure 16 shows the results of confirming the reduction in time loss due to piston shape. This test was performed with the plug height changed in accordance with the piston shape to maintain a piston top surface distance from the plug of 6 mm. The test was performed under conditions of engine speed = 2000 rpm

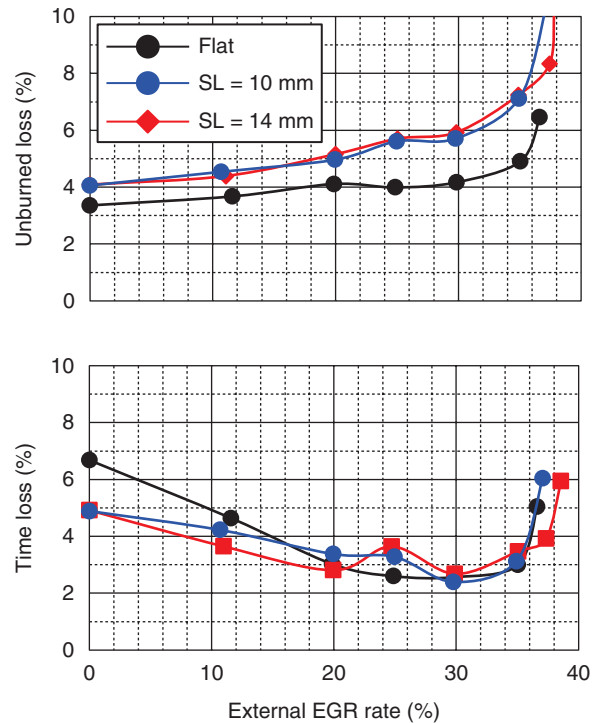


Fig. 15 Effects of squish on unburned loss and time loss

and IMEP = 810 kPa.

Compared to the flat piston, the shallow-dish shapes enabled shortening of the combustion period and reduction of time loss at the same external EGR rate. In addition, combustion was possible without lowering the combustion speed even at higher external EGR rates. This also enabled reduction of time loss by reducing knocking.

3.3. Reduction of Unburned Loss by Changing the Combustion Chamber Wall Surface

A flatter combustion chamber increases unburned loss in engines with a high compression ratio. The reason for this is thought to be that the combustion chamber end-part becomes narrow, so the flame does not reach the end.

3.3.1. Investigation of end-part shape to reduce unburned loss

The combustion chamber end-part shape was investigated focusing on the top land height and the squish height.

The top land is the clearance area between the piston and the sleeve as shown in Fig. 17. This area has almost no thickness, so flame is unable to propagate.

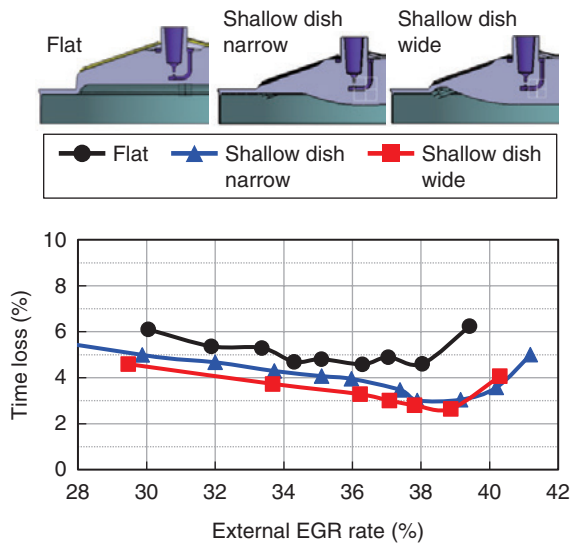


Fig. 16 Time loss reduction by piston shape

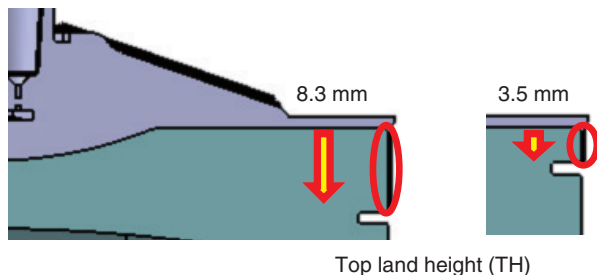


Fig. 17 Definition of top land height

Therefore, unburned loss corresponding to the volume of the top land part is always present. The volume of the top land part is proportional to the first order of the bore, and the combustion chamber volume is proportional to the second order of the bore, so unburned loss becomes more noticeable in engines with a smaller bore. Reduction of the top land part is considered necessary to reduce unburned loss.

This study used a top land height of 3.5 mm, which is the minimum value for which Honda has mass-production results.

The squish height is the part between the piston and the head at the combustion chamber periphery as shown in Fig. 18. This part has a thin height of around 1 mm, making it a challenge for flame to propagate.

For enhancement it was thought necessary to increase the squish height to promote flame propagation, and to minimize the top land height to reduce the unburned area.

This study used a squish height of 0.8 mm, which is the minimum value for which Honda has mass-production results.

3.3.2. Test results

Figure 19 shows the combustion test results for different top land heights. The test was performed under conditions of engine speed = 2000 rpm and IMEP = 810 kPa.

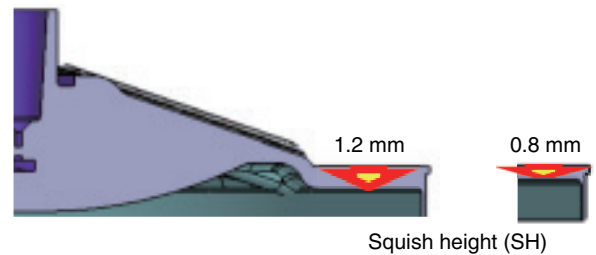


Fig. 18 Definition of squish height

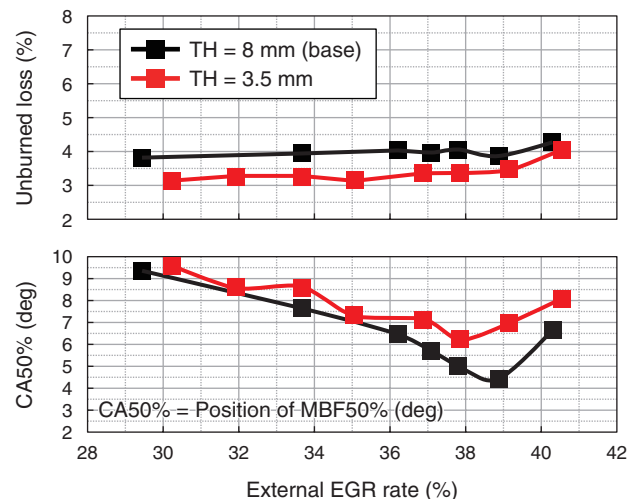


Fig. 19 Unburned loss reduction by piston shape

Unburned loss tended to increase together with the external EGR rate for both pistons. The piston shape with a lower top land height reduced unburned loss at any external EGR rate. However, pistons with a reduced top land height also exhibited a delayed ignition angle due to knocking. The reason for this is thought to be that more of the unburned HC was able to be combusted.

Figure 20 shows the combustion test results for different squish heights. The test was performed under conditions of engine speed = 2000 rpm and IMEP = 810 kPa.

These results show that flame propagation can be enhanced by setting the squish height to 2 mm or more, and that the unburned area can be reduced by setting the squish height to 1 mm or less, both of which enable unburned loss to be reduced. A delay in the ignition timing angle due to knocking was seen in the same manner as when reducing the top land height.

In order to increase the squish height, it is necessary to secure volume at the combustion chamber periphery. This poses a restriction on realizing the combustion chamber concept of securing volume near the plug as described in sections 3.1 and 3.2.

There is no difference in effect when expanding or shrinking the squish part, so it is necessary to reduce the squish height from the viewpoints of EGR limit and combustion period.

3.4. Brake Thermal Efficiency Enhancement Effects of Changing the Combustion Chamber Shape

The efficiency enhancement effects of changing the combustion chamber as described in sections 3.1 to 3.3 were confirmed. Table 3 shows the specifications of the final piston shape, and Fig. 21 shows the final combustion chamber shape. The top land height and squish clearance were set to the minimum values for which there are results

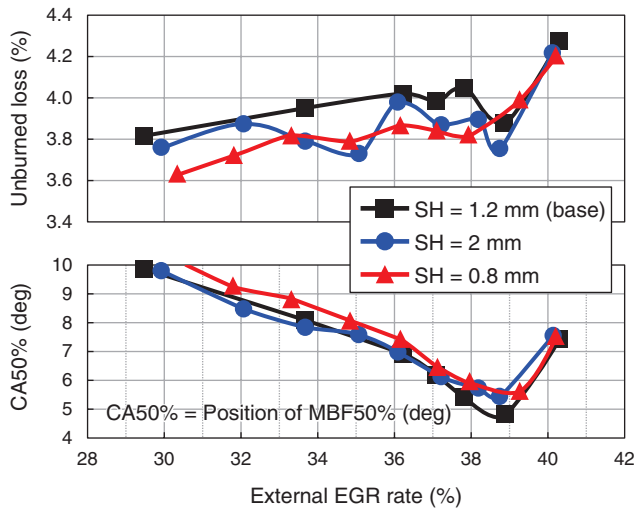


Fig. 20 Unburned loss and knock impact of squish height

in a mass-production engine.

Figure 22 shows the efficiency enhancement effects obtained by this combustion chamber shape. These test results show the effects of optimizing the direct injector spray in addition to the effects of the combustion chamber shape introduced in this paper. The test was performed under conditions of engine speed = 2000 rpm and IMEP = 810 kPa.

Use of a shallow-dish piston and modification of the direct injection spray enhanced the EGR limit and reduced the combustion period, which enhanced efficiency by 0.6 pt compared to the base piston. In addition, reduction of unburned loss by reducing both the top land height and the squish clearance enhanced brake thermal efficiency by 0.6 pt. These measures achieved enhancement of 1.2 pt compared to the base flat piston.

Lastly, to clarify the positioning of brake thermal efficiency in this study, Fig. 23 shows the load characteristics for a shallow-dish piston (TH = 3.5 mm/SH = 0.8 mm) and optimized injector. The external EGR rate was set to maximize the brake thermal efficiency at each load.

As a result, an external EGR rate of 38% was introduced and achieved a maximum brake thermal efficiency of 45.2%.

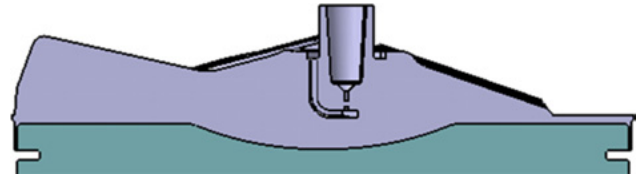


Fig. 21 Final shape of combustion chamber

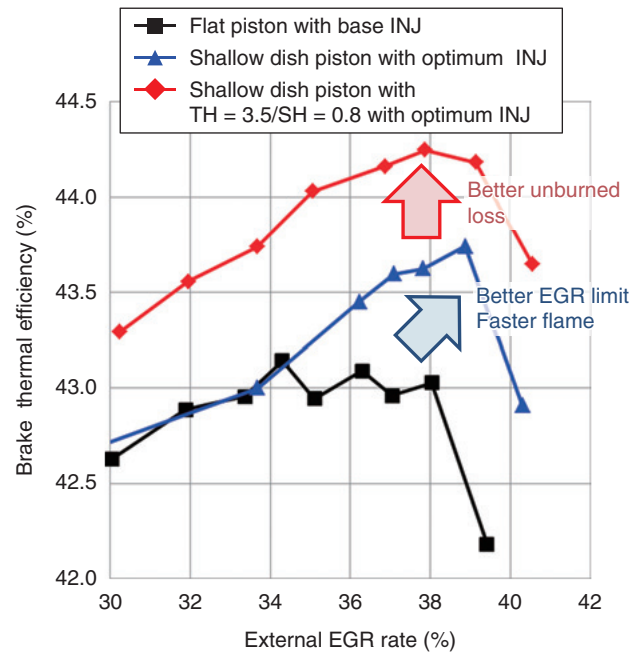


Fig. 22 Brake thermal efficiency due to combustion chamber optimization

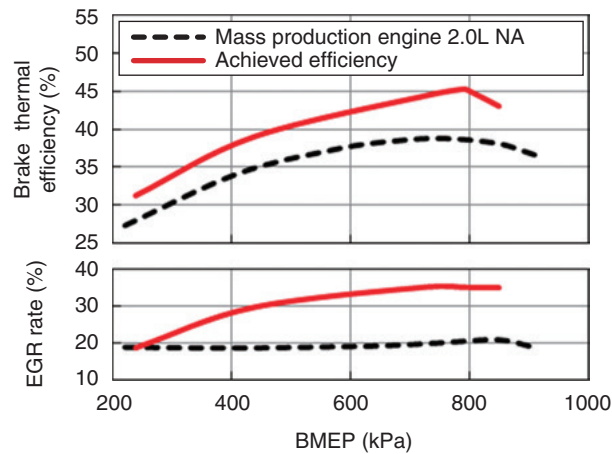


Fig. 23 Final achieved brake thermal efficiency

4. Conclusion

An effort was made to enhance combustion by optimizing the combustion chamber to enhance the brake thermal efficiency at the stoichiometric air-fuel ratio of a direct injection Atkinson engine with a high compression ratio and a long stroke. The following results were obtained.

(1) Delaying the timing angle at which the initial flame contacts the piston and stabilizing initial flame behavior were found to effectively enhance ignitability when the air-fuel mixture is highly diluted by EGR. Ignitability was enhanced and a higher EGR limit was obtained by setting the piston top surface distance from the plug to 6 mm or more and by changing the piston top shape to a shallow-dish shape with the center directly below the plug.

(2) Promotion of turbulence strength by maintaining the flow during the compression stroke, flame propagation from the center of the combustion chamber, and reducing the combustion chamber size to reduce the flame propagation distance were found to effectively shorten the combustion period.

A shallow-dish piston shape was used to maintain tumble during the compression stroke, dead volume such as squish was reduced around the combustion chamber periphery, and the combustion chamber volume was collected around the plug. In addition, the piston top shape used a shallow-dish shape with the center directly below the plug so that the flame grows from the center of the combustion chamber. This resulted in reduced time loss.

(3) Minimizing the volume of the combustion chamber end-part where the flame does not propagate was found to effectively reduce unburned loss.

In particular, minimizing both the top land height and squish height within the ranges that achieve the required durability enabled reduction of unburned loss.

The direct injector spray shape was optimized for the combustion chamber described above. As a result, a brake thermal efficiency of 45.2% was achieved in a direct injection Atkinson engine with a high compression ratio and a long stroke.

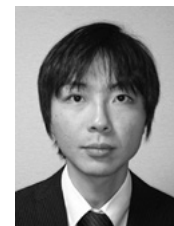
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