

Numerical Study on Enhancing Power and Thermal Efficiency of Large Motorcycle Gasoline Engine Using Pre-chamber Jet Combustion Simulated by Combustion CFD

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ABSTRACT

A study of combustion methods was conducted using 3D combustion simulation with the aim of enhancing power at full load and thermal efficiency at partial load for a big-bore spark ignition gasoline engine for large motorcycles. The effect of passive pre-chamber jet combustion on power was confirmed at full load. It was further confirmed that a jet sprayed from the pre-chamber into the main chamber caused an increase in turbulent kinetic energy that sped up combustion, and that increasing the compression ratio from 10.1 to 12.1 resulted in an enhancement of 3.6% in indicated work at full load compared to conventional SI combustion. At partial load, the effect of pre-chamber jet 2plug combustion, achieved by ignition in the pre-chamber after the ignition of the main-chamber side-plug with the aim of enhancing combustion stability, was confirmed. The jet sprayed from the pre-chamber facilitates ignition by coming into contact with the flame surface generated by the earlier ignition of the main-chamber side-plug. It was confirmed that this resulted in an increase in indicated thermal efficiency of 1.7 points at partial load compared to conventional SI combustion.

1. Introduction

In order to control global warming, under a scenario in which increases in temperature caused by climate change are kept below 1.5°C, it will be necessary⁽¹⁾ to reduce emissions of the greenhouse gas CO₂ by at least 45% from 2010 levels by 2030. The transportation sector accounts for approximately 20% of total domestic CO₂ emissions, and although motorcycles account for only 0.4% of that, this amounts to CO₂ emissions of 780,000 tons⁽²⁾, and it is important that enhancements are also made to internal combustion engines for motorcycles in order to reduce CO₂.

In addition to the demand to enhance riding performance by increasing maximum power at full load, spark ignition (SI) gasoline engines for large motorcycles are also being required to reduce CO₂ by improving fuel economy through enhancements in thermal efficiency at partial load. Here, the relationship between maximum engine power and fuel economy for the mass-produced motorcycle models of Honda is presented. Figure 1 shows the relationship between maximum engine power and fuel economy in Worldwide-harmonized Motorcycle Test Cycle (WMTC)

mode. The graph plots data for motorcycles in WMTC mode class 3-2, which applies to those models with a top speed of at least 140 km/h. As shown by the graph, maximum power and mode fuel economy are in conflict with each other. Increases in volumetric efficiency and higher engine speed are generally used to enhance maximum power. Increasing volumetric efficiency requires intake and exhaust resistance to be lowered, and it is essential to increase the diameter of the intake and exhaust valves, and to increase the bore diameter to permit this. Since increasing engine speed raises piston speed, it is important to shorten the stroke to keep piston speed under control. Accordingly, the tendency for an engine for which the aim is to enhance maximum engine power is to specify a big bore and a short stroke. However, while specifying a big bore and a short stroke is effective in increasing maximum power, at partial load this results in noticeable reductions in both intake air speed and tumble flows formed in the cylinder. This leads to insufficient air-fuel mixture formation and reduced turbulence at combustion timing, resulting in reductions in combustion speed. These factors cause a lowering in thermal efficiency at partial load. This means that enhancing both maximum

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power and partial-load thermal efficiency in a big-bore, short-stroke engine requires increases in combustion speed to be achieved by techniques other than in-cylinder flow.

The relationship between displacement per cylinder and bore for motorcycle models currently in production at Honda is shown in Fig. 2. As shown in the graph, this research targeted motorcycle models currently in production with a bore diameter of 92 mm, which are classified as having big-bore engines. These models had achieved⁽³⁾ increases in combustion speed by performing ignition at two plugs, one at the side and one in the center of each cylinder, and this research sought to realize further increases in combustion speed.

A passive pre-chamber was used in prior research examples^{(4), (5)} with the aim of increasing combustion speed in SI engines for large motorcycles. A passive pre-chamber is one that is not equipped with a fuel injection nozzle within the pre-chamber and that draws in the air-fuel mixture from the main-chamber side. In the examples of prior research^{(4), (5)}, increases in full-load output power were achieved, but there were no reports of enhancements in partial-load thermal efficiency. In addition, although those research studies used 3D combustion simulation (hereinafter, “combustion CFD”),

performance evaluations did not take into account the knocking that can be an issue with actual engines.

In this research, a combustion CFD was built for SI engines used in large motorcycles to enable knocking detection through a detailed elementary reaction. Furthermore, combustion CFD was used to analyze the combustion mechanism of flame propagation combustion for a variety of approaches, such as conventional SI combustion, pre-chamber jet combustion, and pre-chamber jet 2plug combustion. Performance evaluations of maximum power and partial-load thermal efficiency were also conducted. This enabled the determination of combustion methods that simultaneously enable enhanced maximum power at full load and higher partial-load thermal efficiency for the reduction of CO₂ in conventional SI engines for large motorcycles, and the results are discussed in this paper.

2. Issues with Large Motorcycle SI Combustion and Increases in Combustion Speed

When seeking to enhance thermal efficiency in an SI engine, it is generally thought necessary to increase theoretical efficiency by raising the compression ratio. Because this facilitates the occurrence of knocking, the ignition timing needs to be retarded more than the Minimum advanced for Best Torque (MBT), which increases time loss. Accordingly, to increase thermal efficiency, it is important to mitigate knocking and reduce time loss.

Knocking is closely related to auto-ignition of end gases⁽⁶⁾, and because the distance to the bore-end gas portion is particularly large in big-bore SI engines, knocking is easy to induce. To mitigate knocking, it is necessary to complete combustion before auto-ignition occurs, and raising combustion speed is an effective means of achieving this^{(7), (8)}. A summary of flame surface behavior in flame propagation combustion and mechanisms related to combustion speed is shown here in Fig. 3. According to the concept of wrinkled laminar flame proposed by Damköhler et al.⁽⁹⁾⁻⁽¹¹⁾, combustion speed is closely related to the surface area of the wrinkled laminar flame, which is amplified by turbulence. Consequently, Turbulent Kinetic Energy (TKE) within the combustion chamber needs to be increased to increase combustion speed. One method of increasing combustion speed in SI engines that has been attracting attention is pre-chamber jet combustion, in which a jet sprayed from the pre-chamber can amplify TKE on the main-chamber flame surface and increase the wrinkled laminar flame surface area to speed up combustion. This has the result of mitigating knocking, and enabling the compression ratio to be raised and thermal efficiency to be increased^{(7), (8)}.

In this research, the effect of pre-chamber jet combustion^{(7), (8)} using a passive pre-chamber was confirmed. In the passive type of pre-chamber jet combustion, fuel is brought into the pre-chamber by drawing in air-fuel mixture from the main chamber, which means that the pre-chamber

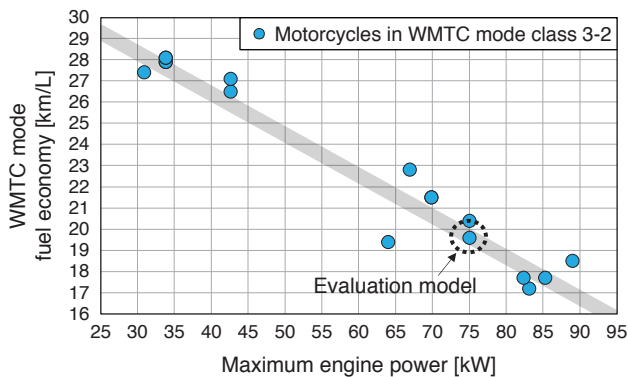


Fig. 1 Maximum engine power and fuel economy of motorcycles in WMTC mode class 3-2

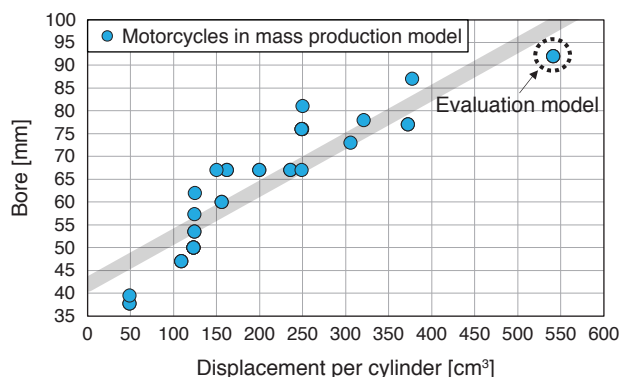


Fig. 2 Displacement per cylinder and bore of motorcycles in mass production model

can easily run short of fuel^{(12), (13)}. Moreover, because the temperature falls when burned gases in the pre-chamber pass through the hole, this creates issues for combustion stability at partial load in particular⁽¹²⁾. For this reason, it is challenging to guarantee enhanced performance over a wide operating range from full load to partial load with a single combustion specification⁽¹³⁾, creating an issue for the popularization of pre-chamber jet combustion^{(12), (13)}.

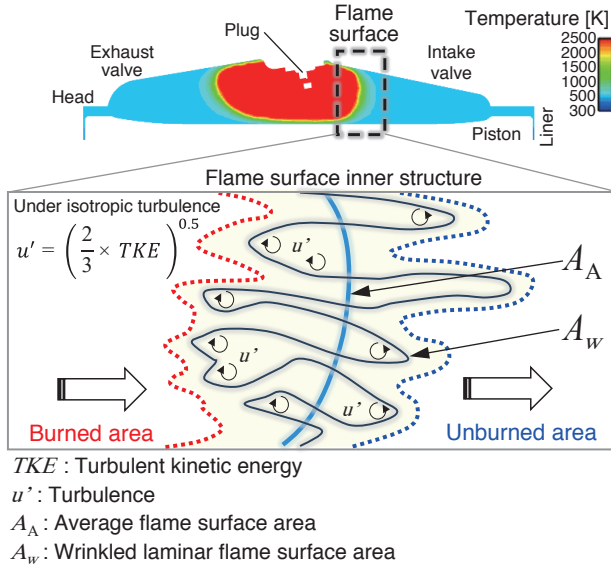


Fig. 3 Schematic diagram of wrinkled laminar flame

3. Combustion Analysis Techniques

3.1. Evaluation Load and Specifications

Figure 4 shows full-load power characteristics for the target models, with operating points in WMTC mode class 3-2 plotted by engine speed and power, and the bubble size indicating the frequency of occurrence.

Figure 4 also shows the evaluation load point conditions for this research. For the evaluation point for maximum power at full load, engine speed was set to 7500 rpm, throttle opening to 100%, and excess air ratio to 0.85. For the evaluation point for thermal efficiency at partial load, engine speed was set to 4000 rpm, which is frequently used in WMTC mode, throttle opening to 26%, and excess air ratio to 1.0.

Figure 5 shows schematic diagrams of the combustion chamber and specifications of the engine used for analysis with the aim of increasing combustion speed. Figure 5(a) shows the parallel twin SI engine for large motorcycles (hereinafter, “conventional SI combustion”) that forms the basis of this study. This engine has two plugs, center and side, for each cylinder, providing simultaneous ignition as well as phased ignition timing control⁽³⁾. Figure 5(b) is based on conventional SI combustion but specified with a passive pre-chamber to replace the center plug, with

ignition performed only in the pre-chamber (pre-chamber jet combustion). Figure 5(c) shows a pre-chamber jet combustion specification with an additional side plug between the exhaust valves, with ignition performed both in the pre-chamber and by the side plug (pre-chamber jet 2plug combustion). The pre-chamber specifications used in this research and shown in Fig. 5(d) refer to those obtained by the authors in previous research^{(7), (8)} that had the effect of enhancing thermal efficiency through knocking mitigation, and were set to the same values of 10 holes and a hole diameter of 0.5 mm. On the other hand, because the bore diameter was increased from 73 mm in the previous research to 92 mm for this research, the volume was increased from 0.5 cm³ to 0.8 cm³ to increase the reach of the jet.

3.2. Computational Techniques and Reproducibility in Actual Engine

The combustion CFD software used in this research was Convergent Science’s CONVERGE v3.0.19, and an overview of the model is provided in Table 1. The flame propagation combustion model used in this research is a detailed chemical reaction (SAGE)⁽¹⁴⁾, which reproduces combustion through the interaction of detailed elementary reactions and turbulent diffusion. While this has a high calculation load, because the direct combustion chemical reaction is solved using an elementary reaction, it is likely that the dependence on the model coefficients can be reduced even in cases where the specifications and forms of combustion are changed. Furthermore, the use of an elementary reaction calculation enables the reproduction of knocking caused by auto-ignition phenomena driven by low-temperature oxidation reactions, which has been reported in examples of previous research in which it was applied to SI combustion and pre-chamber combustion^{(4), (15), (16)}. In addition, an SIP-Gr2.0-S5R^{(17), (18)} model that has been confirmed to reproduce regular gasoline in elementary reactions with high accuracy was used for the fuel elementary reaction formula.

Using this combustion model, the combustion CFD was compared to an actual engine. Figure 6 shows a comparison of combustion behavior between the combustion CFD and

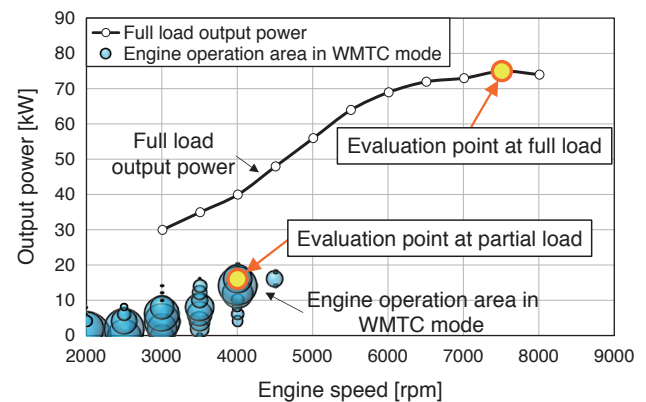


Fig. 4 Schematic diagram of evaluation load conditions

the actual engine, for conventional SI combustion under test specifications at the evaluation point for maximum power at full load and at the evaluation point for thermal efficiency at partial load. For combustion CFD using identical model coefficients, it was confirmed that in-cylinder pressure and heat release rate under full and partial load were within the range of cyclic variation in an actual engine.

3.3. Knocking Detection Techniques and Flame Surface Analysis Techniques

During the analysis of a variety of flame propagation combustion types, techniques using combustion CFD for knocking detection were studied. In previous research, it was concluded that knocking detection can be performed^{(19), (20)} by using the absolute minimum derivative values for heat release rate shown in equation (1) to detect sharp declines in heat release rate caused by auto-ignition. In this research, the value in equation (1) is referred to as the Knocking Detection Index (KDI) and knocking detection was verified in combustion CFD.

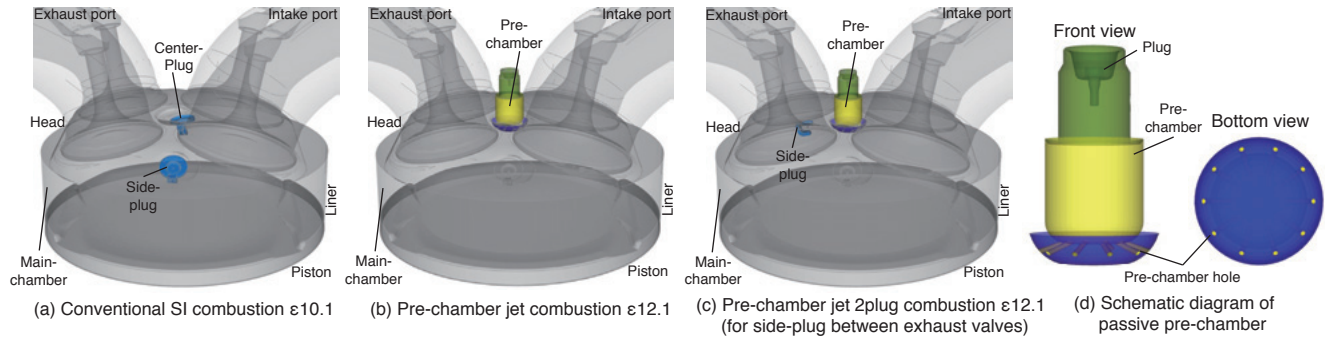
$$\text{Knocking detection index (KDI)} = \left| \min \frac{d}{d\theta} \left(\frac{dQ}{d\theta} \right) \right| \quad (1)$$

Here, θ is the crank angle [deg.], and Q is heat release [J].

Table 1 Calculation software and models

CFD software	CONVERGE v3.0.19
Turbulence model	RANS ζ -f
Spray model	Rosin-Rammler and Rayleigh-Taylor
Combustion model	SAGE detailed chemical reaction
Ignition model	Add energy source to ignition point
Fuel chemical reaction mechanism	Regular gasoline SIP-Gr2.0-S5R
Mesh size [mm]	0.125-1.0 (adaptive mesh refinement)
Model parameter	Turbulent schmidt number: 0.5 Reaction multiplier: 1.15

Figure 7 shows combustion CFD heat release rates and derivative values for heat release rates under conditions in which auto-ignition occurs for conventional SI combustion at the evaluation point for maximum power at full load. At the time that auto-ignition occurred, it was observed that the heat release rate fell back after first rising, and the derivative values of the heat release rate fell sharply immediately after auto-ignition compared to before auto-ignition. Furthermore, Fig. 8 shows KDI values when changes were made to ignition timing. When combustion center of gravity timing (Mass Fraction Burned 50%: MFB 50%) was advanced gradually from 20 deg. ATDC, the KDI



	(a) Conventional SI combustion ε10.1	(b) Pre-chamber jet combustion ε12.1	(c) Pre-chamber jet 2plug combustion ε12.1 (for side-plug between exhaust valves)
Displacement	1082 [cm³]		
Number of cylinders	2		
Bore	92 [mm]		
Stroke	81.4 [mm]		
Compression ratio: ε	10.1	12.1	
Maximum power	75 [kW] / 7500 [rpm] (base)		
Valve system	SOHC 4-valve		
Fuel injection system	Port injection (injection pressure: 343 [kPa])		
Fuel	Japanese regular gasoline (research octane number: 91)		
Ignition system	2plug: Center-plug: conventional Side-plug: conventional	1plug: Center-plug: pre-chamber Side-plug: no-ignition	2plug: Center-plug: pre-chamber Side-plug: conventional (side-plug placement between exhaust valves)
(d) Pre-chamber specifications	-	Hole number: 10	
		Hole diameter: 0.5 [mm]	
		Volume: 0.8 [cm³]	

Fig. 5 Schematic diagrams and specifications of engine

value increased at a constant slope, with the advanced side sloping from MFB 50% at 11 deg. ATDC. This shows that from MFB 50% at 11 deg. ATDC, a spike in heat release occurs due to auto-ignition combustion on the advanced side and the KDI value increases. This technique is interpreted

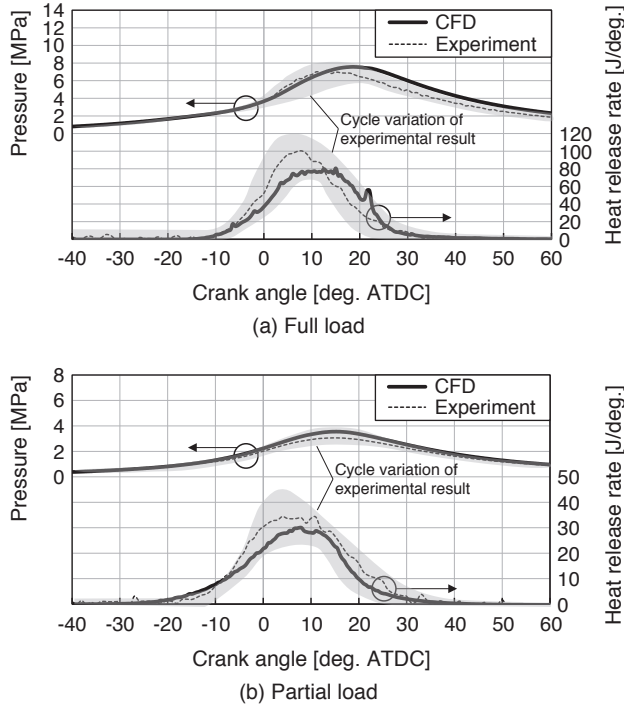


Fig. 6 Comparison of in-cylinder pressure and heat release rate between experiment and CFD

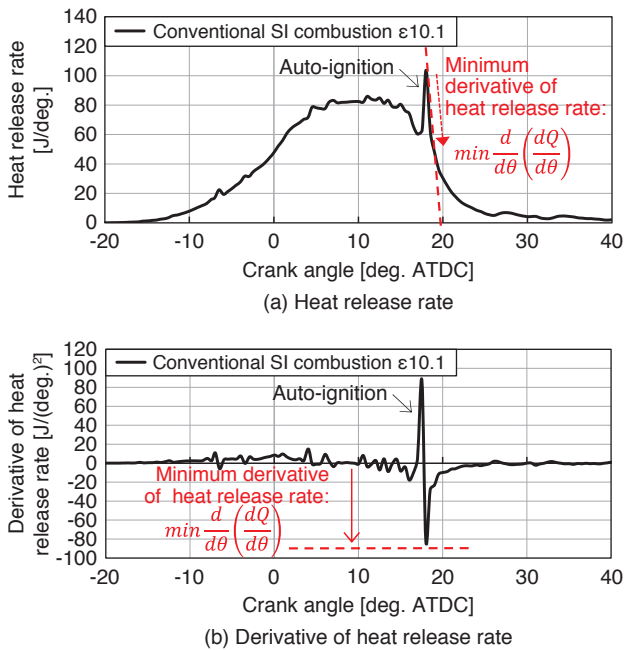


Fig. 7 Heat release rate and its derivative under auto-ignition condition

as resulting in clear changes in combustion behavior before and after auto-ignition and, as shown in Fig. 8, calculating the intersection enables detection of knocking limit MFB 50%. From this point, these techniques were applied after taking knocking into account, and performance evaluations were conducted for each specification.

In addition, when analyzing flame surface behavior, because it has the characteristic of peaking at the flame front of excessive oxidants of HO_2 radicals and the like, techniques in which it is used as a marker of flame surface position have been proposed^{(15), (21)}. In those research studies, the flame surface position was observed using a similar technique.

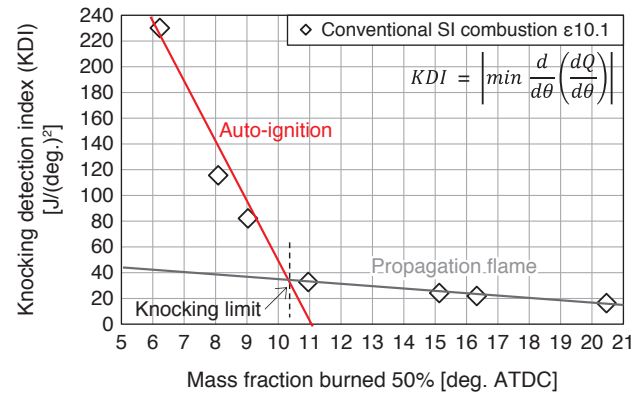


Fig. 8 Knocking judgement using knocking detection index

4. Results of Combustion CFD Analysis at Full-load Power

The specifications used for analysis at full-load power were the conventional SI combustion of Fig. 5(a) and the pre-chamber jet combustion of Fig. 5(b). In previous research by the authors, pre-chamber jet combustion led to increased combustion speed that had the effect of mitigating knocking, enabling an increase in compression ratio of 1.8, and achieving enhancements in thermal efficiency^{(7), (8)}. This research references those results, specifying that the compression ratio was increased from 10.1 in conventional SI combustion to 12.1. In addition, for conventional SI combustion, the phased ignition of the side plug from the center plug was set to 4 deg., the same as for a mass-produced model.

Figure 9 shows the effects of changes in ignition timing on knocking detection using KDI values under the respective specifications. For both conventional SI combustion and pre-chamber jet combustion, because advances in MFB 50% timing resulted in auto-ignition occurring and significant changes in the KDI, this was used to perform knocking detection. For pre-chamber jet combustion, the slope of changes in the KDI value was

significant, and the knocking limit MFB 50% timing was retarded by 2.4 deg.

Figure 10 shows the results of analysis of combustion duration and indicated work based on changes in ignition timing, with MFB 50% timing on the horizontal axis. The conditions under which maximum indicated work was

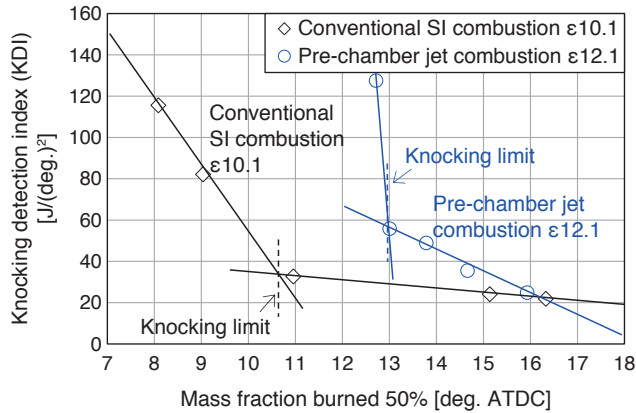


Fig. 9 Knocking limit comparison using knocking detection index

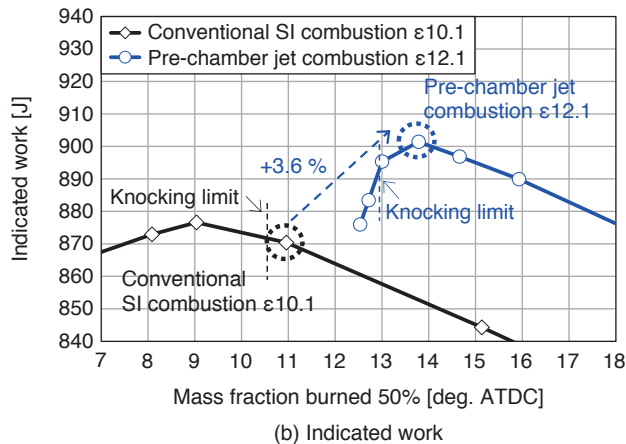
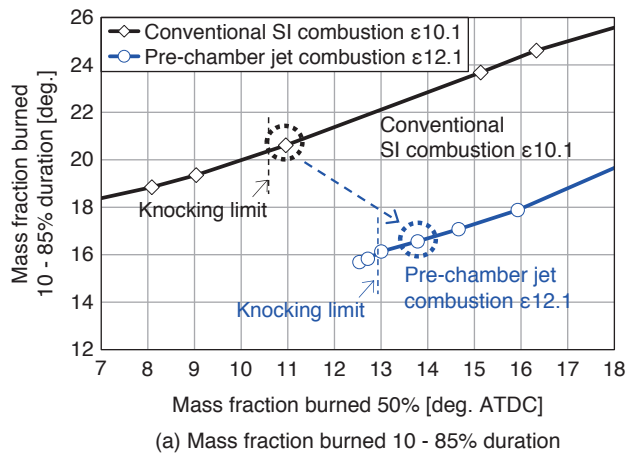


Fig. 10 Combustion duration and indicated work

obtained are shown by the line with circle markers. For this research, combustion duration was defined as combustion duration required for Mass Fraction Burned to progress from 10% to 85%.

In Fig. 10(a), when compared at the same MFB 50% timing of 14 deg. ATDC, combustion duration for pre-chamber jet combustion was shortened by 6.2 deg. Furthermore, in Fig. 10(b), indicated work for pre-chamber jet combustion increased by 5.9% when compared at the same MFB 50% timing of 14 deg. ATDC. In addition, for conventional SI combustion, advancing ignition timing to the knocking limit obtained the maximum indicated work. By comparison, pre-chamber jet combustion maximum indicated work was obtained before ignition timing at the knocking limit. When compared under the conditions at which maximum indicated work was obtained, not only was combustion duration for pre-chamber jet combustion shortened by 4.1 deg., but higher theoretical thermal efficiency caused by increases in the compression ratio also achieved enhancements in indicated work at full load of 3.6%.

Figure 11 shows in-cylinder pressure and heat release rate under the conditions at which maximum indicated work was obtained. The graph confirms that, compared to conventional SI combustion, pre-chamber jet combustion has high heat release rates and rapid combustion at or after TDC. In addition, it was confirmed that, as shown in Fig. 12 by the temperature iso-surface around adiabatic flame temperature and the cross-sections of the HO₂ mass fraction, although for conventional SI combustion the flame surface advances gradually on the surface of a sphere centered on the spark plug, in pre-chamber jet combustion the jet sprayed into the main chamber causes the flame surface to spread outwards.

Furthermore, Fig. 13 shows average in-cylinder TKE behavior. In pre-chamber jet combustion, the sprayed jet increased turbulent kinetic energy in the main chamber. This resulted in rapid flame propagation combustion, enabling an increase in indicated work at full load.

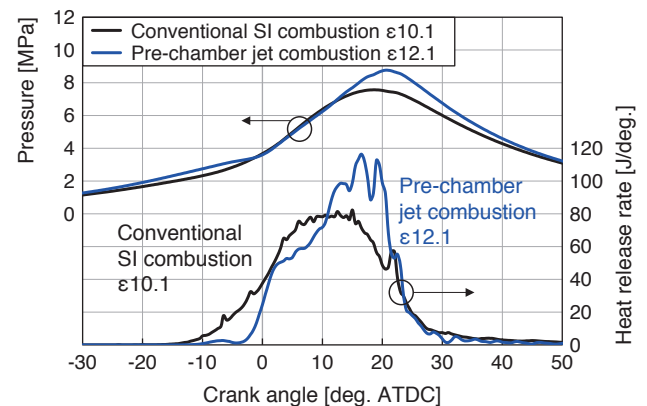


Fig. 11 In-cylinder pressure and heat release rate

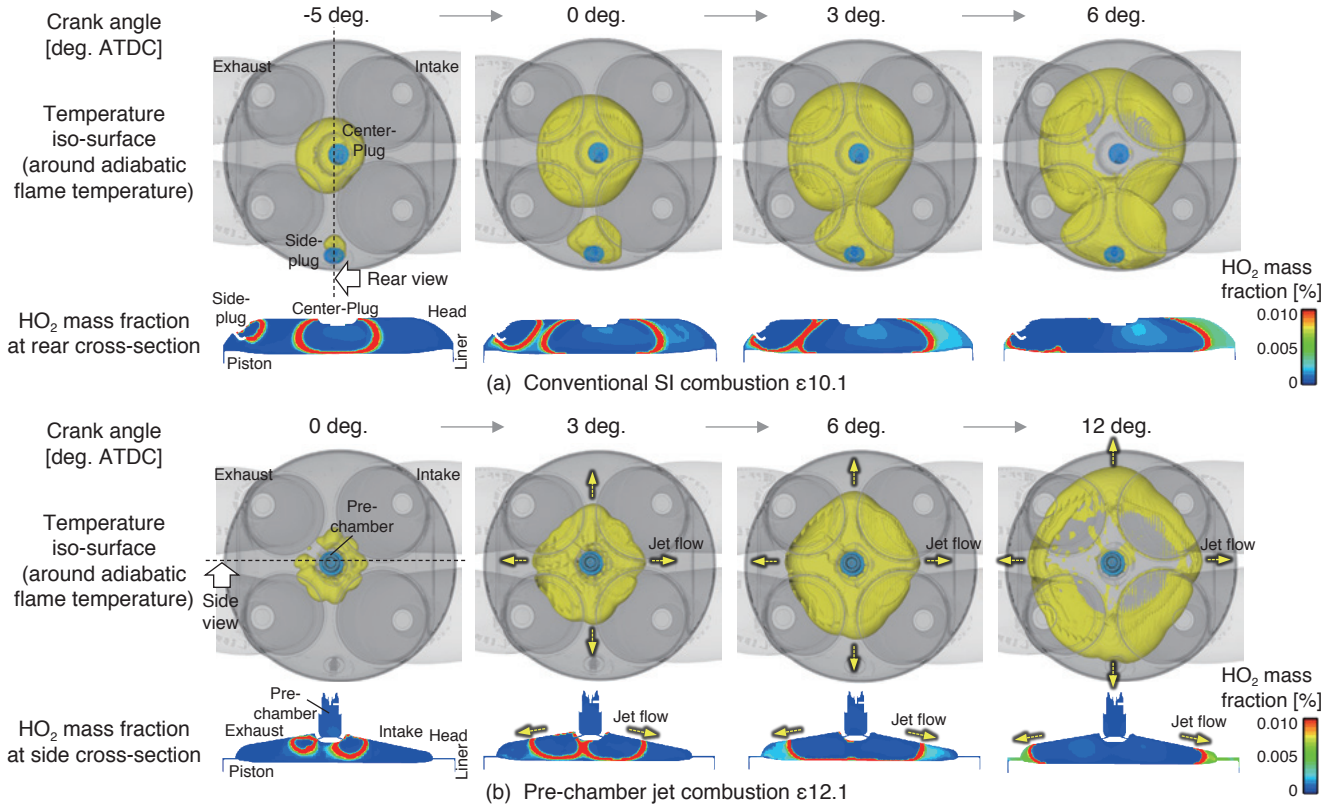


Fig. 12 Flame surface behavior

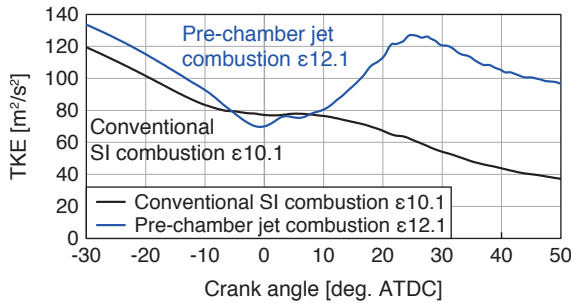


Fig. 13 In-cylinder turbulent kinetic energy

5. Results of CFD Analysis of Combustion at Partial Load Point

5.1. Pre-chamber jet combustion

Figure 14 shows the behavior of in-cylinder pressure and heat release rate when ignition timing is changed during pre-chamber jet combustion using ignition only in the pre-chamber, comprising the specification shown in Fig. 5(b). The graph confirms that for pre-chamber jet combustion at partial load, heat release rates were insignificant immediately after ignition for all ignition timings, and misfires subsequently occurred.

Here, the factors leading to misfires for pre-chamber

jet combustion are analyzed. Figure 15 shows residual gas rates and their distribution in the pre-chamber directly before ignition, both under full-load conditions where ignition was possible and under partial-load conditions that led to misfires. As shown in Fig. 15(a), under partial-load conditions, residual gas in the pre-chamber increased from 12.8% to 23.5%. As shown in Fig. 15(b), residual CO_2 increased throughout the pre-chamber. Regarding the cause, it is thought that because intake flow rates and flows are reduced under partial-load conditions, this facilitates increases in residual gas in the main chamber, and also results in insufficient pre-chamber scavenging.

Furthermore, Fig. 16 shows gas temperatures at the inlets and outlets of the pre-chamber holes, and in addition to the previously mentioned reductions in combustion temperature caused by increases in residual gas under partial-load conditions, reductions in temperature caused by heat losses when passing through the holes also increased. Accordingly, while the maximum temperature for the sprayed jet under full-load conditions was 2480 K, under partial-load conditions it fell to 2170 K.

In addition, Fig. 17 compares main-chamber gas density and the distance reached by the tip of the jet originating in the hole outlet (hereinafter, “penetration”). Penetration was taken to be the distance reached by HO_2 . As shown by Fig. 17(a), density in the main chamber at the time of jet spray, which occurred at -5 deg. ATDC, was 14.8 kg/m^3

under full-load conditions, while this decreased to 6.9 kg/m^3 under partial-load conditions of low volumetric efficiency. As can be seen from Fig. 17(b), it was confirmed that initial penetration was greater under partial-load conditions than it was under full-load conditions. This is thought to be due to the low gas density of the main chamber facilitating the jet spray.

Additionally, Fig. 18 shows in-cylinder temperature distributions after jet spray. Under full-load conditions, initial penetration was small, with high-temperature gas tending to accumulate in the area around the hole outlet, leading to ignition. Conversely, under partial-

load conditions, the large initial penetration facilitated the blending of the jet with the air-fuel mixture in the main chamber immediately after jet spray, resulting in the temperature of the jet falling during the spreading of the jet through the main chamber, and in ignition not being achieved. In previous research that analyzed the factors leading to a decline in ignitability in pre-chamber jet combustion, it was shown that in the case of a strong jet, jet temperatures fall due to larger heat losses in the area of the holes⁽²²⁾. The phenomenon of the jet spray facilitating the drawing-in of main-chamber premix leading to a reduction in jet temperatures and to ignition instability was also confirmed⁽²²⁾. A similar phenomenon to that of previous research was also observed in this research, suggesting that reductions in ignitability were caused by these factors.

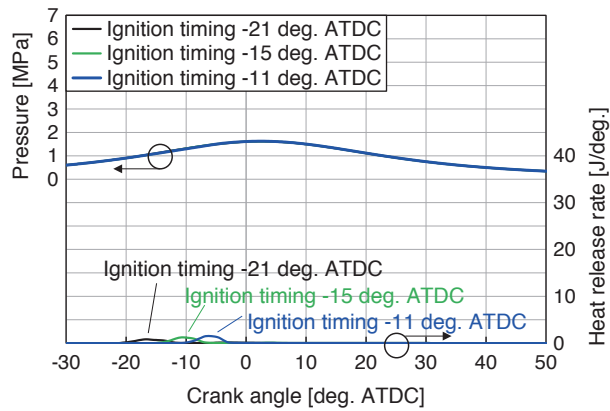


Fig. 14 In-cylinder pressure and heat release rate

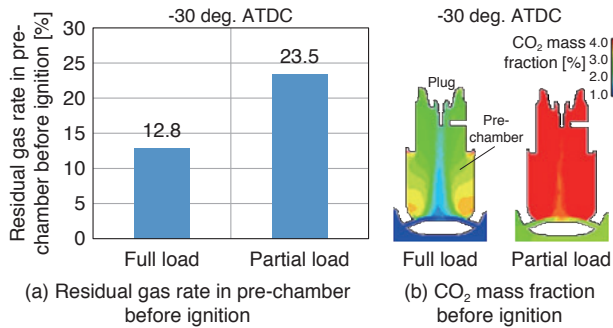


Fig. 15 Residual gas rate and distribution in pre-chamber

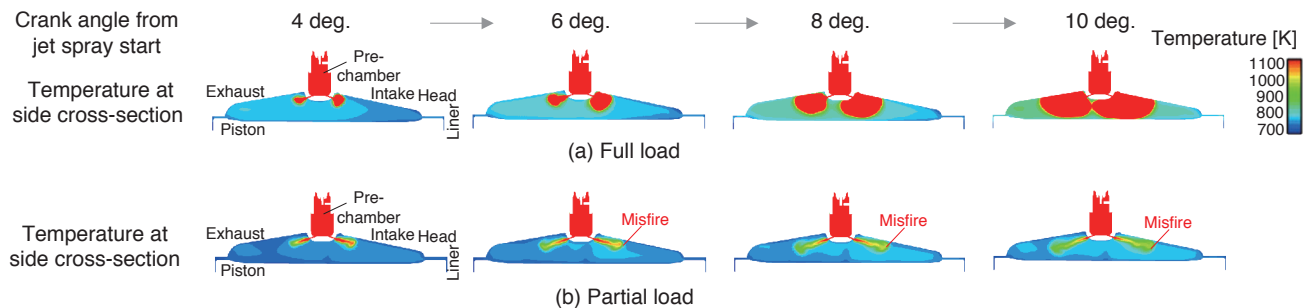


Fig. 18 Jet temperature behavior

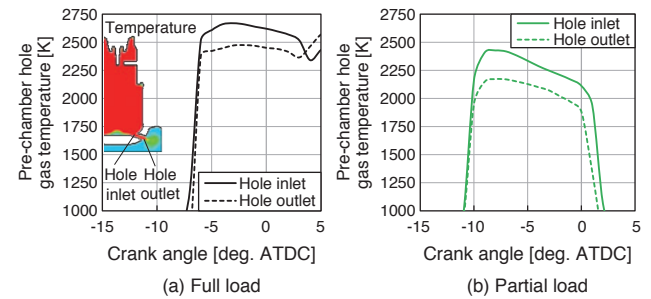


Fig. 16 Pre-chamber hole inlet and outlet gas temperature

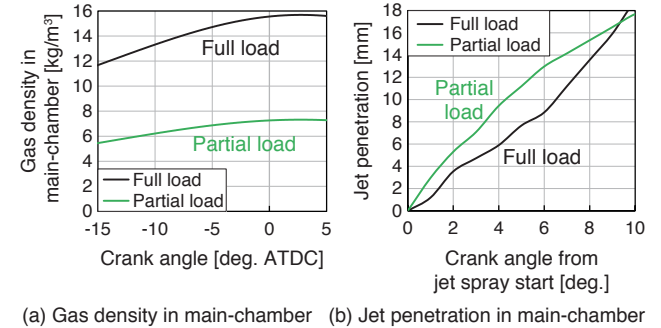


Fig. 17 Gas density and jet penetration in main-chamber

5.2. Pre-chamber Jet 2plug Combustion

In the previous section, it was clearly shown that, for partial loads, pre-chamber jet combustion solely by pre-chamber ignition under these specifications caused jet temperatures to decline and resulted in misfires. To address this, techniques to facilitate ignition by first igniting the main-chamber side-plug and using main-chamber combustion to raise the temperature of in-cylinder gas, then performing pre-chamber ignition and spraying the jet were studied.

As shown in Fig. 5(c), the specification for this study involved the installation of a side-plug in the pre-chamber jet combustion main chamber. Regarding the location of the main-chamber side-plug at the bore end, there were concerns that, due to the use of main-chamber-first ignition, the unburned area at the opposite end of the bore would experience increased heating time, inducing knocking.

Accordingly, the performance evaluation of pre-chamber jet 2plug combustion was performed after changing the location of the main-chamber side-plug to a position between the exhaust valves, towards the center of the bore.

For phased ignition, conventional SI combustion was set to simultaneous ignition using a center-plug and a side-plug in the same way as for mass-produced models. For pre-chamber jet 2plug combustion, advance studies using phased ignition as the parameter were conducted, and phased ignition was set with the aim of facilitating jet combustion by initiating the jet spray at the point at which the flame resulting from the earlier ignition of the main-chamber side-plug reached the area of the pre-chamber. For the load evaluation of the targeted engine, pre-chamber plug ignition was set to take place at 10 deg. after ignition of the main-chamber side-plug.

The results of knocking detection when changing ignition timing for conventional SI combustion and pre-chamber jet 2plug combustion are shown in Fig. 19. Because the advance of MFB 50% timing resulted in significant changes in the KDI value for both, this was used to perform knocking detection. For pre-chamber jet 2plug combustion, the slope of

changes in the KDI value was significant, and the knocking limit MFB 50% timing was retarded by 3.9 deg.

Figure 20 shows the results of performance evaluation at partial load point. Figures 20(a) and (b) show the results of analysis of combustion duration and indicated thermal efficiency, with MFB 50% timing on the horizontal axis. The conditions under which maximum indicated thermal efficiency was obtained for each specification are shown by the line with circle markers. When pre-chamber jet 2plug

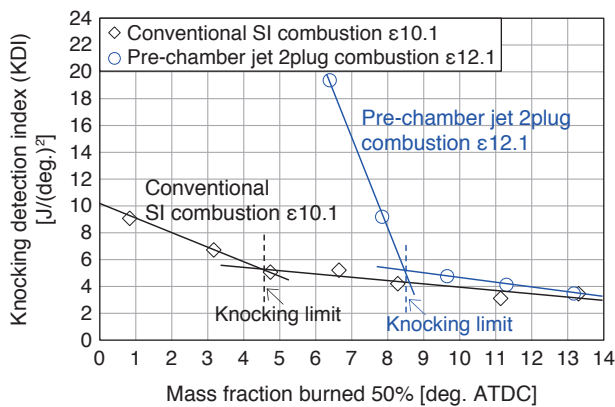
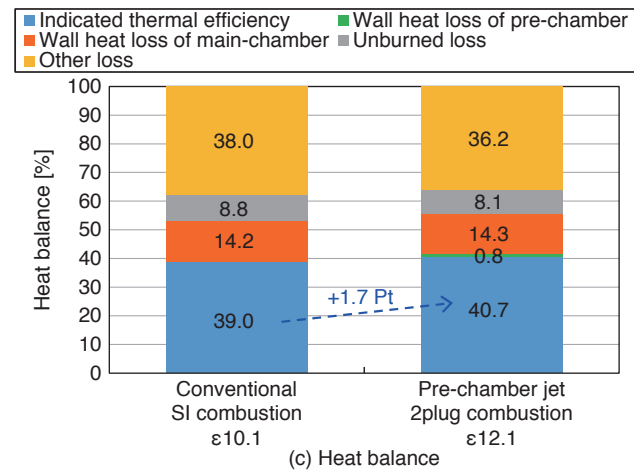
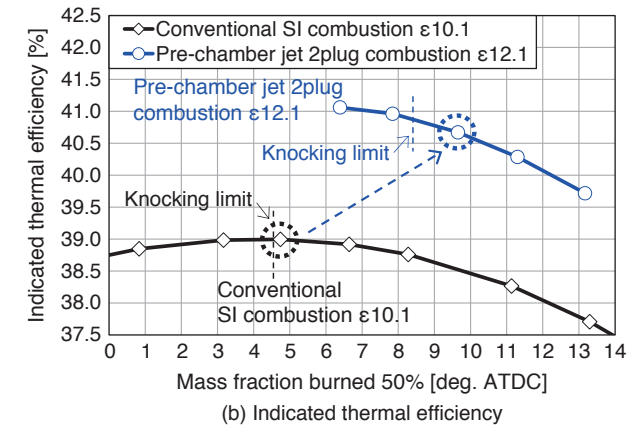
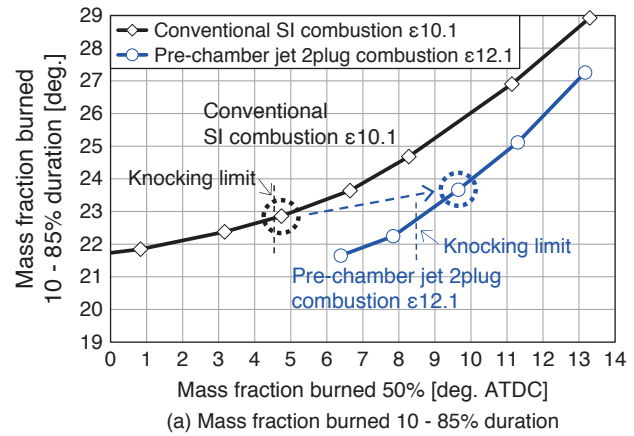


Fig. 19 Knocking limit comparison using knocking detection index

Fig. 20 Combustion duration, indicated thermal efficiency, and heat balance

combustion was compared to conventional SI combustion at the same MFB 50% timing of 10 deg. ATDC, it was confirmed that combustion duration was shortened by 2 deg. and indicated thermal efficiency was increased by 2.1 points.

Also, for the heat balance under the conditions in which maximum indicated thermal efficiency was obtained shown in Fig. 20(c), heat losses increased for pre-chamber jet 2plug combustion when compared to conventional SI combustion, but the effects of raising the compression ratio from 10.1 to 12.1 achieved a 1.7-point increase in indicated thermal efficiency.

Figure 21 shows the behavior of in-cylinder pressure and heat release rate under the conditions at which maximum indicated thermal efficiency was achieved. As shown in the graph, misfires do not occur in pre-chamber jet 2plug combustion, and it was confirmed that the ignition of the jet into the main chamber was followed by a sharp rise in heat release.

In addition, as shown in Fig. 22 by the adiabatic flame temperature and the cross-sections of the HO_2 mass fraction, it was confirmed that for partial loads in conventional SI combustion the flame surface advances on the surface of a sphere centered on the spark plug in the same way as for full loads. On the other hand, in pre-chamber jet 2plug combustion, it was confirmed that using pre-chamber ignition to spray the jet when flame propagation reaches the area of the pre-chamber following the earlier ignition

in the main chamber resulted in strong flame propagation originating in the jet.

Furthermore, with a focus on behavior up to the point at which the jet ignites in the main chamber, Fig. 23 shows the distribution of in-cylinder temperatures. The figure confirms that the mechanism driving ignition is that of the jet sprayed from the pre-chamber contacting the high-temperature flame surface caused by the earlier ignition of the main-chamber side-plug.

In addition, Fig. 24 shows average in-cylinder TKE behavior. In conventional SI, in-cylinder TKE declined

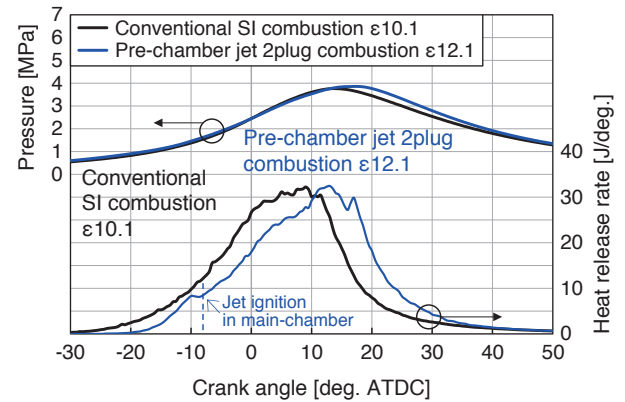


Fig. 21 In-cylinder pressure and heat release rate

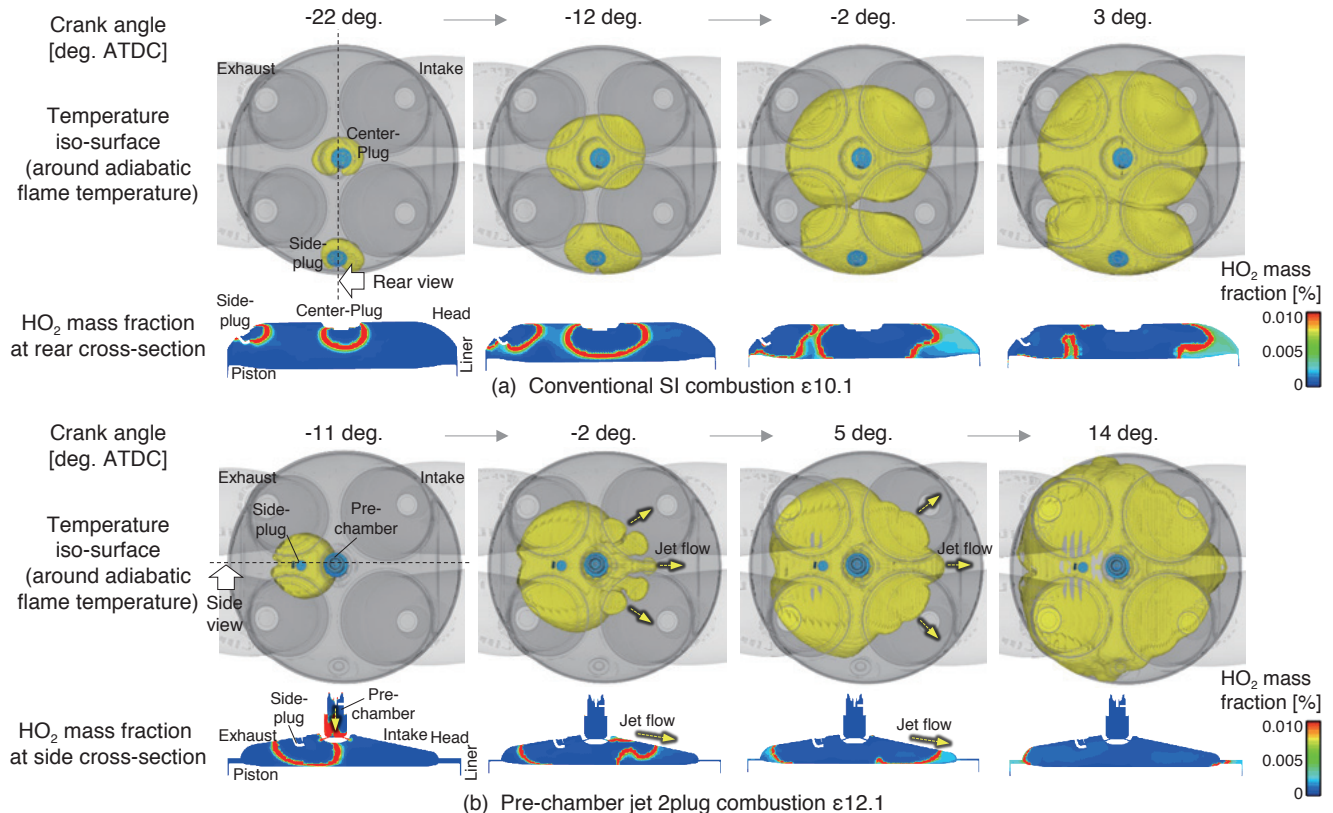


Fig. 22 Flame surface behavior

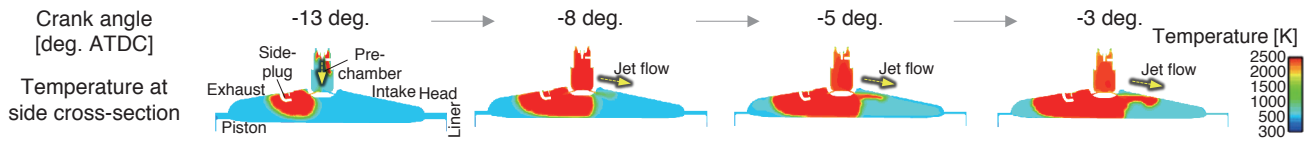


Fig. 23 Temperature behavior in pre-chamber jet 2plug combustion $\epsilon 12.1$

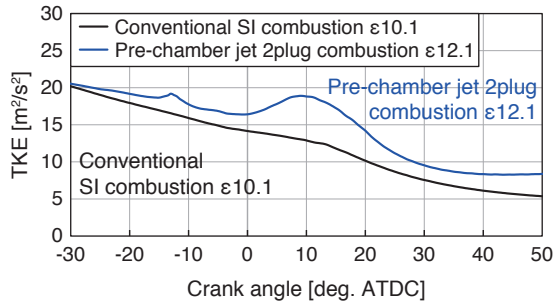


Fig. 24 In-cylinder turbulent kinetic energy

over time, while in pre-chamber jet 2plug combustion, it was confirmed that spraying the pre-chamber jet after main-chamber ignition led to higher in-cylinder TKE. This resulted in a sharp rise in heat release after the jet ignited, increasing indicated thermal efficiency.

6. Conclusion

A study of full-load maximum power and partial-load thermal efficiency in conventional SI engines for large motorcycles was conducted using combustion CFD incorporating knocking detection. The conclusions reached are as follows.

- (1) Using detailed chemical reaction models, a combustion CFD was built that incorporates knocking detection techniques that use absolute minimum derivative values for heat release rate. This enabled performance evaluations of maximum power and evaluation of thermal efficiency related to engine combustion specifications to be conducted as a simulation-based study.
- (2) At the evaluation point for maximum power at full load, in pre-chamber jet combustion, a jet sprayed from the pre-chamber increased turbulent kinetic energy in the main chamber and sped up combustion.
- (3) At the evaluation point for maximum power at full load, the use of pre-chamber jet combustion reduced combustion duration and mitigated knocking in comparison to conventional SI combustion. This enabled the compression ratio to be raised from 10.1 to 12.1, shortening combustion duration by 4.1 deg. and achieving an enhancement of 3.6% in indicated work at full load.

- (4) At the evaluation point for thermal efficiency at partial load, it was confirmed that pre-chamber jet combustion using ignition only in the pre-chamber helped prevent flame propagation from being maintained in the main chamber and resulted in misfires. This indicated that it was the result not only of combustion temperature declines caused by residual gas in the pre-chamber, and jet temperature declines caused by heat losses in the area of the holes, but also by temperature declines resulting from the spreading of the jet in the main chamber.
- (5) At the evaluation point for thermal efficiency at partial load, for the specification in which compression ratio was raised from 10.1 to 12.1, the effects of pre-chamber jet 2plug combustion were confirmed by relocating the main-chamber side-plug to a position between the exhaust valves, towards the center of the bore, and igniting the pre-chamber after main-chamber side-plug ignition. It was confirmed that ignition of the jet that is sprayed from the pre-chamber is made possible by contact with the flame surface caused by earlier ignition in the main chamber, and that it facilitates combustion in the main chamber.
- (6) At the evaluation point for thermal efficiency at partial load, it was confirmed that pre-chamber jet 2plug combustion resulted in an increase in indicated thermal efficiency of 1.7 points compared to conventional SI combustion.

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