

Technology for Enhancing Thermal Efficiency of Gasoline Engine by Pre-chamber Jet Combustion

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ABSTRACT

A two-point increase in brake thermal efficiency was achieved over an engine with a brake thermal efficiency of 45% from a previous study by using a single-cylinder gasoline engine adopting pre-chamber jet combustion. These results were achieved with a mechanical compression ratio of 16:1 by optimizing the pre-chamber design, coating the pre-chamber inner walls with thermal insulation, and optimizing the combustion chamber shape, including compression ratio settings. The resulting configuration also achieved NOx emissions of 30 ppm and hydrocarbon emissions of 2000 ppm, which puts it in compliance with SULEV20 emission regulations. In addition, combustion stability was kept to a CPmi of 3% or lower, the rate of in-cylinder pressure increase to 300 kPa/deg or lower, and noise vibration requirements for actual vehicles were satisfied.

1. Introduction

Given the recent surge in initiatives to reduce CO₂ emissions, one of the attributed causes of climate change, the share of electric vehicles (EVs), fuel cell vehicles (FCVs), and other zero emission vehicles (ZEVs) is expected to increase. Still, ZEVs are projected to take some time to see widespread use. In the meantime, hybrid electric vehicles (HEVs) and plug-in hybrid electric vehicles (PHEVs) with internal combustion engines will likely be the mainstream for some time. In addition to power technology research on motor and battery technology, further improvement in the fuel economy of HEVs and PHEVs will require research on technology for enhancing brake thermal efficiency for internal combustion engines. In its research to date, Honda has achieved a gasoline engine brake thermal efficiency of 45%⁽¹⁾ and targets a brake thermal efficiency of 50% as the next step. Achieving a 50% brake thermal efficiency will require adopting lean burn in a high-compression engine.

Lean burning requires lean limit performance to greatly suppress knocking and to meet strict emission regulations. In its research on Homogeneous Lean-charge Spark Ignition (HLSI) combustion⁽²⁾, Honda has achieved both a high lean limit performance and low NOx emissions. However, homogeneous lean burning is known to be less effective in suppressing knocking than EGR combustion⁽³⁾. Another method, stratified combustion, is believed to suppress knocking by thinning in-cylinder end gas. With stratified

combustion, the issue to address is that NOx emissions are increased more with it than with homogeneous combustion. Stratified combustion has been researched along with Direct Injection Stratified Charge (DISC) combustion⁽⁴⁾. DISC combustion can combine high lean limit performance and low NOx emissions if adopted in conjunction with EGR. As combining with EGR will lessen the resulting increases in thermal efficiency, however, the authors saw it as necessary to use stratified combustion that realizes low NOx emissions without EGR to maximize the effect of enhancement in thermal efficiency.

Thus, in this study, pre-chamber jet combustion, a form of stratified combustion, was researched as means of achieving the performance requirements for lean burn. The engine configuration for pre-chamber jet combustion is as shown in Fig. 1. A pre-chamber is installed in the center of the combustion chamber, and that pre-chamber contains a spark plug and a dedicated direct injector (DI) to control the air-fuel ratio (AFR) at which energy from a mass-produced ignition system can achieve spark. In addition, a port injector (PI) is separately equipped to supply fuel to the main combustion chamber. Thus, stratified combustion is performed with different AFRs of air-fuel mixtures in the main chamber and pre-chamber. Furthermore, for the pre-chamber jet combustion, the ratio of the rich air-fuel mixture is determined by setting of the pre-chamber volume, allowing the dedicated DI injector to control AFR to the target values. This is what allows the configuration to

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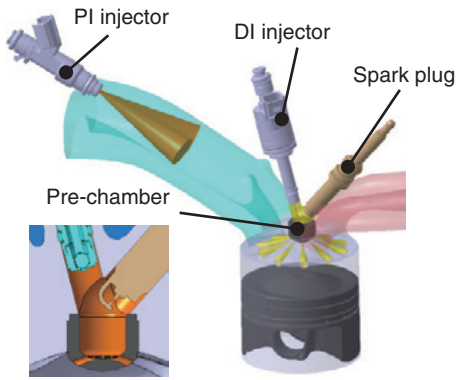


Fig. 1 Engine configuration for pre-chamber jet combustion

keep NOx emissions low with stratified combustion. In this paper, the authors will introduce the research on this pre-chamber jet combustion.

2. Setting Target Values

In setting the targets for achieving 50% brake thermal efficiency, a three-point increase will be achieved with advances in waste heat recovery, the injection system, injectors, cooling loss reductions, and other technology, and the remaining two points will be achieved with lean burn. The goal of this research is to enhance on the thermal efficiency of the previously studied engine with 45% brake thermal efficiency⁽¹⁾ (hereinafter “the previous engine”) by two or more points by incorporating lean burn. When lean burn is used, a three-way catalytic converter cannot be used to filter exhaust emissions. Thus, meeting stricter emission regulations will require a special post-treatment catalyst for lean burn and a reduction in feed emissions from the combustion chamber. To start, the feed emission reduction target values were examined. SULEV20 was set as target emission regulation, assuming that a special post-treatment catalyst for lean burn⁽⁵⁾ would be adopted to achieve this target. The authors assumed that the catalytic system will filter 99.6% of hydrocarbons (HC) and 95.9% of NOx. In the examination, mode engine rpms and load operation frequency were used as calculated from driving data in FTP-75 mode, simulating the mode through regular integration. From these calculations, as an efficient point load, the targets for feed NOx emissions at 50 ppm or less and for feed HC emissions at 2800 ppm or less were set. In addition, the values for combustion stability and rate of in-cylinder pressure rise ($dP/d\theta$) were required to satisfy noise vibration requirements for actual vehicles. Therefore, the following five performance targets for achieving maximum brake thermal efficiency, including a target for brake thermal efficiency enhancement were set. The indicator used for combustion stability is indicated mean effective pressure (CPmi).

- (1) Brake thermal efficiency target:
Increase at least two points from 45%-efficient technology
- (2) Emission targets for achieving SULEV20:
Feed NOx emissions of 50 ppm or lower
Feed HC emissions of 2800 ppm or lower
- (3) Actual vehicle noise vibration requirements:
CPmi of 3% or lower
 $dP/d\theta$ of 350 kPa/deg or less

Next, the AFR required to achieve target performance in terms of brake thermal efficiency and NOx emissions was examined. The discussion will start off with an explanation of the results from examining the AFR required for achieving the target brake thermal efficiency. Using one-dimensional simulations, the authors calculated change in brake thermal efficiency with Minimum Advanced Best Torque (MBT) driving and main burn duration, defined as 10% to 90% mass fraction burned (MFB), fixed at 25°, calculating for both EGR combustion and lean burn driving conditions. The calculations were based on performance results of the previous engine, and driving conditions were set to achieve maximum brake thermal efficiency. The calculation results are given in Fig. 2. These calculations revealed that lean limit performance would need to be an AFR of 28:1 or higher in order to increase brake thermal efficiency by two points.

Next, to clarify the AFR needed to achieve NOx emissions of 50 ppm or less, the characteristics for NOx emissions were determined at the AFR used in the previous engine. As NOx emissions correlated to maximum in-cylinder temperature, the emissions change by retard of ignition timing due to knocking. Therefore, NOx emissions were checked with driving conditions which were not affected by knocking: engine speeds of 2000 rpms and an indicated mean effective pressure (IMEP) of 470 kPa. The results, given in Fig. 3, show that in order to keep NOx emissions to 50 ppm or less, lean limit performance will need to be an AFR of 30:1 or higher. From the above two results, a lean burn capable of running at an AFR of 30:1 or higher will be necessary.

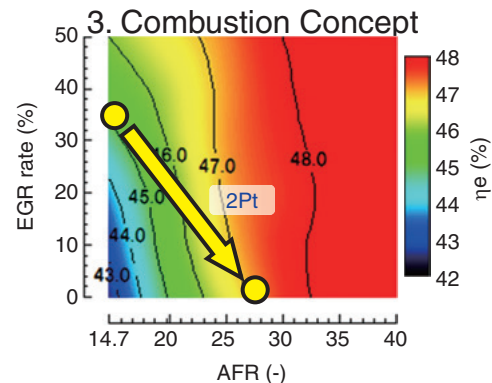


Fig. 2 Brake thermal efficiency for EGR combustion and lean combustion using 1-D simulation

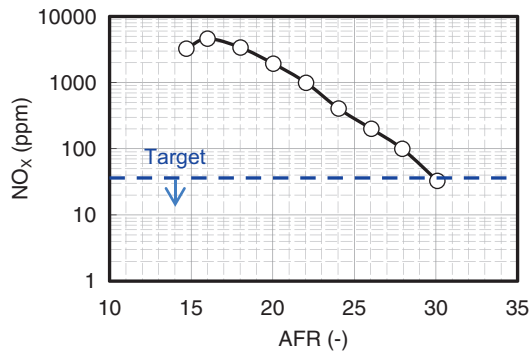


Fig. 3 Characteristics of NO_x emissions by AFR for base engine

The combustion process for pre-chamber jet combustion will now be explained through images visualizing pre-chamber jet combustion, shot from a direct bottom view, and the Computational Fluid Dynamics (CFD) calculations. The AFR is kept different between the pre-chamber and main chamber by having the PI injector supply fuel to the main chamber in the intake stroke and having the dedicated DI injector inject fuel into the pre-chamber in the compression stroke. Next, a spark is ignited by the spark plug in the pre-chamber [Fig. 4(1)], starting combustion in the pre-chamber and generating a difference in pressure between the main chamber and pre-chamber. This pressure difference is then used to force a flame to shoot out from the pre-chamber to the main chamber [Fig. 4(2)]. From this thermal energy, the air-fuel mixture in the main chamber will start combustion [Fig. 4(3)]. In order to achieve such a combustion format, the ability of pre-chamber jet combustion to maintain stable ignition performance and

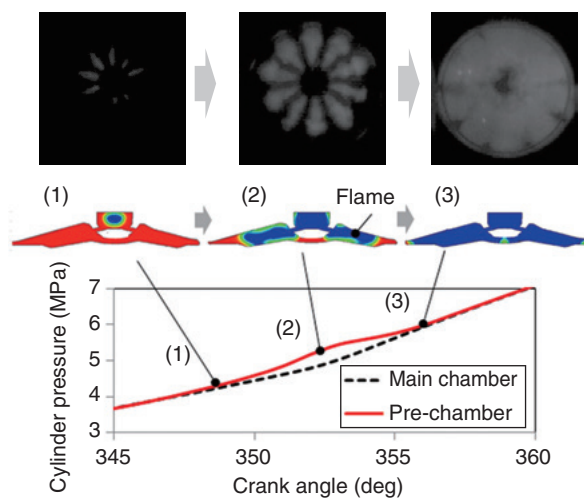


Fig. 4 Visualized direct bottom view and CFD calculations for pre-chamber jet combustion

multi-point ignition caused by the main chamber air-fuel mixture allow for rapid combustion. All this achieves a high lean limit performance and is expected to suppress knocking.

Pre-chamber jet combustion also offers advantages in terms of lowering NO_x emissions. In the past, stratified combustion secured stable ignition even with varying in-cylinder flow and DI injection cycles, necessitating an AFR and mixture distribution rich enough for fuel to ignite near the plug. In addition, high AFR mixture content was often found near the plug. These increased NO_x emissions have been an issue. In pre-chamber jet combustion, fuel is injected into the pre-chamber in the compression stroke, containing fuel within the pre-chamber without discharging it to the main chamber. Therefore, the pre-chamber volume can be configured to adjust the area where the AFR is richer than the overall in-chamber AFR. As another advantage, the air-fuel mixture in the pre-chamber can be controlled to the target AFR, simultaneously stabilizing the rich air-fuel mixture in the plug vicinity each cycle. The above characteristic allows this form of combustion to reduce NO_x emissions with stratified combustion.

4. Engine Design

The following is an explanation of the impacts of each design parameter of the pre-chamber jet combustion engine on achieving the target performance.

4.1. Evaluation Engine Specifications

The specifications in Table 1 are for the test engine used in this study. This was a single-cylinder engine bored out to a diameter of 73 mm while keeping the stroke/bore ratio and compression ratio, two important elements in the previous study, the same as in the previous engine. An Atkinson cycle, which has also been an important element, is again adopted in this study, holding the intake valve open longer. Compression ratio for the compression stroke was set to the effective compression ratio as calculated from the cylinder volume for compression at top dead center (TDC) and that when the intake valve closes. Compression ratio for the expansion stroke is shown as the expansion ratio (mechanical compression ratio).

4.2. Pre-chamber Design

The authors contemplated the pre-chamber design, which determines the thermal energy supplied to the main combustion chamber from the pre-chamber and in turn greatly impacts combustion in the main chamber in a pre-chamber jet combustion engine. As shown in Fig. 5, the main design parameters for the pre-chamber are the pre-chamber volume, number of nozzles, nozzle diameter, and the nozzle cross-sectional area as calculated from the number of nozzles and nozzle diameter. In order to arrive at the optimum design, the impacts on each performance by these parameters were confirmed. As will be explained

Table 1 Test engine specifications for pre-chamber jet combustion

Number of cylinders	1
Displacement volume (cm ³)	458
Bore (mm)	φ 73
Stroke (mm)	109.5
Stroke/bore ratio	1.5
Expansion ratio	17.0
Effective compression ratio	12.5
Aspiration	Supercharger
Fuel supply (main/pre)	(PI/DI)
Ignition energy (mJ)	60

in detail below, the pre-chamber volume and nozzle cross-sectional area largely contributed to achieve the target values.

Figure 6 is a graph showing the impacts on each performance if pre-chamber volume is changed at an engine speed of 2000 rpms, IMEP of 810 kPa, and AFR of 35:1, which are the conditions for achieving maximum brake thermal efficiency with previous engine. As the pre-chamber volume is reduced, NO_x emissions and dP/dθ both decrease, whereas the 10-90% MFB angle tends to increase. There was no decline in thermal efficiency down to a pre-chamber volume of 2.0 cm³, beyond which smaller volumes resulted in reduced thermal efficiency. Thus, a smaller pre-chamber volume results in a decrease in rich air-fuel mixture ratio, consequentially reducing NO_x emissions. Smaller volumes also reduce the amount of gas in the pre-chamber, resulting in less thermal energy supplied from pre-chamber to main chamber and in turn both reducing the dP/dθ and extending burn duration. If thermal energy supplied from pre-chamber to main chamber cannot maintain combustion stability at an AFR of 35:1, HC increases and thermal efficiency will presumably decline greatly. From the above performance results, pre-chamber volume was set to 2.0 cm³, which minimizes NO_x emissions and dP/dθ without reducing thermal efficiency.

Figure 7 is a graph showing the impacts on each performance if the pre-chamber nozzle cross-sectional area is changed at an engine speed of 2000 rpms, IMEP

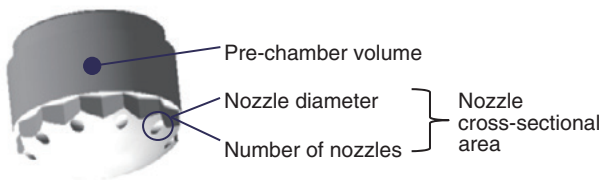
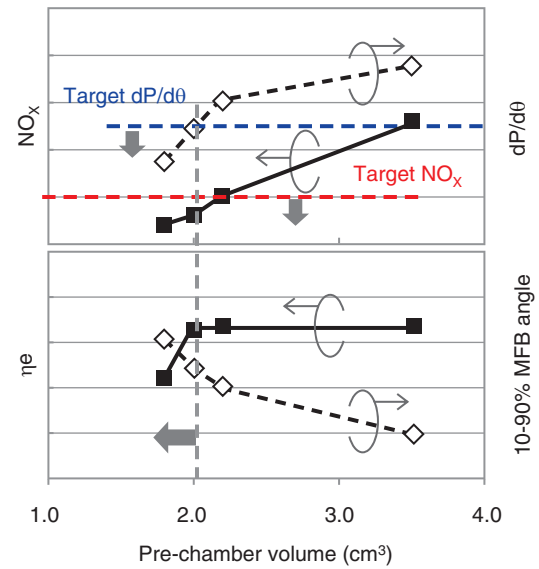
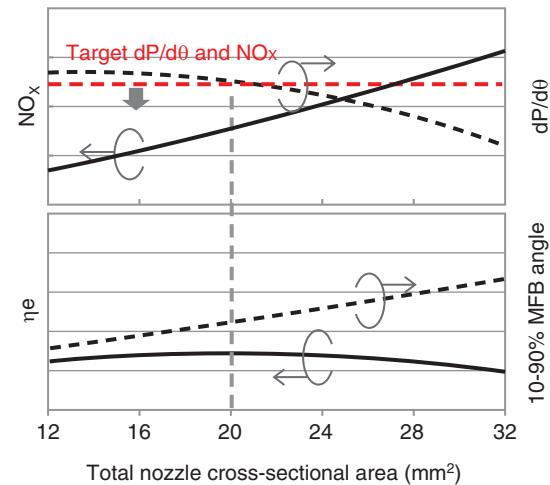


Fig. 5 Pre-chamber design parameters

of 810 kPa, and AFR of 35:1. As the pre-chamber nozzle cross-sectional area is reduced, NO_x emissions and the 10-90% MFB angle will also reduce, but dP/dθ will be increased. A pre-chamber nozzle cross-sectional area of 20 mm² maximizes thermal efficiency; thermal efficiency is reduced both above and below this value. The reason for this is that, as the pre-chamber nozzle cross-sectional area is reduced, pre-chamber pressure rises, in turn increasing the thermal energy supplied to the main chamber and keeping combustion stable. Thus, ignition timing can be retarded while keeping the same combustion stability for lower NO_x emissions by lowering the maximum in-chamber temperatures. As for the change in the thermal efficiency, if the pre-chamber nozzle cross-sectional

Fig. 6 Comparison of pre-chamber volume on NO_x and thermal efficiencyFig. 7 Comparison of NO_x and thermal efficiency by total nozzle cross-sectional area

area is too large, pre-chamber pressure rises less, in turn decreasing the thermal energy supplied from pre-chamber to main chamber. Therefore, at an AFR of 35:1, chamber combustion stability cannot be maintained and thermal efficiency drops due to an increase in HC. Conversely, if the pre-chamber nozzle cross-sectional area is too small, and the thermal energy supplied from pre-chamber to main chamber increases the pre-chamber pressure rises beyond that required to provide combustion stability at an AFR of 35:1. Presumably, this might greatly increase cooling loss in the pre-chamber and decrease thermal efficiency. In contrast, if thermal energy supplied from pre-chamber to main chamber increases, $dP/d\theta$ will rise due to the rapid combustion. From the above performance results, pre-chamber nozzle cross-sectional area was set to 20 mm^2 , which keeps NOx emissions and $dP/d\theta$ at or below the target values and maximizes thermal efficiency. Specifically, the pre-chamber was designed with 10 nozzles at a nozzle diameter of $\phi 1.6 \text{ mm}$.

Each performance for the pre-chamber after design optimizations is shown in Fig. 8. This figure relates the performance results and AFR under heavy load at an engine speed of 2000 rpm and IMEP of 810 kPa. Just as in the previous study, in their brake thermal efficiency calculations, the authors used supercharger loss, estimated using mechanical friction and one-dimensional simulations equivalent to an inline four-cylinder engine. In terms of combustion stability, the pre-chamber achieves a CPmi of 3% or lower up to an AFR of 38:1 and operates at MBT at

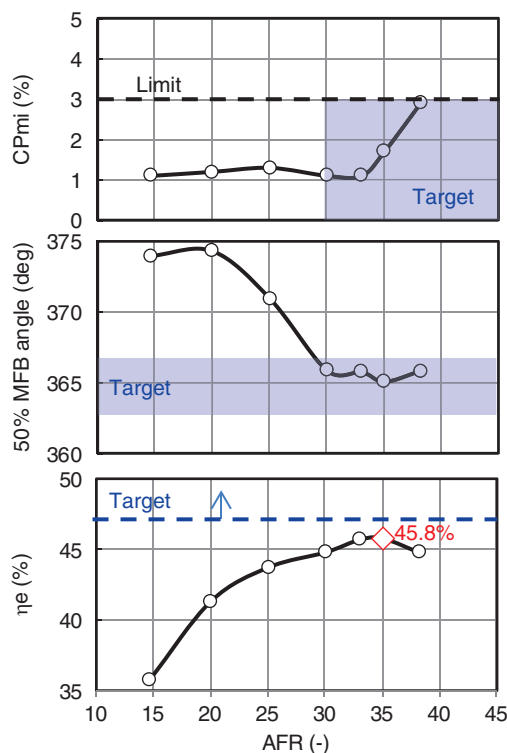


Fig. 8 Lean characteristics under heavy load

AFRs of 30:1 or higher.

4.3. Technology for Reducing Pre-chamber Cooling Loss

Given the greater specific surface area (surface-to-volume ratio) for pre-chamber combustion, cooling loss to the wall surface could be higher. Thus, to enhance thermal efficiency, the authors worked to reduce cooling loss in the pre-chamber. To this end, the area of the pre-chamber inner wall was coated with thermal insulation as shown in red in Fig. 9. This area covers 37% of the surface area of the pre-chamber. Figure 10 shows the effect of coating at an engine speed of 2000 rpm, IMEP of 810 kPa, and AFR of 35:1. Thus, the reduction in cooling loss and accompanying increase in thermal energy to the main chamber have reduced unburned loss, increasing thermal efficiency by 0.5 points. This did not change time loss as knocking was not affected; however, waste heat also increased as not all of the above loss reduction translates to thermal efficiency.

4.4. Combustion Chamber Surface-to-volume Ratio

In the pre-chamber jet combustion engine, the combustion chamber surface-to-volume ratio is increased over that of standard gasoline engines at the same compression ratio. Also, at the same pre-chamber volume, this effect increases with the compression ratio. Thus, the relationship between compression ratio and the combustion chamber surface-to-volume ratio was investigated. The thermal efficiency sensitivity map shown in Fig. 11

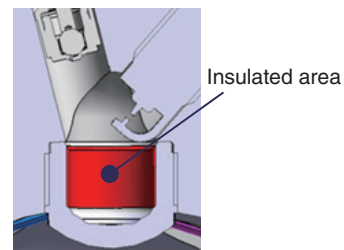


Fig. 9 Insulated area of pre-chamber

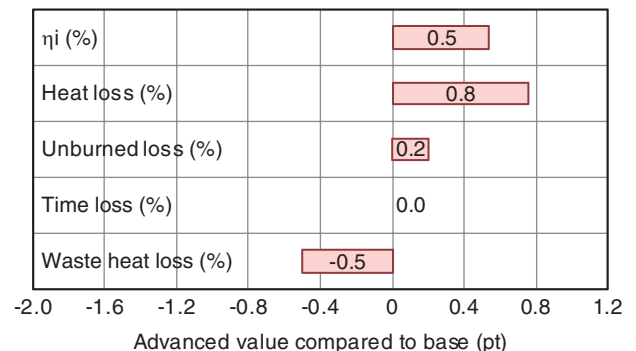


Fig. 10 Effects of coating on pre-chamber

gives the results from comparing compression ratio and surface-to-volume ratio. As step 1 in the figure, to confirm the impact of the combustion chamber's surface-to-volume ratio, the authors confirmed the performance of the combustion chamber's surface-to-volume ratio when lowering the compression ratio. It was found that, while reducing compression ratio did reduce the theoretical thermal efficiency, it also greatly reduced cooling loss and unburned loss, conversely resulting in a net enhancement in thermal efficiency. Next, as step 2 in the figure, the combustion chamber shape to further reduce its surface-to-volume ratio and enhance thermal efficiency was optimized. From the above, including the revised combustion chamber shape, the compression ratio for the pre-chamber jet combustion engine was set to 16:1.

4.5. In-cylinder Flow

In the previous study, the intake ports for the 45% brake thermal efficiency engine adopted high tumble in order to increase flame velocity. In pre-chamber jet combustion, however, the authors expected the in-cylinder flow demands to change due to the change in combustion format. Thus, an in-cylinder flow for optimum pre-chamber jet combustion was selected. Two port specifications with different tumble ratios were compared for performance as shown in Fig. 12. Port A had a tumble ratio of 0.4 while Port B had a tumble ratio of 1.8. Engine speed is 2000 rpm and IMEP is 510 kPa for these comparison results. Comparing port performance proved that differences in tumble ratio had no effect on burn duration, but the higher tumble ratio in the Port B specification reduced combustion stability. As to why burn duration remained unchanged, given that the spark plug in pre-chamber jet combustion resides in the pre-chamber, it is possible that enhanced flow does not promote initial flame kernel growth as it normally does in an SI engine. Next, in-

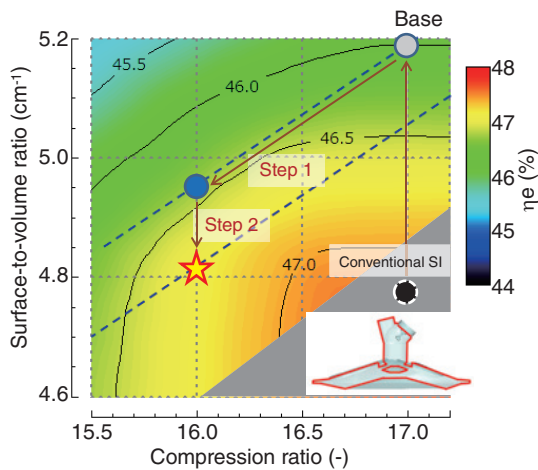


Fig. 11 Effects of surface-to-volume ratio and compression ratio on brake thermal efficiency in pre-chamber jet combustion engine

cylinder flow was analyzed using CFD to analyze the drop in combustion stability, as shown in Fig. 13. This figure shows the distribution of in-cylinder flow at a crankshaft angle of 80 deg before top dead center in the compression stroke after fuel is injected into the pre-chamber. Whereas almost no gas is exchanged between the pre-chamber and main chamber in Port A, gas is clearly exchanged between pre-chamber and main chamber in Port B due to the residual tumble. In the compression stroke, fuel is injected by the dedicated DI injector in the pre-chamber to keep it in the pre-chamber. This fuel is forced into the main chamber by residual tumble, resulting in a different amount of fuel in the pre-chamber when combustion starts in each cycle. Thus, presumably due to this variation in the thermal energy supplied from pre-chamber to main chamber from cycle to cycle, combustion in the main chamber also varies for each cycle, reducing combustion stability. From the above, Port A with the lower tumble ratio was selected for the pre-chamber jet combustion in-cylinder flow specifications.

5. Verification Results

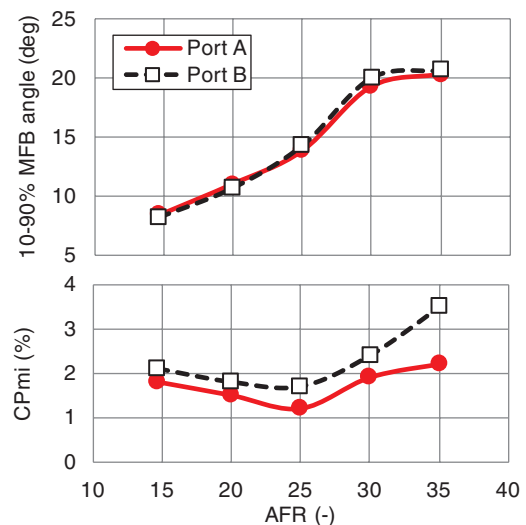


Fig. 12 Performance results for intake ports with different tumble ratios

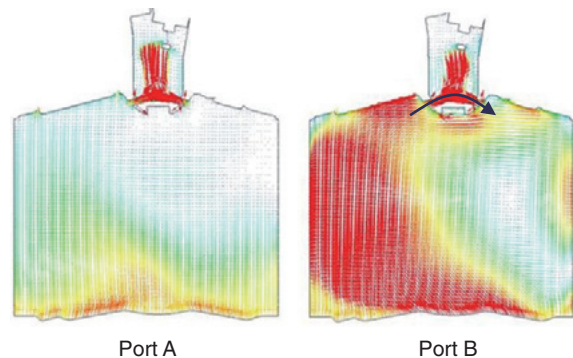


Fig. 13 In-cylinder flow with CFD at crank angle of 80° BTDC

Table 2 Final test engine specifications

Number of cylinders	1
Engine type	DOHC 4-valve
Displacement volume (cm ³)	458
Bore (mm)	φ 73
Stroke (mm)	109.5
Stroke/bore ratio	1.5
Expansion ratio	16.0
Effective compression ratio	12.5
Pre-chamber volume (cm ³)	2.0
Nozzle diameter (mm)	φ 1.6
Number of nozzles	10
Pre-chamber specification	Thermal insulation coating
Intake port	Weak flow
Aspiration	Supercharger
Fuel supply (main/pre)	PI/DI
Ignition energy (mJ)	60

The final values achieved with a single-cylinder engine applying all the supporting technology for a pre-chamber jet combustion engine described above were confirmed. The engine specifications in Table 2 are for the final specifications. Just as in the previous study, the authors calculated brake thermal efficiency using supercharger loss, as estimated using mechanical friction and one-dimensional simulations equivalent to an inline four-cylinder engine.

Figure 14 relates each performance and AFR under heavy load at an engine speed of 2000 rpms and IMEP of 870 kPa. In terms of combustion stability, the pre-chamber achieves a CPmi of 3% or lower up to an AFR of 40:1 and operates at MBT at AFRs of 35:1 or higher, achieving a two-point increase over 45% brake thermal efficiency. At the same AFR of 35:1 which increases brake thermal efficiency, the other performance targets were also achieved: NO_x emissions were 30 ppm, HC emissions were 2000 ppm, CPmi was 1.2%, and dP/dθ was 300 kPa/deg.

A comparison of brake thermal efficiency at an engine speed of 2000 rpms under each load between the pre-chamber jet combustion engine and the 45% brake thermal efficiency engine for the previous study is shown in Fig. 15. It was confirmed that brake thermal efficiency also increased outside maximum load conditions.

The details in this study were verified on a single cylinder engine. Hereafter, toward practical applications in a multi-cylinder engine for automobiles applying these achievements, two outstanding topics for future study are development of turbocharger system compatible with both

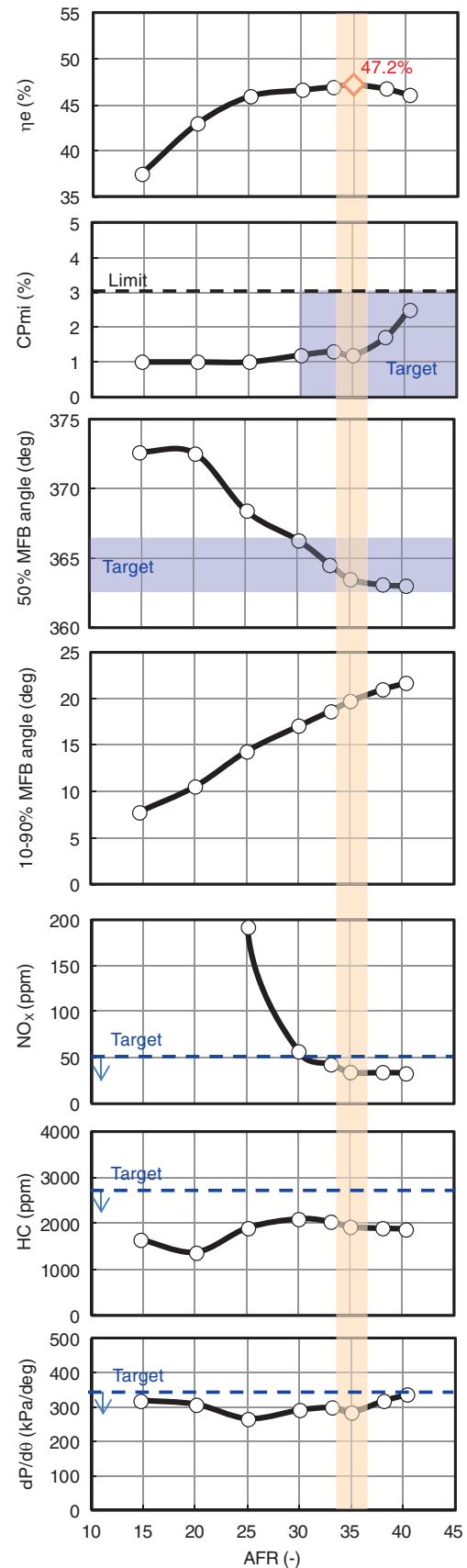


Fig. 14 Lean characteristics at high load with final test engine specifications

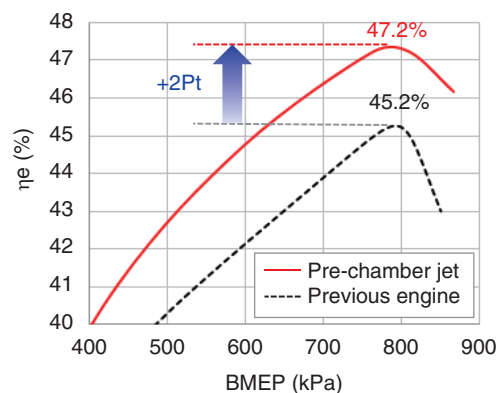


Fig. 15 Achieved brake thermal efficiency at engine speed of 2000 rpm

partial load and output point operation and development of a lean burn-specific catalyst.

6. Conclusion

Pre-chamber jet combustion was applied to a single-cylinder gasoline engine with mechanical compression ratio of 16:1, optimizing the pre-chamber design and reducing the losses caused by the addition of pre-chambers. These developments achieved the following performance at an engine speed of 2000 rpms, IMEP of 870 kPa, and AFR of 35:1.

- (1) Brake thermal efficiency target:
Increased two points from 45%-efficient technology
- (2) Emission targets for achieving SULEV20:
Feed NO_x emissions of 30 ppm
Feed HC emissions of 2000 ppm
- (3) Actual vehicle noise vibration requirements:
CPmi of 3% or lower up to AFR of 40:1
dP/dθ of 300 kPa/deg

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