

Research on Combustion Improvement Techniques by Intake Valve Offset and Squish Effect

Takamori SHIRASUNA*

Hideki SAITO*

Tomokazu NOMURA*

ABSTRACT

Reported in this paper are the technologies for improvement of combustion efficiency by applying two simple methods to a single cylinder, 110 cm³ displacement, four-stroke, two-valve gasoline engine.

In the first attempt, we tried to improve combustion efficiency by increasing tumble of the air-fuel mixture flow. To increase tumble, we devised an offset intake valve design in which a part of the intake valve was located outside of the cylinder bore. With this offset intake valve configuration, a part of the inlet port perimeter was blocked causing disturbance of air-fuel mixture flow along the cylinder wall that resulted in a strong turbulence. The increased turbulence permitted lean burn at an air-fuel ratio leaner by two points, reducing Indicated Specific Fuel Consumption by 4.8% from that of the base engine. With the intake valve shifted outwards against the cylinder bore, the spacing next to the exhaust valve increased, allowing the intake valve diameter to be enlarged to compensate for the deterioration of the maximum power.

In the second attempt, we tried to improve combustion efficiency by increasing the reversed squish flow of the air-fuel mixture. As the means to increase reversed squish flow, we employed a slant-parallel squish configuration. With the application of this squish arrangement, the margin against knocking generation was enhanced and the compression ratio was increased to 9.5 from the original 9.0 while reducing the Indicated Specific Fuel Consumption by 2.6%.

The offset intake valve design coupled with the high compression ratio produced by the slant-parallel squish design lowered the Indicated Specific Fuel Consumption by 6.0% compared to the base engine.

1. Introduction

Recent years have seen increasing demands to reduce CO₂ emissions to cope with global warming. In the field of mobility, fuel economy improvement therefore has been a key theme for research. Various research and developments for fuel economy improvement have been currently conducted in the field of small motorcycles, which account for more than 90% of the world's motorcycle sales. Gasoline engines for motorcycles require simple methods of improvement of fuel economy. Extension of the lean burn range and/or increase of the compression ratio by controlling knocking are some of the effective ways to satisfy the requirement. To extend the lean burn range, it is known that such options as a multi-spark ignition system⁽¹⁾ that strengthens ignition, application of a sub combustion chamber⁽²⁾, in-cylinder turbulence to accelerate combustion, etc. are effective for the said purpose. For example, with respect to in-cylinder turbulence, research has been conducted using a tumble generator valve⁽³⁾ or a swirl generator valve⁽⁴⁾. Application of such a device, however, requires additional parts and control devices.

In the meantime, research has been conducted for

other options such as reshaping of the intake port⁽⁵⁾ and use of masking⁽⁶⁾ to intensify tumble flow from a simple mechanical arrangement without using additional parts. However, such an option tends to increase resistance in the intake passage, causing reduction of intake air volume. There have been only a few cases of research on improvement of fuel economy from intensified tumble flow.

In this research, we defined the partially open throttle tumble ratio as a new indicator of tumble flow intensity, and clarified the relationship between tumble flow intensity and fuel economy improvement through basic experiments using stationary tumble generator flaps (hereafter referred to as "flap") of various sizes. The tumble flow was further intensified to extend the lean burn range by the use of masking effects from the intake valve offset to improve fuel economy, at the same time increasing the intake valve diameter was to prevent lowering of maximum power resulting from reduced intake air volume.

Such ways as improvement of combustion chamber cooling, reduction of residual gas, application of squish zone, etc. are known to control knocking when the compression ratio is raised. Among these, application of

* Motorcycle R&D Center

squish is a simple construction without using any device. Many studies on squish have been reported^{(7), (8)}. With a squish zone created, burning velocity increases regionally in the far end of the combustion chamber due to the reversed-squish flow of mixture that enters the squish zone when the piston descends after passing TDC. From that effect, flame propagation reaches the unburnt mixture in the far end of the combustion chamber before the knocking occurs, which is considered to be the reason squish helps to control knocking⁽⁹⁾. Also, it is considered that slant-parallel squish is the shape of the squish zone effective in creating reversed-squish flow. Such knowledge, however, are obtained from research on four valves per cylinder engines. There are only a few cases of research in single cylinder, two-valve engines (two valves per cylinder) on the effective squish shape, contribution of the squish shape to the control of knocking, and increase of compression ratio permitted by the control of knocking. Using CFD, we analyzed the reversed-squish flow in various types of squish shape in the two-valve per cylinder, small displacement engine. We then conducted tests to clarify the correlation between intensity of reversed squish flow and knocking using actual engines. Moreover, we conducted combustion analysis to clarify how squish helped to control knocking.

Lastly, we conducted tests on the combined configuration of offset intake valve and squish, and investigated the degree of influence on fuel economy and power output for extension of the lean-burn range and increase of the compression ratio in accordance with the respective goals of the aforementioned configuration measures.

2. Test Engine and Test Conditions

2.1. Major Specifications of Test Engine

A commercially available engine for small motorcycles was used for the tests. Table 1 shows the major specifications of the test engine. The engine was an air-cooled, four-stroke, single cylinder, two-valve gasoline engine with a displacement of 109.2 cm³. Fuel was supplied by a carburetor. It should be noted, however, that in the tests to alter the air-fuel ratio (A/F), the pressure in the carburetor float chamber was regulated to control A/F. A rotary encoder was positioned on the crankshaft to obtain crank angle signals of 360 pulses per revolution. For measurement of the pressure in the cylinder, a piezoelectric pressure transducer (Kistler 6041A) was used. The engine was linked with the dynamometer via the transmission system, and controlled at constant engine speeds and loads. Testing on the unit engine was conducted at a constant transmission ratio.

2.2. Conditions of Test Operations

To measure the benefits of the combustion improvement, the lean burn limit, combustion analysis and fuel economy benefits were assessed under partially loaded conditions, and the power output and the knocking were assessed at wide open throttle conditions. The operation conditions are:

Table 1 Specifications of experimental engine

Engine type	Air cooled, four stroke, single cylinder, gasoline engine
Bore × Stroke [mm]	50.0 × 55.6
Displacement [cm ³]	109.2
Compression ratio	9.0
Valve diameter [mm x number]	Intake: 25.5 × 1 Exhaust: 21.0 × 1
Valve lift [mm]	Intake: 5.8 Exhaust: 5.6
Ignition system	Electronically controlled capacitor discharge ignition
Fuel system	Carburetor
Fuel type Research octane number (RON)	Regular gasoline 90
Maximum brake power output	6.27 kW at 8000 r/min
Maximum brake torque	8.72 Nm at 6000 r/min

Partially loaded condition: Engine speed: 3750 r/min; BMEP400 kPa; MBT

Fully loaded condition: Engine speed: 3000-8500 r/min; Throttle: Wide open throttle; A/F 12.4

2.3. Lean-burn Limit of Test Engine

Figure 1 shows the Indicated Specific Fuel Consumption (ISFC) and the Coefficient of Variation of Indicated Mean Effective Pressure (COV_{IMEP}) when the A/F is altered under partially loaded condition. The A/F that indicated the lowest ISFC was 16.5 in the test engine. Also, the COV_{IMEP} at A/F of 16.5 was 5%. As an increase of COV_{IMEP} occurred if the A/F was further shifted to the lean side, a further reduction of ISFC was thought difficult. To further lower ISFC, an extension of the lean burn range by an increase of turbulence in the cylinder, etc. was necessary. As the means to extend lean burn range, we focused our attention on the intensity of tumbling.

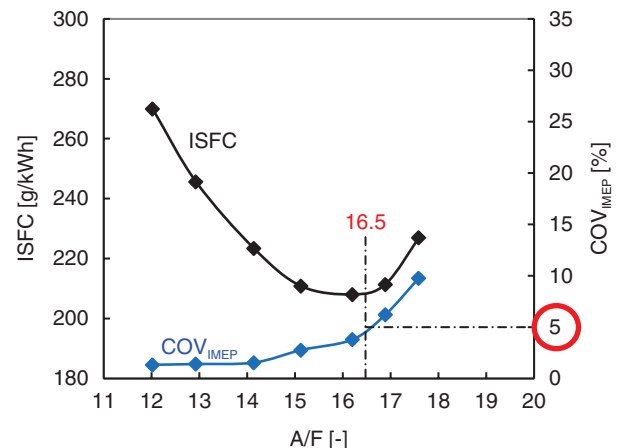


Fig. 1 ISFC and COV_{IMEP}

3. Indicator of Tumble Flow Intensity and Prospective Output in Constant Flow Tests

In this research, tumble ratio (TR) was used as the indicator of intensity of tumble flow. Also, as indicator of prospective power output, the integrated value of effective intake valve opening area over the crank angle (Z_i) was used. These indicators were measured in the steady flow rig tests.

3.1. Definition of Tumble Ratio

TR is expressed by the ratio of the tumbling angular velocity (ω_R) of the air in the cylinder against the angular velocity of the crankshaft (ω_E) when the charging of mixture is completed, and defined by Eqs. (1), (2) and (3)⁽⁷⁾.

In this research, since TR was estimated by relative value instead of absolute value, volumetric efficiency (η_v) was simply treated as 1.

$$TR = \frac{\omega_R}{\omega_E} = \frac{L_d \eta_v \int_{\alpha_1}^{\alpha_2} C_f N_r d\alpha}{(\int_{\alpha_1}^{\alpha_2} C_f d\alpha)^2} \quad (1)$$

$$L_d = \frac{BS}{n_v D_v^2} \quad (2)$$

Where,

TR : Tumble ratio

ω_R : Tumble angular velocity

ω_E : Angular velocity of crankshaft

L_d : Engine shape factor

η_v : Volumetric efficiency

α_1 : Crank angle at inlet valve opening

α_2 : Crank angle at inlet valve closing

C_f : Flow coefficient

N_r : Non-dimensional rig tumble

B : Cylinder bore

S : Engine stroke

n_v : Number of intake valves

D_v : Intake valve diameter

N_r is expressed by the ratio of tumbling angular velocity against flow velocity in the intake port.

$$N_r = \frac{\omega_R B}{V_0} = \frac{8M}{\dot{m} V_0 B} \quad (3)$$

Where,

V_0 : Theoretical intake flow velocity

M : Torque measured by tumble flow meter

$\dot{m}$: Air mass flow rate through port

TR was assessed by the two methods of actual measurement in the steady flow rig and CFD analysis that simulated steady flow testing.

Figure 2 shows the test equipment used for the steady flow tests. Table 2 shows the conditions of the steady flow tests.

In this research, TR was also assessed at partial throttle openings. As the intake resistance increases at a partial opening, the TR lowers. It was therefore necessary to improve accuracy of TR measurement. A compact impulse

Table 2 Conditions of steady flow test

Valve lift [mm]	1, 2, 3, 4, 5, 6
Throat diameter [mm]	0.535, 1.31, 2.75
Pressure of surge tank B (ΔP_2) [kPa]	5
Load cell rated capacity [N]	0.5
Rated torque [Nm]	5×10^{-3}

tumble meter for exclusive use was prepared, and the experiment was conducted.

Also, actual measurements of TR were conducted two times with the branch pipe reversed. The total of each TR measured on the left and right sides was evaluated as the representative TR .

Figure 3 shows the CFD analysis model used for TR

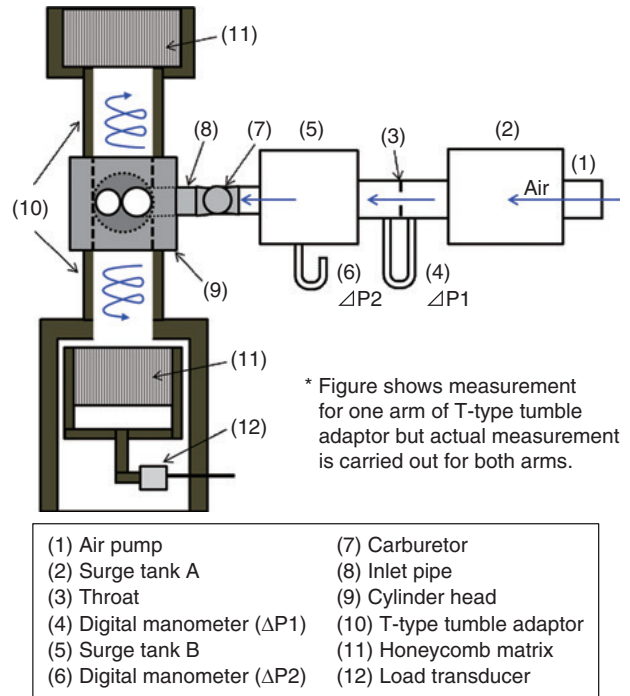


Fig. 2 Flow test rig

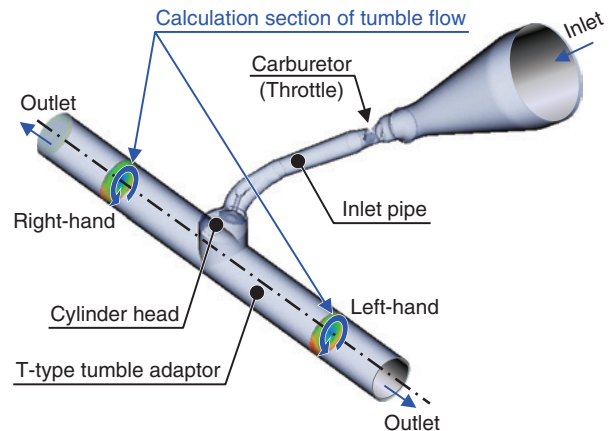


Fig. 3 CFD model for tumble ratio calculation

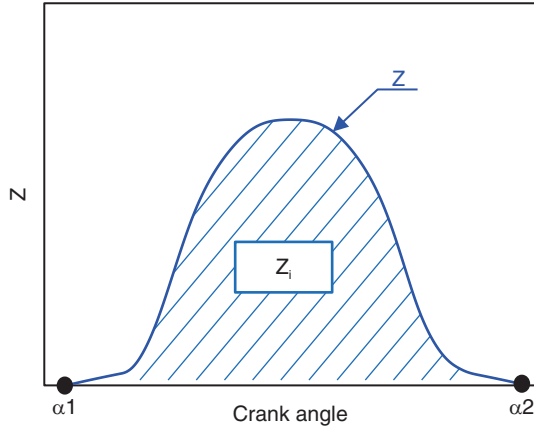
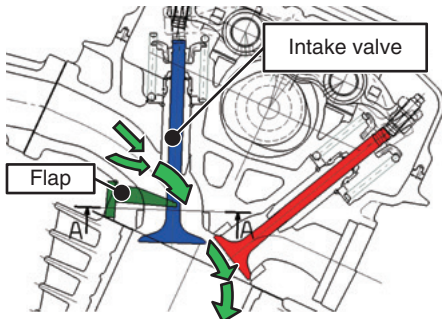
Table 3 CFD conditions for steady flow test

Software name	SCRYU/Tetra
Solution type	Steady state flow
Valve lift [mm]	1, 2, 3, 4, 5, 6
Pressure difference of Inlet and outlet [kPa]	5
Cell type	Tetrahedron
Boundary cell type [mm × layers]	Prism 0.3 × 2
Number of cells	4,500,000
Differencing scheme	MUSCL scheme
Pressure velocity coupling	SIMPLEC
Turbulent model	Standard k-ε

calculation. Table 3 shows the calculation conditions of CFD analysis.

3.2. Prospective Power Output Indicator

The factor most influential for engine power output is the amount of air drawn into the combustion chamber in a cycle. As an indicator for assessment of the amount of intake air, Z_i in the steady flow tests is effective, and is defined by the Eq. (4). On the other hand, Z_i is defined by Eq. (5) at the CFD analysis⁽¹¹⁾. And Fig. 4 shows a conceptual diagram of Z_i . In this research, Z_i was used as an indicator of prospective power output.

Fig. 4 Definition of Z_i 

Name	Base (Without flap)	Flap A	Flap B	Flap C	Flap D
Flap shape (Section A-A)					
Flap area ratio [%]	0	19.4	36.0	48.6	60.9

Fig. 5 Installed flaps and shapes

$$Z_i = \int_{\alpha_1}^{\alpha_2} Z d\alpha = \int_{\alpha_1}^{\alpha_2} \beta A \sqrt{\frac{\Delta P_1}{\Delta P_2}} d\alpha \quad (4)$$

Where,

Z_i : Integrated value of effective intake valve opening area over crank angle

Z : Effective intake valve opening area

β : Flow coefficient of throat

A : Cross section of throat

ΔP_1 : Pressure difference of throat

ΔP_2 : Pressure of surge tank B

$$Z_i = \int_{\alpha_1}^{\alpha_2} Z d\alpha = \int_{\alpha_1}^{\alpha_2} C_f A_v d\alpha \quad (5)$$

Where,

A_v : Intake valve opening area

4. Correlation between Tumble Flow Intensity and Fuel Economy Improvement

Regarding the relationship between tumble flow intensity and fuel economy improvement, only a few examples have been reported up to now. In an attempt to identify this relationship, four types of flap were installed to alter tumble flow intensity in the intake port, and conducted CFD analysis and research on the actual engine. We also identified issues that could arise when the fuel economy was improved by intensifying of tumble flow.

4.1. Intensifying of Tumble Flow by Tumble Flow Generator Flap

To intensify tumble flow, protruding flaps of various size were positioned inside of the intake port. We analyzed how the flap affects Z_i and TR by CFD analysis that simulated constant flow tests. Figure 5 shows the installed flap and these shapes. The areas protruding in the intake port were increased respectively in the order of A, B, C and D.

Figure 6 shows Z_i and TR calculated in the CFD analysis. The tendency was that TR increased along with an increase of surface area of the flap, which meant that the flap effectively generated tumble flow. No decrease of Z_i at an identical throttle opening was seen in the small throttle opening range regardless of the flap size. As the throttle opening increased,

however, the tendency was that Z_i at an identical throttle opening decreased proportionally to the flap size. The reason could be considered that when the throttle opening is small, as the throttle valve determined the minimum cross-sectional area of intake passage, the installation of a flap has no effect on the suction resistance. It could be considered that when the throttle opening was large, the cross-sectional area of intake passage became smallest where the flap was installed, causing an increase of suction resistance and lowering of Z_i .

Figure 7 shows Z_i and TR actually measured in the steady flow tests. The data shows the same tendency as in the CFD analysis shown in Fig. 6, even in the partial opening in addition to the wide open throttle (WOT). It was confirmed that the assessment by CFD analysis was appropriate for relative comparison of Z_i and TR . Thus, we decided to assess Z_i and TR later by CFD analysis data. TR at a throttle opening

of 15 degrees, which was the typical operating condition, was also defined as the partially open throttle TR (POT TR).

Figure 8 shows CFD analysis data indicating distribution of flow velocity in the intake port and cylinder. The data was calculated at a throttle opening of 15 degrees and valve lift of 4.5 mm. The flow of air in the intake port was directed to the exhaust valve side by the flap. The directed air entered the cylinder at a high flow velocity, and induced strong tumble flow. Along with an increase in size, the flap restricted flow of air into the cylinder from the cylinder sleeve side of the intake valve, preventing induction of inverse tumble flow. These effects contributed to the increase of POT TR .

4.2. Intensity of Tumble Flow and Fuel Economy Improvement

The impact of four types of flap on fuel economy

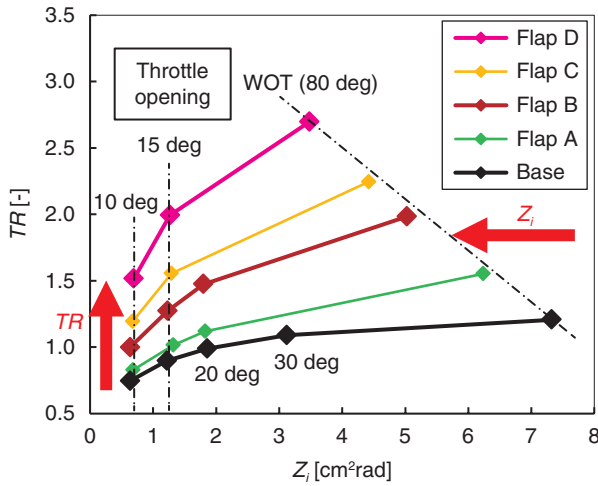


Fig. 6 Z_i and TR (CFD)

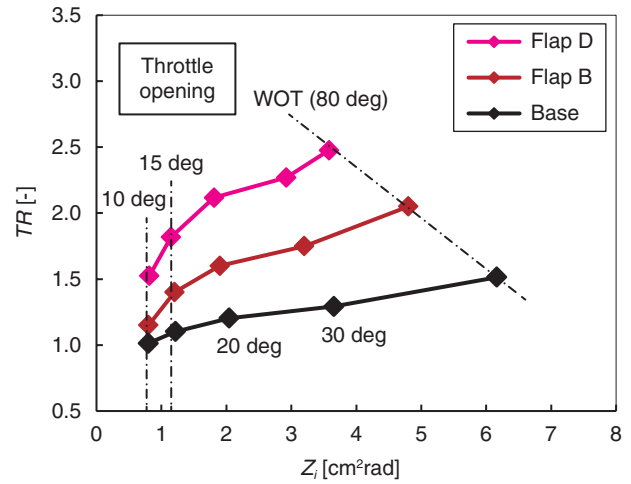


Fig. 7 Z_i and TR (Actual measurement)

Name	Base (Without flap)	Flap A	Flap B	Flap C	Flap D
Flap shape					
Section of axis of valves					
Section A-A (20 mm below from cylinder top surface)					
POT TR [-]	0.9	1.0	1.3	1.6	2.0

Fig. 8 CFD calculation (in-cylinder flow)

improvement was analyzed by experiments. Figure 9 shows the relationship between A/F and ISFC under the typical operating conditions in IDC. Figures 10, 11, 12 and 13 show results of the combustion analysis. In addition, the crank angle from ignition to the Mass Fraction Burn of 10% was defined as the initial combustion period ($MFB_{0-10\%}$), and the crank angle from the Mass Fraction Burn of 10% to 90% was defined as the main combustion period ($MFB_{10-90\%}$).

Compared to the basic design, the design equipped with the flap showed a tendency for the ISFC in the lean range to lower along with an increase of flap surface area. It has been clarified from the combustion analysis that in the basic design, $MFB_{10-90\%}$ increased rapidly and the degree of constant volume also lowered rapidly along with the leaning of A/F. Also, COV_{IMEP} increased rapidly from near an A/F of 17, indicating that the combustion became unstable.

Accordingly, lowering of ISFC with the basic design is unattainable in the range of A/F leaner than 17.

Meanwhile in the flap-equipped design, $MFB_{0-10\%}$ and $MFB_{10-90\%}$ were reduced in all A/F zones. Also, the steep increase of $MFB_{10-90\%}$ along with the leaning of A/F was lessened. Thus, both the steep lowering of the degree of

constant volume and the steep increase of COV_{IMEP} in the lean range were alleviated. Consequently, the lean burn limit has been extended, making it possible to lower the ISFC.

Figure 14 shows the relationship between POT TR and ISFC lowering ratio. The ISFC lowering ratio was calculated

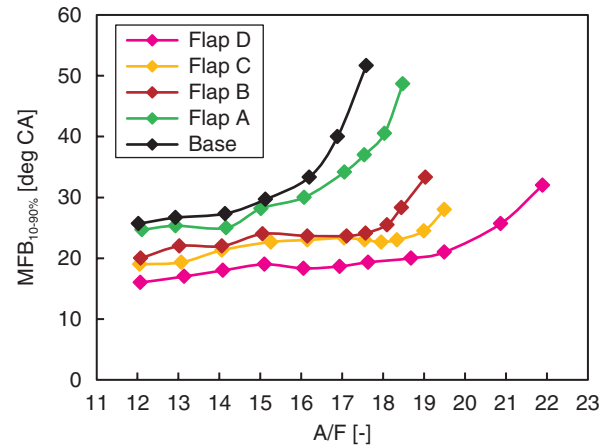


Fig. 11 Relationship between A/F and $MFB_{10-90\%}$

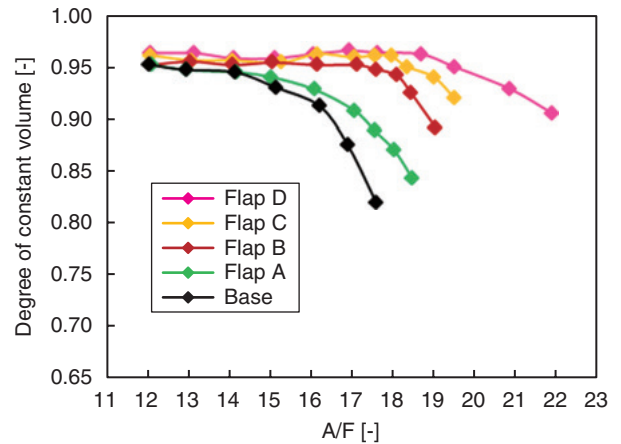


Fig. 12 Relationship between A/F and degree of constant volume

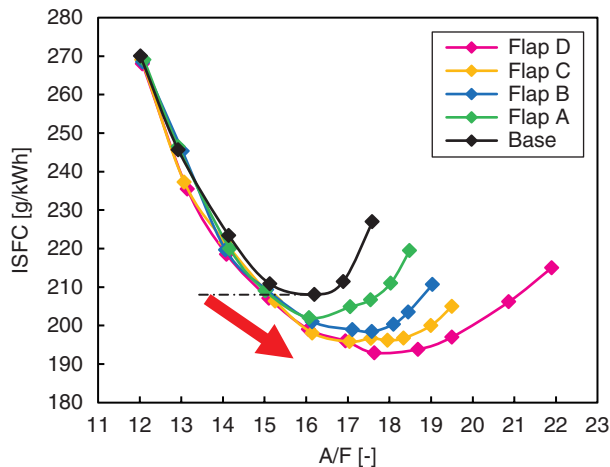


Fig. 9 Relationship between A/F and ISFC

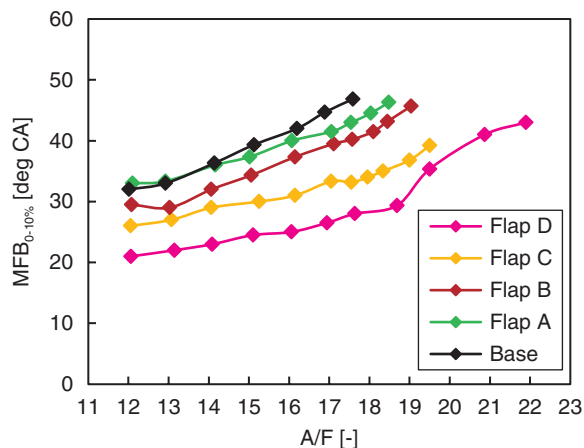


Fig. 10 Relationship between A/F and $MFB_{0-10\%}$

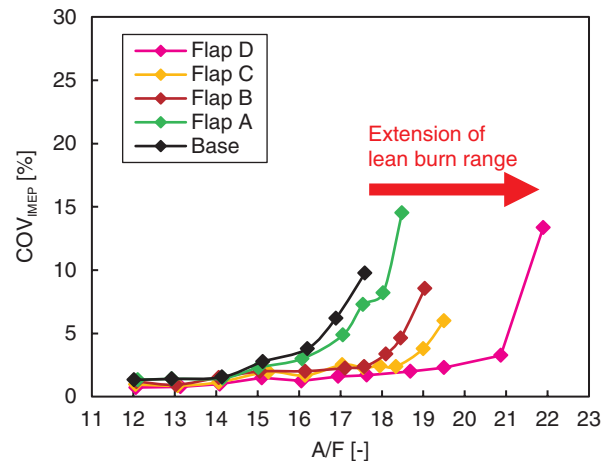


Fig. 13 Relationship between A/F and COV_{IMEP}

using the lowest ISFC when each flap was equipped in Fig. 9. It could be understood that the ISFC lowering ratio was correlated to the POT TR, and we therefore concluded that the POT TR is an effective indicator of lowering of ISFC.

Figure 15 shows the relationship between the POT TR and the maximum power. Although the intensifying of tumble flow with a flap was effective in reduction of ISFC, as the Z_i was lowered at the same time, the maximum power decreased along with an increase of POT TR, which meant that there is a trade-off between the so-called fuel economy and the maximum power.

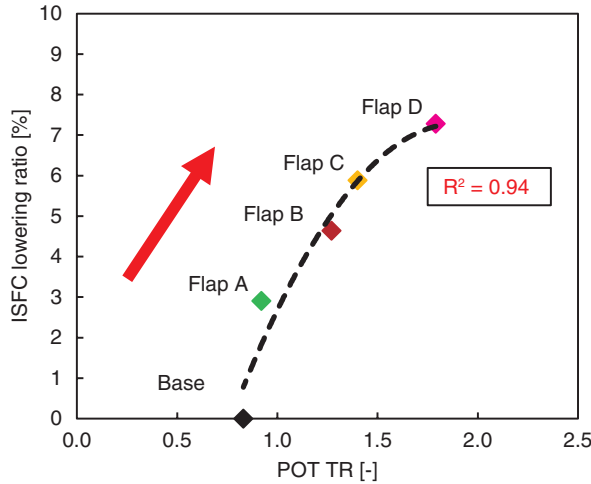


Fig. 14 Relationship between POT TR and ISFC lowering ratio

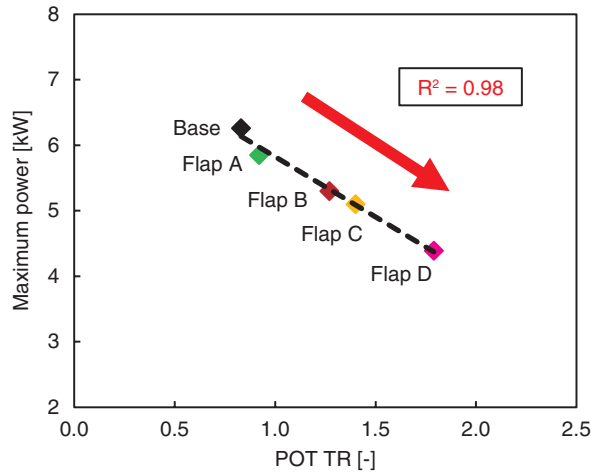


Fig. 15 Relationship between POT TR and maximum power

5. Strengthening of Tumble Flow and Prevention of Lowering of Maximum Power by Offsetting of Intake Valve

The lowering of maximum power due to reduced intake air volume was a concern when a flap was used. To cope

with this issue, we have invented a way to reduce lowering of intake air volume while controlling flow of intake air from the cylinder sleeve side of the intake valve.

5.1. Mechanical Arrangement of Intake Valve Offset

We worked out a new way to intensify tumbling while reducing lowering of maximum power output.

In an attempt to intensify tumble flow while reducing lowering of maximum power, we created a new way to realize vigorous tumble flow. Figure 16 shows the mechanical arrangement. The new design features a part of the intake valve jutting out beyond the perimeter of the cylinder bore. A recess was created by machining the top of the cylinder sleeve on the intake valve side. The recess prevented interference with the intake valve. The flow of mixture into the cylinder from the cylinder sleeve side was disturbed by masking effects from the recess. From these effects, vigorous tumble flow identical to fitting a stationary tumble generator flap was induced. With the intake valve located outside, the clearance between the intake valve and the exhaust valve increases, which allows enlarging of the diameter of the intake valve, preventing lowering of maximum power. This method is hereafter referred to as “intake valve offset.”

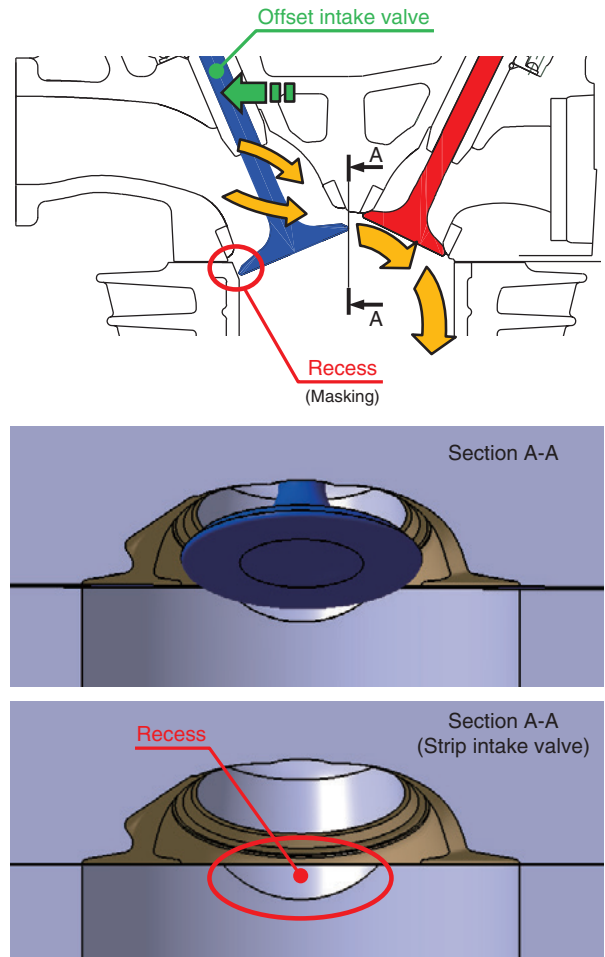


Fig. 16 Offset intake valve design

The parameters other than the intake valve were fixed when designing the amount of intake valve offset and increasing valve diameter. Preconditions when designing are shown as follows.

Compared to the base engine:

- (1) No change in valve included angle
- (2) No change in exhaust valve layout
- (3) No change in piston ring location (height)
- (4) No change in valve timing & valve lift
- (5) Relocation of intake valve should be parallel to cylinder head gasket surface and toward outside of cylinder bore

Under the above-described preconditions, the intake valve offset was determined by the following steps. The amount of intake valve offset is interrelated to how far away the intake valve is relocated and the intake valve diameter. Accordingly, how far away the intake valve should be relocated was determined first. Then, an extension of the intake valve diameter was considered.

The details of designing of intake valve offset are as follows.

- (1) How far away the intake valve should be relocated

To attain maximum masking effects from the intake valve and the cylinder sleeve, the recess was designed to allow for maximum intake valve relocation. The recess was set at the location that allows retention of the minimum necessary space [Fig. 17(1)] between the bottom of the recess and the top of the top ring when the piston was at top dead center. After that, the clearance between the intake valve and the recess was determined at the minimum necessary amount to prevent interference of both parts in order to obtain maximum masking effects [Fig. 17(2)].

- (2) Extension of intake valve diameter

When the clearance between the intake valve and the exhaust valve was the same as in the basic design, it was possible to extend the intake valve diameter by a maximum of 3 mm from the basic design as per (1). It should be noted, however, that in this instance, as a step occurred on the combustion chamber wall [Fig. 17(3)], the flow of drawn mixture was disturbed in the exhaust valve side by the step, causing possible lowering of Z_i or TR . Such a step increased along with an extension of intake valve diameter.

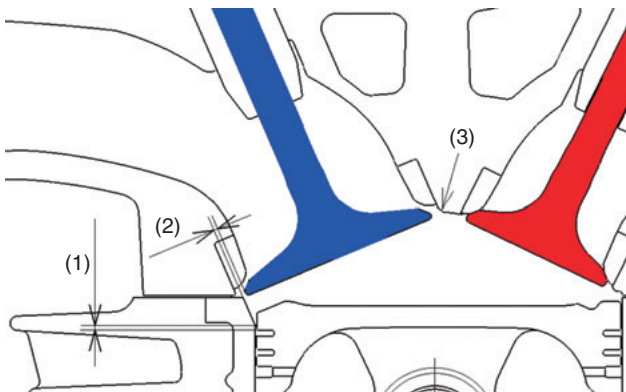


Fig. 17 Steps to determine intake valve offset

Therefore, allowable intake valve diameter free of negative impacts was investigated by CFD analysis. Thus, the intake valve diameter was finally extended by 2 mm. At that time, the offset of the intake valve was 2.3 mm from that of the basic model.

Figure 18 shows the combustion chambers of the base engine, one with the intake valve diameter increased by 2 mm by offsetting of the intake valve, and the other with the intake valve diameter increased by 3 mm by offsetting of the intake valve.

5.2. Analysis for Effects of Intake Valve Offset by CFD

The impact of intake valve offset on POT TR and Z_i was analyzed using CFD.

First, the N_r per valve lift was calculated at a throttle opening of 15 degrees. Figure 19 shows the results. When the intake valve was offset, the N_r increased in the full range

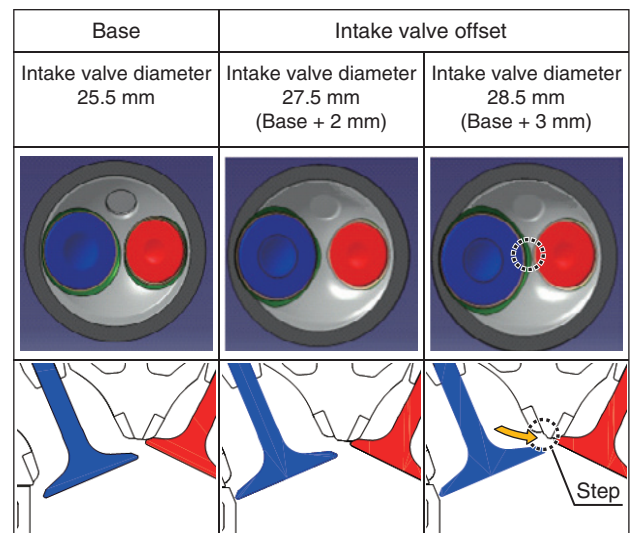


Fig. 18 Combustion chamber with intake valve offset

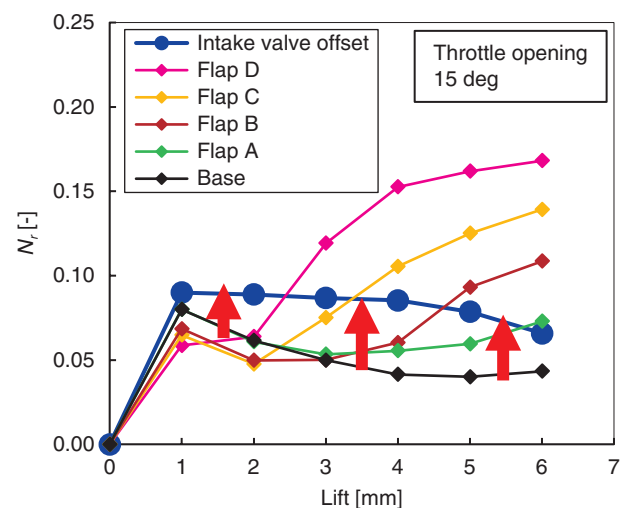


Fig. 19 Relationship between valve lift and N_r

of valve lift when compared to the basic design. Also, with the intake valve offset, the N_r tended to increase in the range where the valve lift was low compared to the flap. This was because the recess was placed close to the intake valve seat, allowing achievement of greater tumble effect than the flap in the low valve lift range.

Next, the C_f per valve lift was calculated at WOT. Figure 20 shows the results. When the C_f of intake valve offset and that of flap B were compared, the C_f values were nearly the same when the valve lift is less than 4 mm. When the valve lift was more than 4 mm, however, the C_f value was higher in the intake valve offset than in flap B. This was because the masking effect was reduced in the high valve lift range when intake valve offset was applied. As a result, C_f increased.

Figure 21 shows the relationship between valve lift and Z . When Z in the offset intake valve was compared to that of flap B, Z in the offset intake valve tended to increase relative to Z of flap B more than in the comparison of C_f . As indicated by the preceding Eq. (5), Z is defined as a product

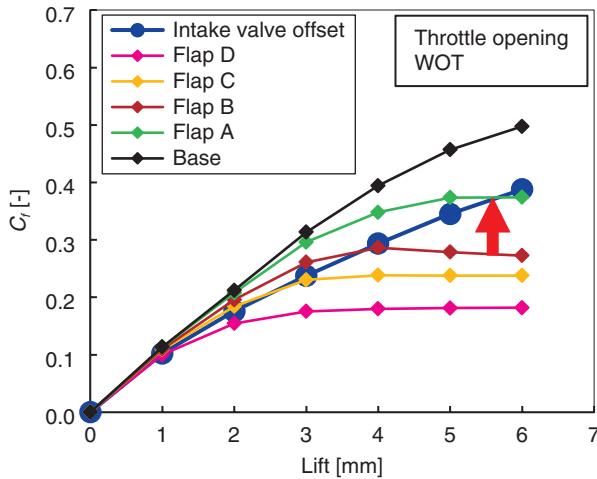


Fig. 20 Relationship between valve lift and C_f

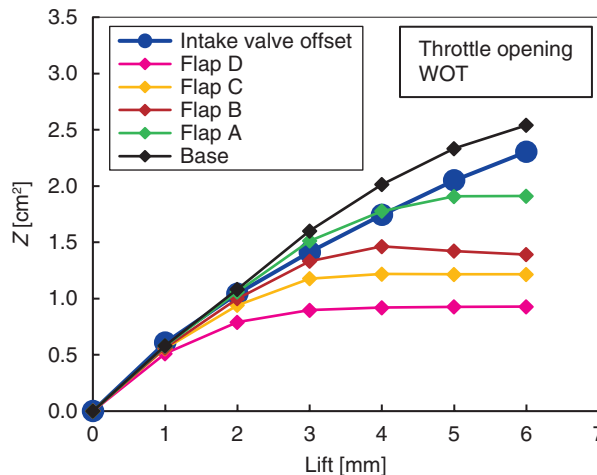


Fig. 21 Relationship between valve lift and Z

of C_f and A_v . In the intake valve offset design, as the intake valve diameter was increased by 2 mm from the basic design, A_v increased, resulting in an increase of Z .

TR and Z_i are calculated from N_r , C_f and Z . Figure 22 shows the results. In the intake valve offset design, POT TR of 1.3, which is the same level as in flap B, was attained. Also, in the intake valve offset design, the Z_i at WOT was more than in flap B although it was lower than in the basic design. Thus, the new design was expected to allow an escape from the trade-off between fuel economy and maximum power.

In this connection, when only offsetting of the intake valve was applied without increasing the intake valve diameter, Z_i decreased by approximately 8% from $6.67 \text{ cm}^2 \cdot \text{rad}$ to $6.15 \text{ cm}^2 \cdot \text{rad}$, although POT TR did not decrease. Figure 23 shows the results of CFD analysis of flow velocity distribution in the intake port and in the

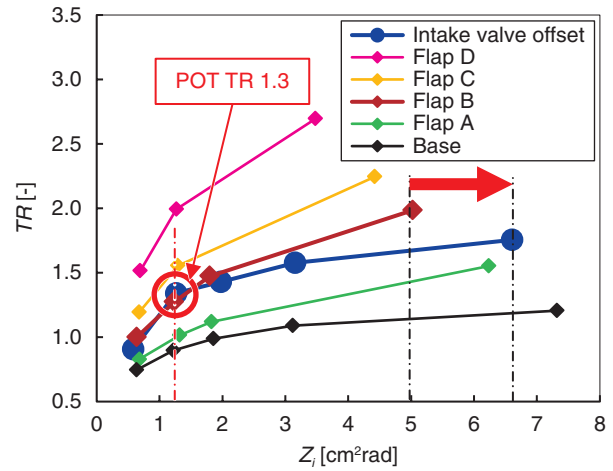


Fig. 22 Relationship between Z_i and TR

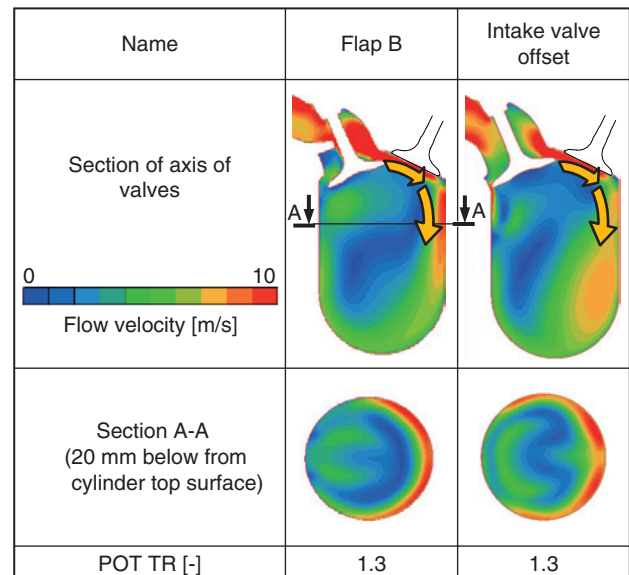


Fig. 23 Flow velocity distribution analyzed by CFD

cylinder. The calculation was conducted at a throttle opening of 15 degrees and valve lift of 4.5 mm, which was where N_v of intake valve offset intersects with that of flap B in Fig. 19. Due to the masking effects from the intake valve offset, the flow of mixture from the cylinder sleeve wall side of the intake valve was disturbed, and the flow from the exhaust valve side to the cylinder became vigorous. Thus, the intake valve offset induced vigorous tumble flow in the cylinder, allowing attainment of the same POT TR as with flap B.

5.3. Fuel Economy Improvement and Prevention of Lowering of Maximum Power

Using the engine incorporating intake valve offset, fuel economy improvement and prevention of lowering of maximum power were verified by experiments. Figure 24 shows the relationship between A/F and ISFC under the typical operating conditions in IDC. Figures 25, 26, 27 and 28 show the results of combustion analyses.

The lowering of ISFC in the lean range was confirmed as the effect of intake valve offset, similar to the case of fitting of a flap in the intake port. When compared with the

minimum value of ISFC, the rate of lowering was 4.8%. This rate of lowering of ISFC was equivalent to that of flap B having almost equal POT TR value. The results of combustion analysis showed the shortening of $MFB_{0-10\%}$ and

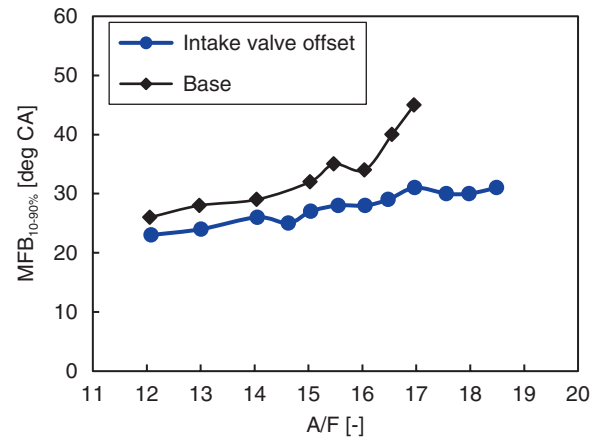


Fig. 26 Relationship between A/F and $MFB_{10-90\%}$

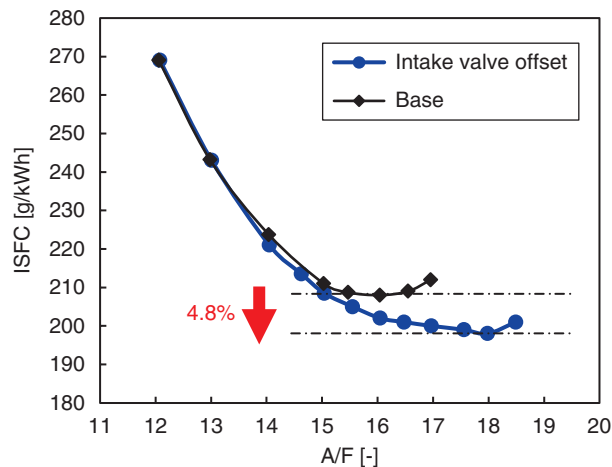


Fig. 24 Relationship between A/F and ISFC

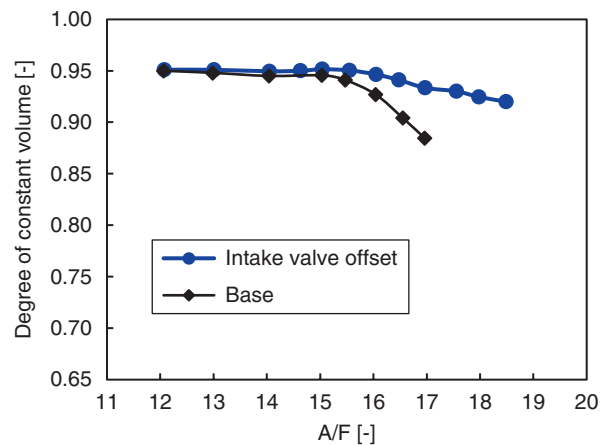


Fig. 27 Relationship between A/F and degree of constant volume

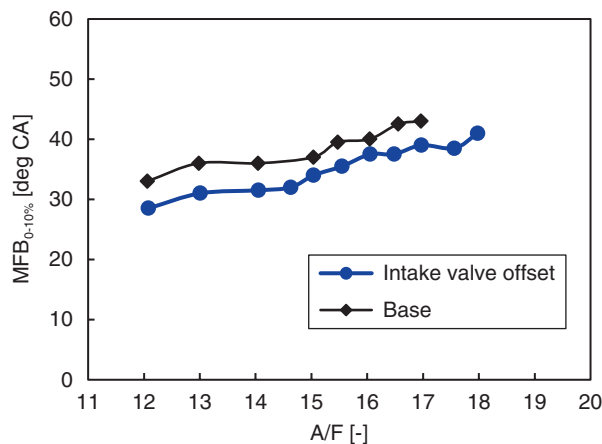


Fig. 25 Relationship between A/F and $MFB_{0-10\%}$

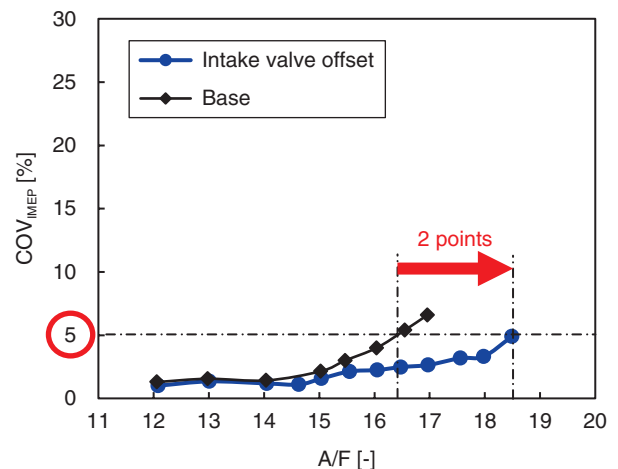


Fig. 28 Relationship between A/F and COV_{IMEP}

MFB_{10-90%} and the increase of degree of constant volume in the engine incorporating intake valve offset. When A/F where COV_{IMEP} reached 5% was defined as a lean burn limit, the lean burn limit of intake valve offset shifted 2 points to the lean side compared to the base engine.

Figure 29 shows power output at wide open throttle. The drop in maximum power of the engine was suppressed to 4% when intake valve offset was applied, while it declined by 15% compared to the base engine when flap B was used.

Figure 30 shows the relationship between POT TR and maximum power. It can be observed that the point showing

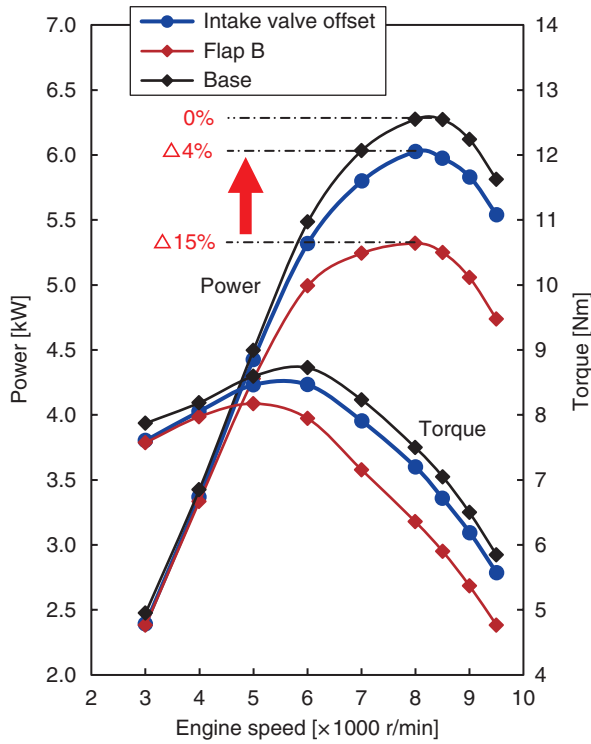


Fig. 29 Power output at wide open throttle

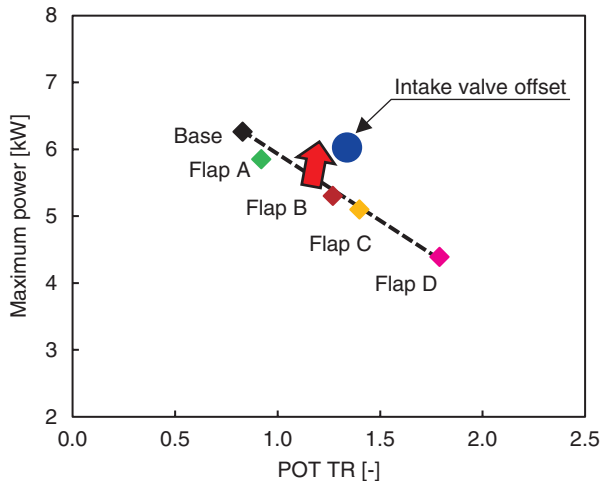


Fig. 30 Relationship between TR and maximum power

the status with application of intake valve offset lies outside the trade-off line between POT TR and maximum power when different shapes of flap were equipped.

Consequently, it was confirmed that the trade-off relationship between ISFC and maximum output power was mitigated without using complicated mechanisms, like variable devices, by applying this technique.

6. Control of Knocking by Squish and Improvement of Fuel Economy from Increased Compression

The following have been analyzed through CFD analysis and actual tests of two-valve per cylinder engines.

- Squish zone shape that is effective for creation of reversed squish flow
- Controlling of knocking by squish
- Fuel economy from increased compression realized by control of knocking

Figure 31 shows the specifications of the test engine. The rate of projected squish surface area indicates the rate of squish surface area against the bore surface area viewed from the cylinder head gasket side.

6.1. Intensity of Reversed Squish Flow from Various Squish Shapes and Impacts to Margin to Knocking Occurrence

To clarify the relationship between the reversed squish flow, which is considered effective for control of knocking, and the shape of squish zone, we conducted CFD analysis. Reversed squish flow occurs when the piston descends after passing TDC. Figure 32 shows the in-cylinder pressure waveform when knocking occurs in the actual engine (wide open throttle, Ne 3000 r/min, 50 cycles). Knocking occurred at the crank angle of 5-10 degrees ATDC. In view of this, we chose the crank angle of 5 degrees ATDC for assessment of intensity of reversed squish flow by CFD analysis.

		Base	Sq-A	Sq-B
Squish shape	Plug side	NA	Taper	Slant-parallel
	Opposite side	Taper	Slant-parallel	Slant-parallel
Squish height [mm]		1.3	0.8	0.8
Projected area of squish [%]		12.3	26.9	26.9
Compression ratio		9.0	9.0	9.0
Cross sectional view (Section A-A)				
Squish shape				

Fig. 31 Specifications of squish

Figure 33 shows the results of CFD analysis. The vector shows the direction and intensity of the flow.

The differences in the flow speeds of the reversed squish between the tapered squish and the slant-parallel squish were compared at the plug side and at its opposite side. In the opposite side of the spark plug, the speed of reversed squish flow that enters the squish zone was higher in the area of the slant-parallel shape (Sq-A and Sq-B) than in that of the tapered shape (basic design). In addition, in the spark plug side, the speed of reversed squish flow was higher in the area of the slant-parallel shape (Sq-B) than in that of the tapered shape (Sq-A). From the above-mentioned findings, it was determined that the slant-parallel squish is effective to intensify reversed squish flow.

To clarify the influence of intensity of reversed squish flow on knocking, the ignition timing when knocking occurred and the MBT were measured through experiments. Here, the variance of crank angle between the ignition timing when knocking occurs and the MBT is defined

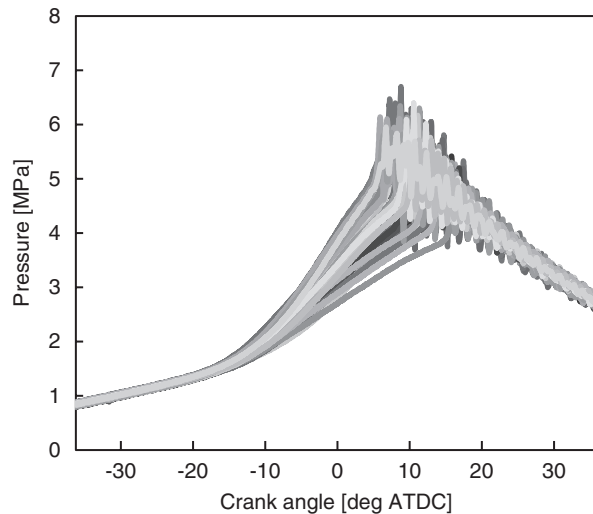


Fig. 32 Pressure waveform with knocking occurrence

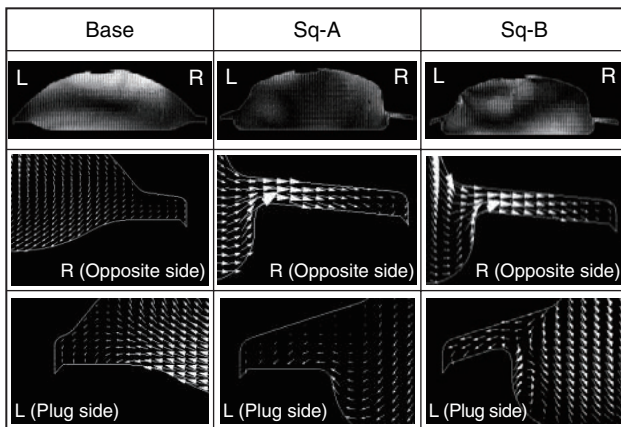


Fig. 33 Effect of squish area location on squish flow formation (5 deg ATDC CA)

as “margin to knocking occurrence” for assessment of knocking control.

Figure 34 shows the ignition timing when knocking has occurred, Fig. 35 shows MBT, and Fig. 36 shows margin to knocking occurrence. It can be understood from Fig. 34 that the ignition timing when the knocking occurred

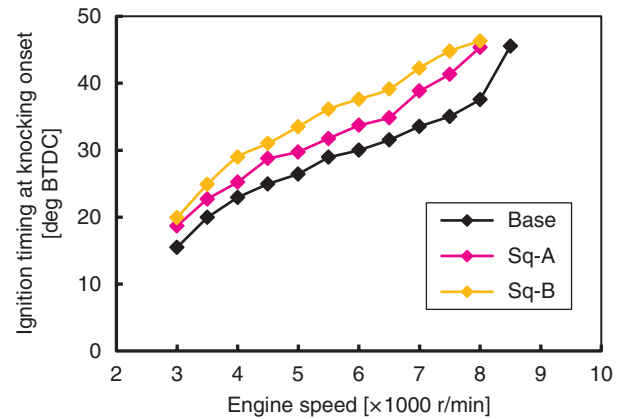


Fig. 34 Effect of squish on ignition timing at knocking onset

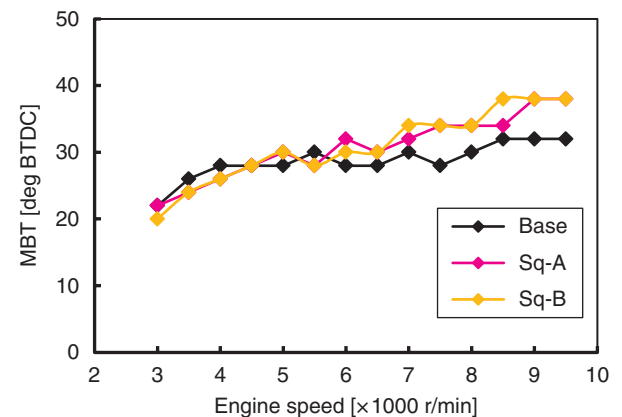


Fig. 35 Effect of squish on MBT

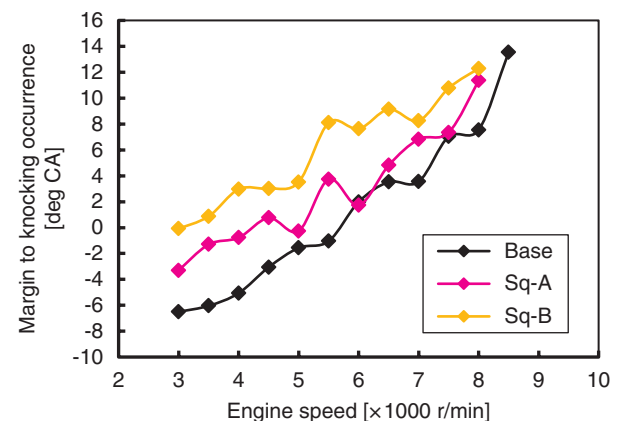


Fig. 36 Effect of squish on margin of knocking occurrence

advanced in the order of basic design, Sq-A, Sq-B along with the intensifying of reversed squish flow. Since there was no large difference in MBT as shown in Fig. 35, the margin to knocking occurrence increased in the order of Sq-A, Sq-B when compared to the basic design as Fig. 36 shows.

The following has been found from the application of squish to the two-valve per cylinder, small displacement engine.

- The slant-parallel squish is effective for intensifying of reversed squish flow
- The intensity of reversed squish flow is correlated to the margin to knocking occurrence.

6.2. Increase of Compression Ratio to Take Advantage of Squish Effect

We attempted to increase compression ratio to take advantage of the margin to knocking occurrence from squish effects. The compression ratio ε for the same margin to knocking occurrence as the basic design was 9.5 (0.5 higher than the basic design). The compression ratio was raised by changing the piston crown height without altering the squish shape from Sq-B. This design is defined as Sq-B_($\varepsilon+0.5$). The ignition timing when the knocking occurred, the MBT and the margin to knocking occurrence of Sq-B_($\varepsilon+0.5$) are shown respectively in Figs. 37, 38, and 39. Figure 37 shows that the ignition timing when the knocking occurred in Sq-B_($\varepsilon+0.5$) was retarded from that of Sq-B, and is the same as that of the basic design. Also, Fig. 38 shows that the MBT was the same as that of the basic design and Sq-B. Accordingly as Fig. 39 shows, the margin to knocking occurrence was lower than Sq-B, and the same as in the basic design.

6.3. Changes of Heat Generation Rate Caused by Squish

To clarify factors that increase margin to knocking occurrence by the application of squish, we conducted combustion analysis. (Analysis conditions: Engine speed: 6000 r/min; Throttle: wide open throttle; A/F 12.4)

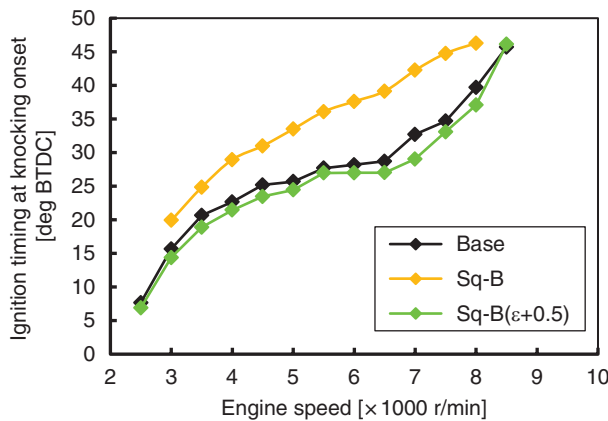


Fig. 37 Effect of compression ratio on ignition timing at knocking onset

Figure 40 shows the rate of heat release and the in-cylinder pressure at various crank angles. The Pmax of Sq-B with the increased margin to knocking occurrence was the same as that of the basic design. When the rate of

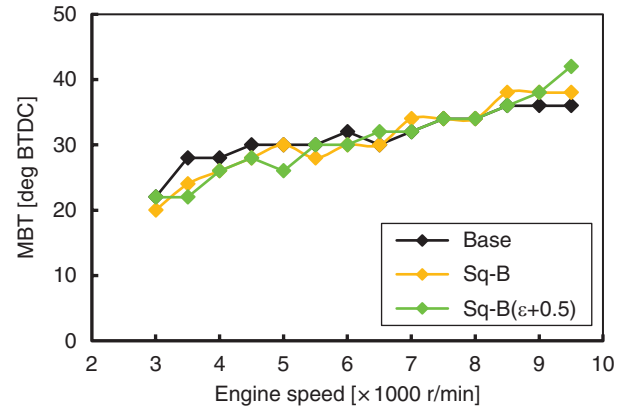


Fig. 38 Effect of compression ratio on MBT

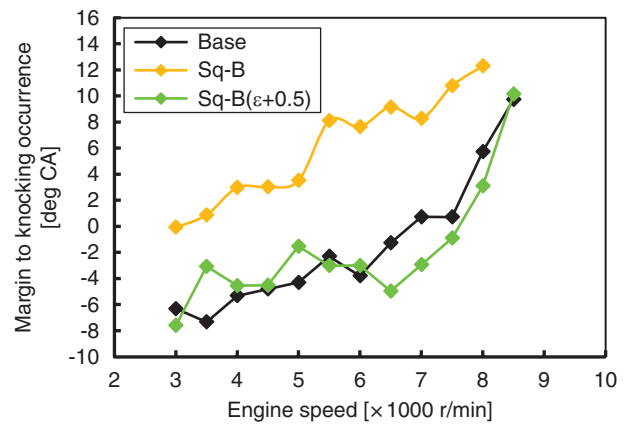


Fig. 39 Effect of compression ratio on margin to knocking occurrence

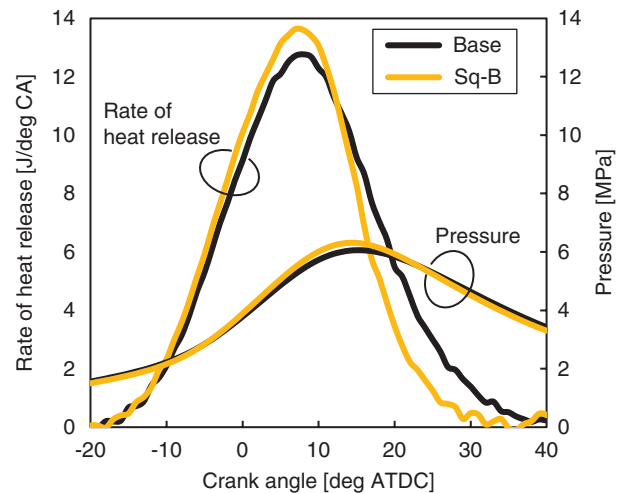


Fig. 40 Comparison of rate of heat release and pressure

heat release was compared, the rate of heat release of Sq-B was higher than in the basic design near the crank angle of 5 degrees ATDC where CFD analysis shows differences in the intensity of reversed squish flow. It can be considered that the accelerated combustion in that period from the squish effect is a factor of the increased margin to knocking occurrence.

6.4. Improvement of Fuel Economy by Control of Knocking Using Squish

Figure 41 shows the ISFC and the $MFB_{50-90\%}$ in the latter combustion period in Sq-B and Sq-B_(ε+0.5) at A/F of 16.5, where the ISFC of the basic design became lowest under the partially loaded conditions. Compared to the basic design, the ISFC lowered by 1.3% in Sq-B, and the $MFB_{50-90\%}$ was shortened by 5 degrees (from 24 degrees to 19 degrees). In Sq-B_(ε+0.5), the ISFC was lowered by 2.6% from the basic design, and the $MFB_{50-90\%}$ was shortened by 5 degrees (from 24 degrees to 19 degrees). It can be considered that the ISFC of Sq-B lowered because of the shortened $MFB_{50-90\%}$. Also, the ISFC of Sq-B_(ε+0.5) lowered because of the improvement of theoretical thermal efficiency due to the increased compression ratio in addition to the shortening of $MFB_{50-90\%}$.

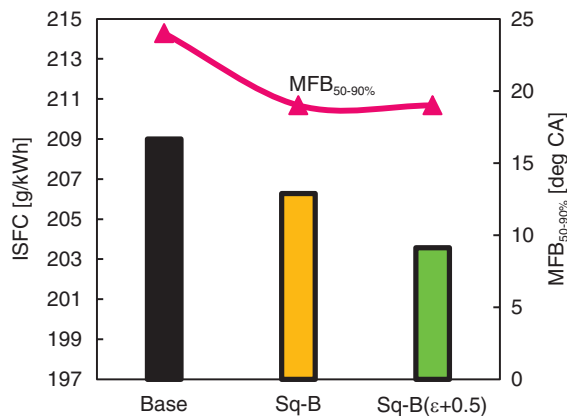


Fig. 41 Comparison of ISFC and $MFB_{50-90\%}$

7. Effectiveness of Combination of Offset Intake Valve and Squish

As mentioned above, the offset intake valve has extended the lean burn range by the intensifying of tumble effects. Also, the squish allowed increase of compression ratio by intensified reversed squish flow. With these concepts combined, the performance was investigated using the specs shown in Fig. 42.

Figure 43 shows COV vs A/F. In the intake valve offset Sq-B_(ε+0.5), the lean burn range was extended by 2 points in A/F from the basic design just like the offset intake valve. Figure 44 shows the ignition timing when the knocking occurred, and Fig. 45 shows the MBT. The angles were

		Base	Intake valve offset Sq-B _(ε+0.5)
Intake valve offset		NA	Applied
Intake valve diameter [mm]		25.5	27.5
Squish shape	Plug side	NA	Slant-parallel
	Opposite side	Taper	Slant-parallel
Squish height [mm]		1.3	0.8
Compression ratio		9.0	9.5
Cross sectional view (Section A-A)			
Squish shape			

Fig. 42 Specifications of intake valve offset Sq-B_(ε+0.5)

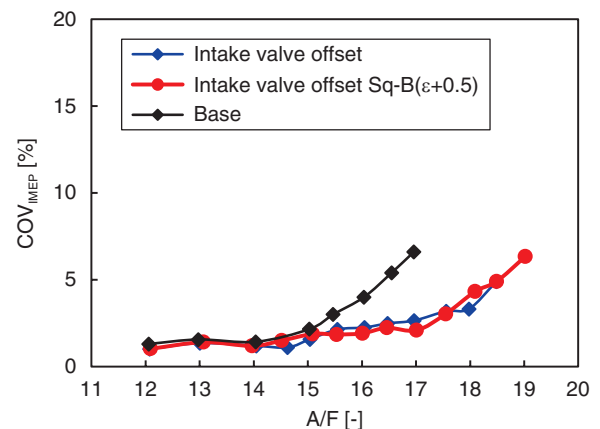


Fig. 43 Relationship between A/F and COV_{IMEP}

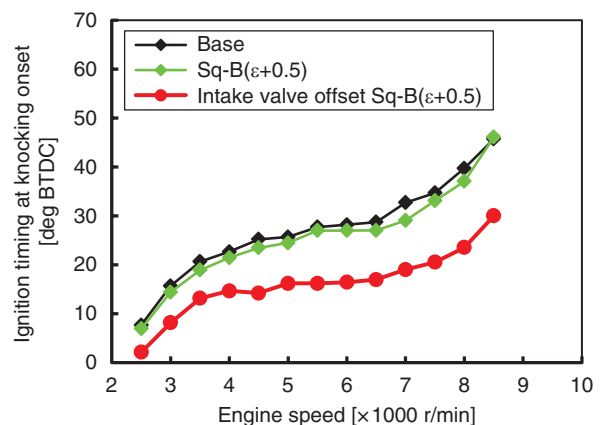


Fig. 44 Comparison of ignition timing at knocking onset

retarded from the basic design both in the ignition timing when the knocking occurred and in the MBT. As Fig. 46 shows, the margin to knocking occurrence of the intake valve offset $Sq-B_{(\varepsilon+0.5)}$ was the same as in the basic design. It is considered that the reason why the ignition timing when the knocking occurred and the MBT in the intake valve offset $Sq-B_{(\varepsilon+0.5)}$ were retarded from the basic design was the shortening of the principal combustion period $MFB_{10-90\%}$ in the offset intake valve design due to the intensified tumbling as shown in Fig. 26.

From the above findings, it has been clarified that effectiveness of both ideas is individually attainable even when the combustion improvement from extension of lean burn range by off-setting of the intake valve is combined with that from increased margin to knocking occurrence by the squish effects.

Figure 47 shows the power output when the throttle is opened widely. Figure 48 shows ISFC vs A/F. Although the maximum power output in the intake valve offset $Sq-B_{(\varepsilon+0.5)}$ decreased by 3% from the basic design, the minimum ISFC has been lowered by 6.0%.

Figure 49 shows the relationship between the lowering

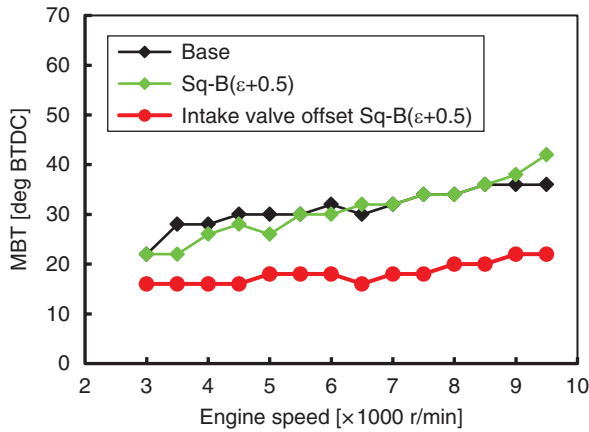


Fig. 45 Comparison of MBT

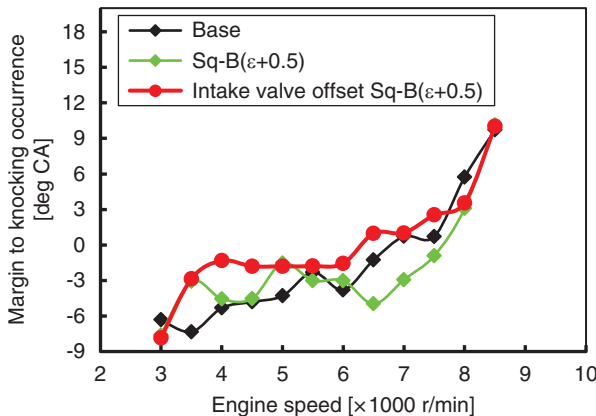


Fig. 46 Comparison of margin to knocking occurrence

ratio of ISFC and the maximum power output under the partially loaded conditions. Each rate of lowering ratio of ISFC when the four kinds of tumble generator flap were equipped was calculated from Fig. 12. If the 6.0% reduction

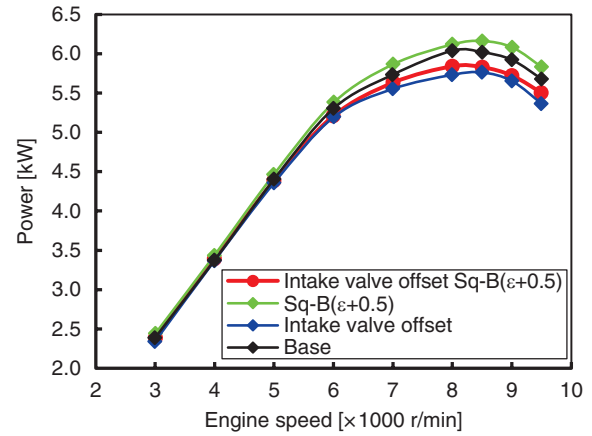


Fig. 47 Power output at wide open throttle

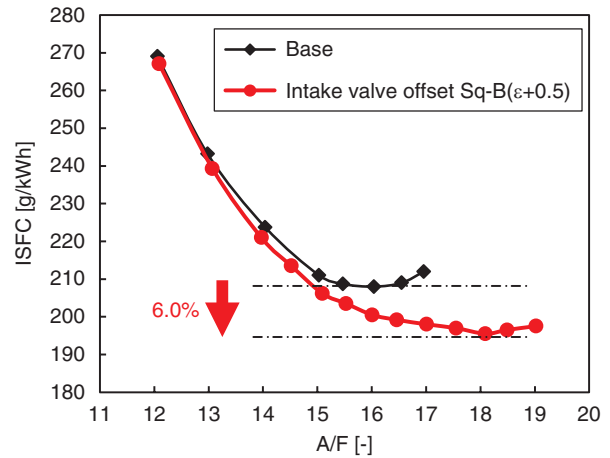


Fig. 48 Relationship between A/F and ISFC

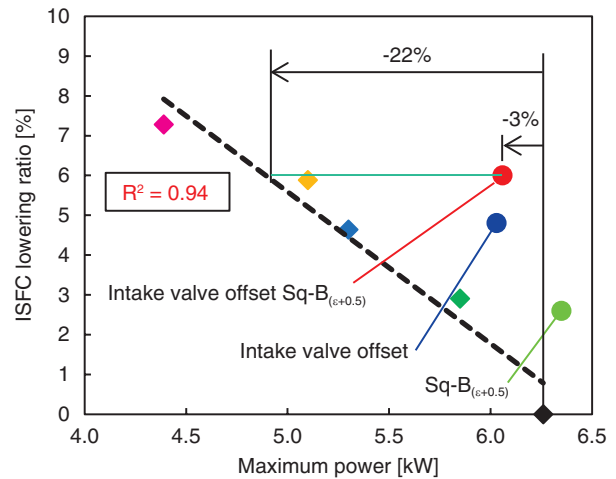


Fig. 49 Relationship between maximum power and ISFC lowering ratio

of ISFC realized by the intake valve offset $Sq-B_{(ε+0.5)}$ were attained only by use of the flap, the maximum power output would be lowered by 22% (from 6.3 kW to 4.9 kW). When the intake valve offset $Sq-B_{(ε+0.5)}$ was used, however, the decrease in maximum power output was only 3% (from 6.3 kW to 6.1 kW).

Based on the results of combined measures mentioned above, it was confirmed that the trade-off relationship between ISFC and the maximum output power could be successfully mitigated without using complicated mechanisms such as variable devices.

8. Conclusion

The extension of lean-burn range by offset intake valve and increased compression ratio using squish have improved ISFC while preventing reduction of the maximum power output.

- (1) It has been clarified that the POT TR is correlated to the rate of ISFC reduction in the partially loaded conditions, which are subjected to these studies. Thus, it has been determined that the POT TR can serve as an index of ISFC reduction.
- (2) The following two principles, which have been clarified in the four-valve per cylinder engine for automobiles are also confirmed in the small displacement, two-valve per cylinder engine for motorcycles.
 - (i) The intensity of reversed squish flow is correlated to the control of knocking.
 - (ii) The slant-parallel squish is effective to intensify reversed squish flow.
- (3) With the intake valve off-set, the POT TR increased, and the lean burn limit has been extended by two points in A/F compared to the basic design. Thus, the ISFC under the partially loaded conditions has been reduced by 4.8% from the basic design.
- (4) The slant-parallel squish controls knocking, allowing an increase of compression ratio (from 9.0 to 9.5), which has reduced the ISFC by 2.6% from the basic design under the partially loaded conditions.
- (5) By the combination of the offset intake valve mentioned in (3) and the increased compression ratio using the slant-parallel squish mentioned in (4), the minimum ISFC under the partially loaded conditions has been reduced by 6.0% from the basic design while keeping the decrease in maximum power output to as little as 3% (from 6.3kW to 6.1kW).

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■ Author ■



Takamori SHIRASUNA



Hideki SAITO



Tomokazu NOMURA