

# Intake Design for Reduction of Duration of Combustion

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## ABSTRACT

Intake design has been conducted as part of research seeking to realize a level of thermal efficiency of 45% in gasoline engines in order to reduce the duration of combustion, which had been extended by EGR dilute combustion. First, the relationship between burning velocity (m/sec) and the square root (m/sec) of turbulent kinetic energy ( $\text{m}^2/\text{sec}^2$ ) was studied, and it was determined that these parameters were effective in reducing the duration of combustion. Next, the fact that differences in turbulent kinetic energy occurred due to the direction of the flow passing the intake valve even in the case of flow motion with the same tumble ratio was focused on, and specifications were selected in order to make it possible to obtain a higher level of turbulent kinetic energy from intake ports displaying the same tumble ratio.

A prototype engine head was manufactured using the selected intake port specifications, and test results showed that the target of a main combustion duration of no more than 20 deg was realized under conditions of dilute combustion at an external EGR rate of 30%.

## 1. Introduction

Today, the realization of increased fuel efficiency and the reduction of exhaust emissions are demanded of automotive manufacturers as part of efforts to reduce the burden on the environment, and associated regulation values are tending to increase in stringency year by year. It is considered that the internal combustion engine will continue to be employed as the source of drive power for motor vehicles for the foreseeable future<sup>(1)</sup>, and the realization of increased thermal efficiency in automotive engines is an important research agenda. Honda has therefore been engaged in research seeking to increase the current maximum level of gasoline engine thermal efficiency of around 39% to 45% or higher<sup>(2)</sup>.

In the course of this research, the results of tests using engines on the market were employed in order to study the necessary conditions for the realization of a level of brake thermal efficiency of 45%. Based on the results of these investigations, the conditions were set at Minimum advance for Best Torque (MBT) operation at a compression ratio of 15 or higher, an absolute boost pressure of 2.5 bar or higher, and an external exhaust gas recirculation (EGR) rate at the 30% level. At the same time, the issue in terms of combustion was to attempt to restrict the main combustion duration during which the mass burn fraction (*MBF*) increases from 10% to 90%,  $MBF_{10-90\%}$ , to a crank angle of no more than 20 deg.

## 2. Research Issues and Approaches

The purpose of the research discussed in this paper was to design an intake that would contribute to the reduction in the combustion period necessary for the realization of a level of brake thermal efficiency of 45%.

During high-EGR combustion, an increase in the quantity of inert gas results in a decline in the laminar burning velocity ( $S_L$ ), presenting the issue of an increase in the combustion period. In addition, because parameters including the amount of intake air, the equivalent ratio, gas temperature, and pressure are already determined, and  $S_L$  is determined by these physical values, restoring  $S_L$  to its original value following its decline is considered challenging. The present development therefore focused on restoring the decline in burning velocity by reducing  $S_L$  to its original value through the realization of increased turbulent burning velocity ( $S_T$ ). This was achieved by increasing the turbulent kinetic energy (*TKE*) of the air-fuel mixture.

Turbulence of the air-fuel mixture is obtained through a process of transition to small-scale operation with the collapse of the bulk flow during the compression stroke. The bulk flow focused on in the present research was a tumble flow, and in this case it is normal to strengthen the tumble flow formed by the intake port or increase the ability to maintain the tumble flow by modifying the shape of the tops of the pistons as measures to increase *TKE* during combustion.

However, this presents an issue. The use of an intake

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port shape that forms a strong tumble flow results in an increase in intake pressure loss. In addition, despite the fact that the tumble ratio is increased in measurements conducted in steady-state flow tests, during actual engine operation, the air-fuel mixture taken into the cylinders flows back to the intake port, and it is not possible to increase the magnitude of the tumble formed in the combustion chamber above a fixed level. As a result, there is a limit to the increase in burning velocity that can be achieved by means of increasing the tumble ratio. Knocking becomes severe in engines employing a high compression ratio. In order to respond to this issue, the research under discussion incorporated the Atkinson cycle in its engine concept, reducing the effective compression ratio by retarding the timing of closure of the intake valve. However, under these conditions the tumble ratio able to be formed in the combustion chamber is lower than the tumble ratio for a standard valve timing.

Because of this, in addition to the generation of a more powerful tumble flow, the achievement of a higher *TKE* from a flow that maintained the same tumble ratio represented another issue for the research.

In general, research focuses on the shape of the combustion chamber, and studies are conducted in order to determine which shapes result in adequate retention of the tumble vortex following its formation<sup>(3), (4)</sup>. Little study has been directed towards the retention of the tumble flow during the intake stroke or the degree of dissipation of kinetic energy during the process of formation of the tumble flow. During the present research it was observed that differences sometimes occurred in the *TKE* obtained for the same combustion chamber shape at the same tumble ratio, indicating that the process of formation of the intake air flow was an important factor. A study was therefore conducted that focused on the direction of the intake air flow.

Using the findings obtained, 3D-CFD was employed in order to determine the optimum intake port design, and a prototype engine head was manufactured. The results of tests conducted to verify that the target reduction in the duration of combustion had been achieved demonstrated the validity of the results of the study.

### 3. Engine Specifications and Indicator for Evaluation of Combustion

#### 3.1. Main Specifications of Engines employed in Study

Table 1 shows the specifications of the engines employed in studies of tumble ratio, strength of turbulence, and duration of combustion.

#### 3.2. Method of Evaluation of Tumble Ratio

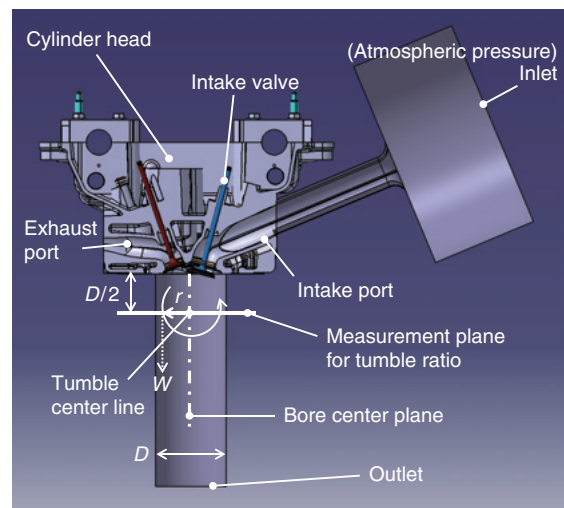
VECTIS, the 3D-CFD software manufactured by Ricardo Software, was the main tool used in the evaluation of the tumble ratio. Standard *k-ε* was employed as the turbulence model, and non-steady-state calculations were conducted until the inlet and outlet flow rates converged.

Figure 1 shows a schematic diagram of the relationships

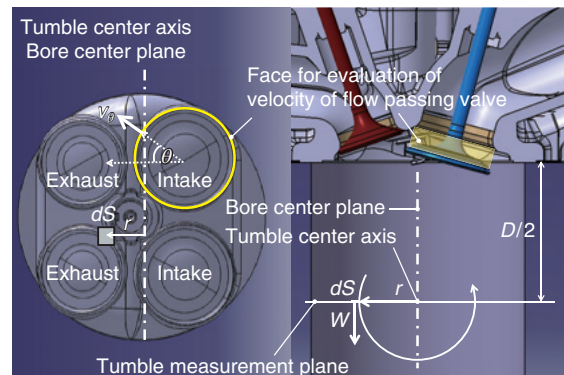
Table 1 Engine specifications

| Designation                | A                 | B          |
|----------------------------|-------------------|------------|
| Engine type                | 4-valve SI engine |            |
| Number of cylinders        | 1                 | 4          |
| Bore × Stroke [mm]         | 81 × 121.6        | 73 × 109.6 |
| Intake valve diameter [mm] | 30                | 28         |
| Fuel supply                | PI                | DI         |

between important positions in the entrance region used in calculations. The pressure differential before and after the intake valve (the pressure differential between Inlet and Outlet in the figure) was set at 6 kPa, and the inlet pressure was set at atmospheric pressure. In relation to the bore diameter, *D*, the measurement plane was positioned *D*/2 mm below the cylinder head gasket plane. Taking the bore center plane including the crankshaft into consideration, the line that intersected the measurement plane was defined as the tumble center line. Angular velocity was calculated from the velocity components passing the measurement plane with the tumble center line as the central axis of rotation, and this angular velocity was used to calculate the tumble ratio<sup>(5)</sup>.



(a) Evaluation point and position of axis



(b) Method of calculation of tumble ratio and direction of intake flow

Fig. 1 Set-up for steady-state 3D-CFD and evaluation point

The infinitesimal area  $dS$  mm<sup>2</sup>, extending for  $r$  mm from the tumble center line in the direction running from the intake port to the exhaust port, was focused on, and the perpendicular downward velocity component was termed  $W$  mm/sec. The average velocity passing the measurement plane,  $S$ , was termed  $W_{ave}$  mm/sec. The angular velocity in the measurement plane,  $S$ , at this time, termed  $\omega$  rad/sec, is expressed by Eq. (1).

$$\omega = \int (W - W_{ave}) r^2 dS / \int r^2 dS \quad (1)$$

Assuming that  $W_{ave}$  and the average piston speed are identical, then the rotational angular velocity of the crank,  $\omega_{mot}$ , is expressed by Eq. (2).

$$\omega_{mot} = (\pi/L) W_{ave} \quad (2)$$

The tumble ratio when the intake valve lifts  $x$  mm,  $TR(x)$ , is defined by Eq. (3) as the ratio between the angular velocity of the fluid,  $\omega(x)$ , and the angular velocity of the crank as determined by Eq. (2).

$$TR(x) = \omega(x) / \omega_{mot} \quad (3)$$

The stroke values shown in Table 1 were employed in the calculation of the average piston speed.

Now, valve lift,  $x$  (CA) mm, instantaneous piston speed,  $pv$  (CA) mm/sec, and average piston speed,  $mpv$  mm/sec, at a crank angle of CA deg during engine operation are considered. With the crank angle when the intake valve is open and closed as  $CA_{IVO}$  and  $CA_{IVC}$  respectively, the average tumble ratio is defined based on these parameters by Eq. (4).

$$TR = \frac{1}{CA_{IVC} - CA_{IVO}} \int_{CA_{IVO}}^{CA_{IVC}} \frac{\omega[x(CA)]}{\omega_{mot}} \left[ \frac{pv(CA)}{mpv} \right]^2 dCA \quad (4)$$

### 3.3. Method of Evaluation of $TKE$

As in the case of the tumble ratio, VECTIS was used in the evaluation of  $TKE$  in the combustion chamber. Ricardo Software's 1D-CFD engine simulation code WAVE was employed to calculate boundary conditions. Boundary conditions at the upstream end of the intake manifold and the exhaust-facing cylinder head face were calculated using a model simulating an engine test environment.

Stoichiometric combustion at an engine speed,  $NE$ , of 2000 rpm was set as the operating condition. With regard to load, the point at which the indicated mean effective pressure (IMEP) reached 810 kPa was selected. When studying differences in combustion duration at the same laminar burning velocity, measurements were taken and calculations were conducted with the volumetric efficiency ( $\eta_v$ ) set at the level at which the IMEP was 810 kPa during MBT operation.

### 3.4. Indicator for Evaluation of Combustion

The timing at which the  $MBF$  was 50%, known as the combustion center of gravity and notated as  $MBF_{50\%}$ , was employed as the indicator for evaluation of combustion.

$MBF_{10-90\%}$  is often used in the evaluation of combustion duration, and is also used as a method of expressing target values. Because  $S_L$  and  $S_T$  are employed in studying the dimension of velocity (m/sec),  $MBF_{10-90\%}$  was also studied by converting it into the dimension of velocity using the bore diameter, as shown in Eq. (5). Here, this parameter is defined as  $S_C$ , the in-cylinder virtual flame velocity.

$$S_C = [(D/2) \times (0.9 - 0.1)] / MBF_{10-90\%} \quad (5)$$

A method of evaluation of the strength of turbulence is an important factor in increasing  $S_T$ . In the present research, the average flow velocity at a fixed point was determined, and the standard deviation of the magnitude of flow velocity (the rms value of change in velocity) was defined as the turbulence velocity ( $u'$ ). The relationship between  $TKE$ , which is often used to evaluate the strength or weakness of turbulent flows in relation to combustion in engines is  $TKE = 1.5u'^2$ , and as in the case of the discussion of  $MBF_{10-90\%}$ ,  $u'$  was employed for comparison with  $S_C$ .

### 3.5. Selection of Timing for Evaluation of Air-fuel Mixture

The  $TKE$  of the air-fuel mixture normally changes with time, and also differs depending on the area of the combustion chamber. In addition, research results show that the position of the flame front varies moment to moment, and that the scale of the eddies that can be used varies depending on the size of the flame front<sup>(6)</sup>. The modeling of all of this would entail a significant computation cost. In order to employ a simplified method and be selective with specifications to a certain extent, the present study used Engine A in Table 1 to determine whether it would be possible to use average values for the combustion chamber at specific timings to represent the situation overall.

Tests were conducted of the effect on  $MBF_{10-90\%}$  of varying the flow as a result of differences in the shape of the intake port or the timing of closure of the intake valve. The conditions discussed in Section 3.3. were used for  $NE$  and the load conditions. Five representative measurement points were selected for comparison: TDC; ignition timing; BTDC 30 deg, which represents the greatest difference to  $TKE$ ; the average value from BTDC 30 deg to ATDC 30 deg, which was assumed to represent the combustion duration; and the average value for the intake compression stroke. Figure 2 shows the relationship between the results and  $TKE$  in the combustion chamber as determined by CFD, and Table 2 shows the respective coefficients of correlation.

A variety of arguments exist regarding methods of evaluation of  $TKE$ . The present research found that  $TKE$  at BTDC 30 deg, which displayed the greatest difference with average  $TKE$  in the combustion chamber, had the strongest correlation with  $MBF_{10-90\%}$ . The coefficient of correlation between the two parameters was also sufficiently greater than 0.8. BTDC 30 deg was therefore adopted as the timing for evaluation of  $TKE$  in the present research. This value will be notated as  $TKE_{BTDC 30 \text{ deg}}$  below.

Table 2 Coefficient of correlation between  $TKE$  and  $MBF_{10-90\%}$

| Measurement point                       | Coefficient of correlation |
|-----------------------------------------|----------------------------|
| TDC                                     | -0.7341                    |
| Ignition timing                         | -0.7165                    |
| BTDC 30 deg                             | -0.9397                    |
| Average from BTDC 30 deg to ATDC 30 deg | -0.9214                    |
| Intake and compression stroke average   | -0.8538                    |

### 3.6. Definition of Flow Passing Valve

In order to consider the degree of turbulence and the direction of the intake flow, the flow passing between the intake valve and the valve seat was studied in the present research. This flow was termed the flow passing the valve, and its magnitude was defined as the flow velocity passing the valve. The positions for evaluation of this parameter are shown in Fig. 1. The intake port of the engine employed in the study was symmetrical with the center of the bore. The single side of the valve shown in Fig. 1(b) was studied. The flow velocity passing the valve in the area forming an angle of  $\theta$  rad with the directional axis from the intake valve in the direction of the exhaust valve, in the opposite direction to the cylinder center, was termed  $v_\theta$ .

The average flow balance ( $FB$ ) was defined as a scale to express whether the flow was directed inwards or outwards. Equations (6) and (7) respectively show the average velocity,  $v_{ave}(x)$  and the  $FB$  value,  $FB(x)$ , of the flow passing the valve at a valve lift of  $x$  mm,  $v_\theta(x)$ .

$$v_{ave}(x) = \frac{1}{2\pi} \int_{\theta=0}^{\theta=2\pi} v_\theta(x) d\theta \quad (6)$$

$$FB(x) = \frac{1}{\pi} \int_{\theta=0}^{\theta=\pi} \frac{v_\theta(x) - v_{-\theta}(x)}{v_{ave}(x)} d\theta \quad (7)$$

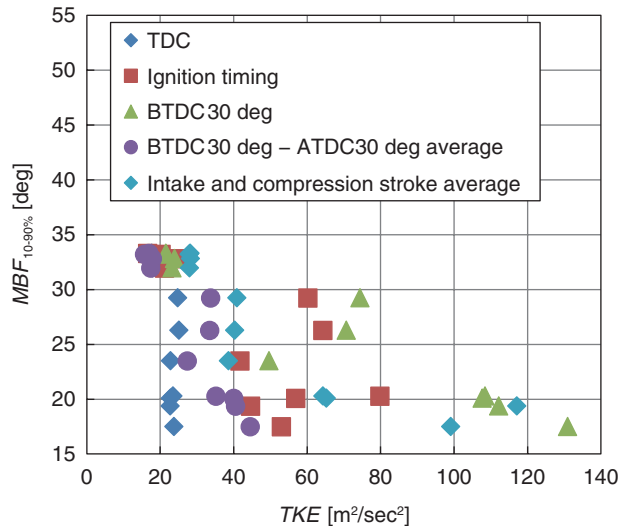


Fig. 2 Relationship between  $TKE$  and  $MBF_{10-90\%}$

With the flow coefficient termed  $Cf(x)$ , the  $FB$  value is defined in the same way as the average tumble ratio by Eq. (8).

$$FB = \frac{1}{CA_{IVC} - CA_{IVO}} \int_{CA_{IVO}}^{CA_{IVC}} \frac{FB[x(CA)]}{Cf[x(CA)]} \left[ \frac{pv(CA)}{mpv} \right]^2 dCA \quad (8)$$

In the present case, 16 divisions were employed in the space around the intake valve, and 15-deg increments were used for the crank angle.

## 4. Correlation between $MBF_{10-90\%}$ and $u'$ of Air-fuel Mixture

First, studies were conducted in order to determine whether the realization of increased  $TKE$  was effective in reducing the duration of  $MBF_{10-90\%}$ . Taking into consideration the expression of  $TKE$  as the dimension of velocity in studying the correlation, the evaluation was conducted for  $u' = \sqrt{TKE/1.5}$ . The relationship between this figure and  $S_C$  was studied.

### 4.1. Relationship between $MBF_{50\%}$ and $S_C$

$S_C$  is also affected by  $MBF_{50\%}$ . It was necessary to correct  $S_C$  in order to render the conditions uniform. Tests were therefore conducted using the two engines shown in Table 1 to study the relationship between  $MBF_{50\%}$  and  $S_C$ .

Figure 3 shows the results of the study of the relationship between the two parameters with different ignition timings at several EGR recirculation rates and  $\eta_v$ , at an engine speed of 2000 rpm. There is a first-order negative correlation between the position of the combustion

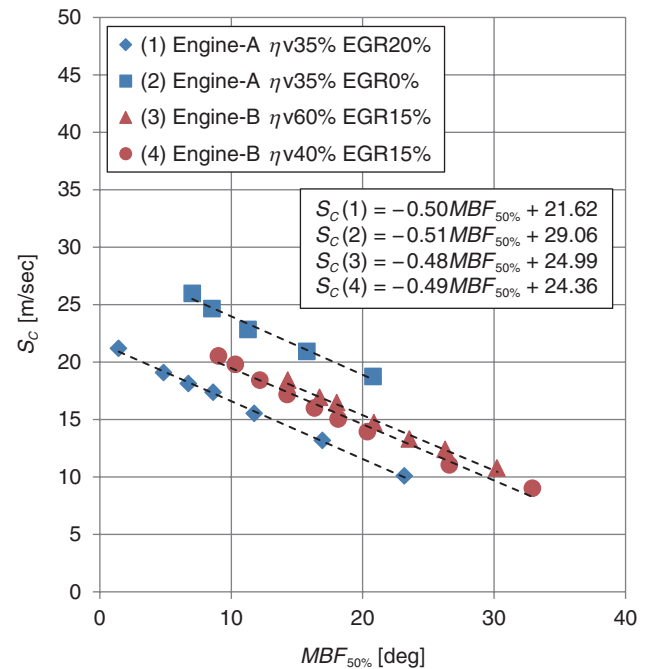


Fig. 3 Relationship between  $S_C$  and  $MBF_{50\%}$  to determine amount of correction

center of gravity and  $S_C$ , and the gradient was judged to be the same in each case. Taking into consideration the fact that the effect of differences in the load points is expressed as changes in the intercepts in the graph,  $S_C$  was corrected based on the relationship shown in Eq. (9), and the relationship between  $S_C$  and  $u'$  was examined.

$$\partial S_C / \partial MBF_{50\%} = -0.5 \quad (9)$$

#### 4.2. Correlation between Duration of Combustion and $u'$ of Air-fuel Mixture

Tests were conducted in which a variety of spacers were employed in the intake ports of the different engines shown in Table 1 in order to determine how the duration of combustion would change with changes in  $TKE$ . Results for an  $S_C$  corresponding to an  $MBF_{50\%}$  of 6.5 deg were isolated from  $MBF_{10-90\%}$  and  $MBF_{50\%}$  obtained from actual engine tests. Figure 4(b) shows the relationship between these

results and  $u' = \sqrt{TKE/1.5}$  obtained from 3D-CFD. Despite possessing different frameworks, the two engines can be approximated by the same lines, and a first-order correlation could be observed between in-cylinder  $S_C$  and  $\sqrt{TKE}$ . The results could be approximated as shown in Eq. (10).

$$S_C = 1.36u' + 10.7 = 1.11\sqrt{TKE} + 10.7 \quad (10)$$

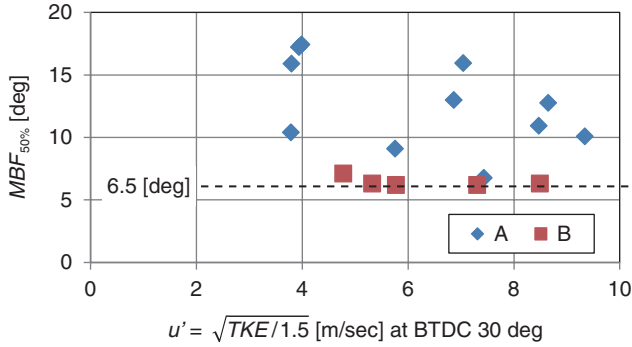
It was therefore considered that increasing  $TKE$  was effective in increasing burning velocity at least within the scope shown in Fig. 4(b).

### 5. Correlation between Tumble Ratio and $TKE$

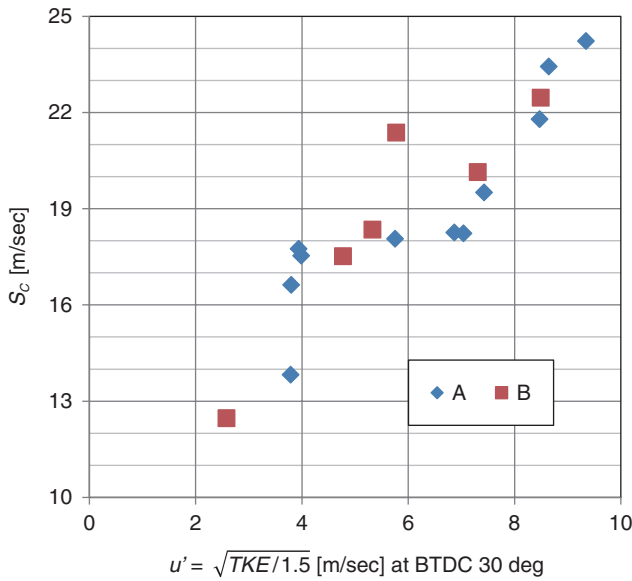
Next, cases in which a different  $TKE$  was obtained at the same tumble ratio were focused on, and a study was conducted in order to determine the factors producing these differences.

#### 5.1. Study of Correlation between $TKE$ and Tumble Ratio

The correlation between the tumble ratio and  $TKE$  was studied using Engine B in Table 1. In order to greatly change the tumble ratio, a tumble plate of the type shown in Fig. 5 was fitted in the intake port. The study was conducted using a shape model with the angle between the horizontal direction and the direction of the plate ( $\theta_1$ ), the plate length ( $L_1$ ), and position in the direction of plate height ( $x_1$ ) as parameters.



(a)  $MBF_{50\%}$  for each measurement point



(b) Relationship between  $u'_{BTDC 30 \text{ deg}}$  and  $S_C$  following correction of  $MBF_{50\%}$

Fig. 4 First-order correlation between  $u'_{BTDC 30 \text{ deg}}$  and  $S_C$  following correction of  $MBF_{50\%}$

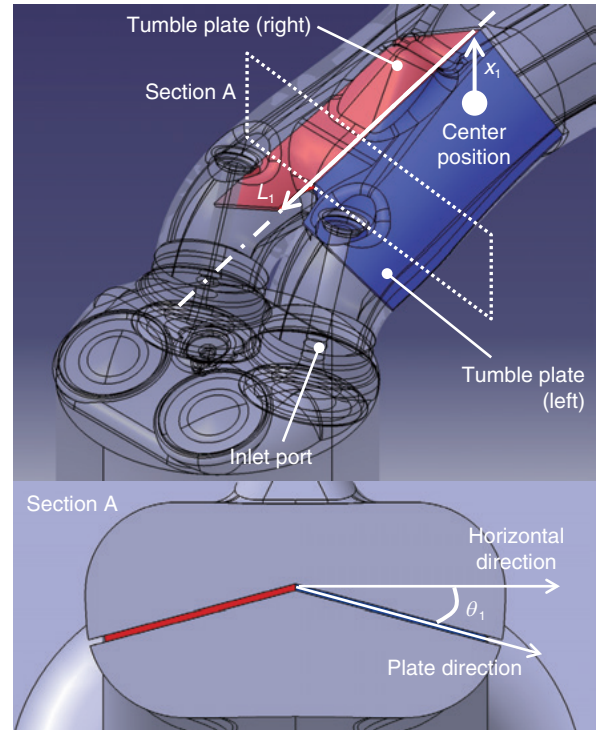


Fig. 5 Shape of tumble plate to change tumble ratio

Figure 6 shows the relationship between  $TKE$  and the tumble ratio determined by the study. There is a positive correlation between the tumble ratio and  $TKE$ , but a range of values for  $TKE$  can be obtained at the same tumble ratio. Within the range  $4.25 < TR < 4.45$  (termed “Range 1” below),  $TKE$  values of  $65.5 \text{ m}^2/\text{sec}^2$  and  $21.4 \text{ m}^2/\text{sec}^2$  are obtained. This equates to  $S_c$  of  $19.6 \text{ m/sec}$  and  $15.8 \text{ m/sec}$  using Eq. (10). If the figures are substituted in Eq. (5), this results in a difference of  $4.35 \text{ deg}$  at  $MBF_{10-90\%}$ .

## 5.2. Quantification of Direction of Flow and Correlation with $TKE$

The study discussed in the previous section was conducted using the same engine framework specifications, including the combustion chamber shape, and the same intake conditions, including the quantity of intake gas. It was therefore judged that the differences in  $TKE$  were due to differences in the behavior of the intake air flow from the intake port into the cylinders. This will be considered in this section. First, a comparison will be conducted of results from Range 1 in Fig. 6, at the same tumble ratio. Characteristic values were taken from this range, and the distribution of the flow velocity passing the valve through horizontal cross-sections. The calculated distribution of flow velocity passing the valve differs depending on the direction of the flow passing the valve, despite the fact that the tumble ratio is the same. In the case of inward flows, there are high-flow-velocity areas close to the center of the sleeve on the exhaust side. As the flow is directed outwards, the high-velocity areas shift to the side.

Figure 8 shows the relationship between Range 1  $FB$  values and  $TKE$ .  $TKE$  displays a tendency towards dependence on the value of  $FB$ . When  $FB$  exceeds 1,  $TKE$  reaches its peak. These results indicate the existence of an optimum direction for the intake air flow.

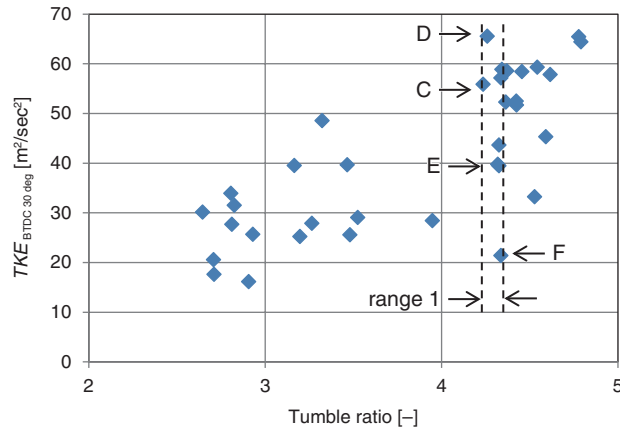


Fig. 6 Distribution of results for  $TKE_{BTDC\ 30\ deg}$  for same tumble ratio

## 5.3. Consideration concerning Relationship between Direction of Flow and $TKE$

The results of steady-state flow calculations showed that  $TKE$  reached its peak at a point at which the flow passing the valve was directed outwards. In order to consider the meaning of this, a study was conducted to determine how tumble vortex are formed, and how they are related to differences in  $TKE$ . Based on the principle that the flow speed at the center of a vortex is  $0 \text{ m/sec}$ , the center of a vortex were observed via iso-velocity surfaces expressing low-flow-

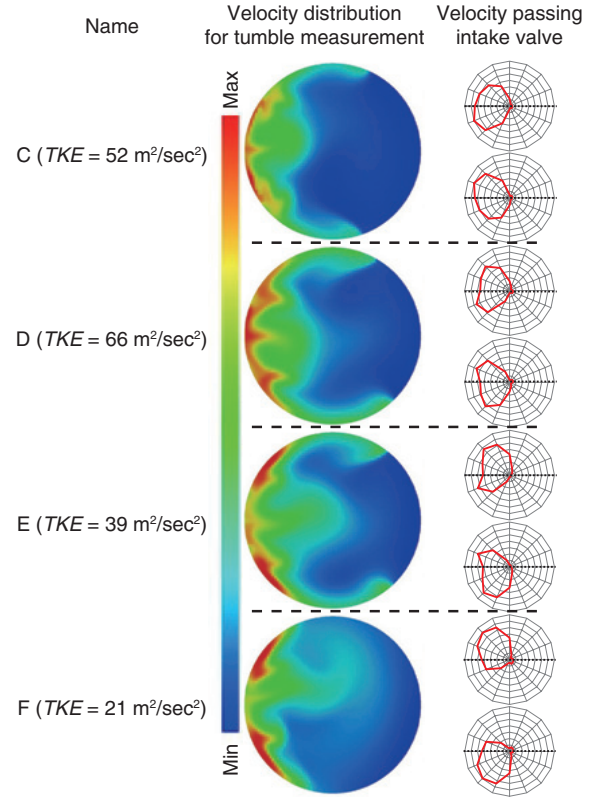


Fig. 7 Distribution of air flow velocity through horizontal cross-section

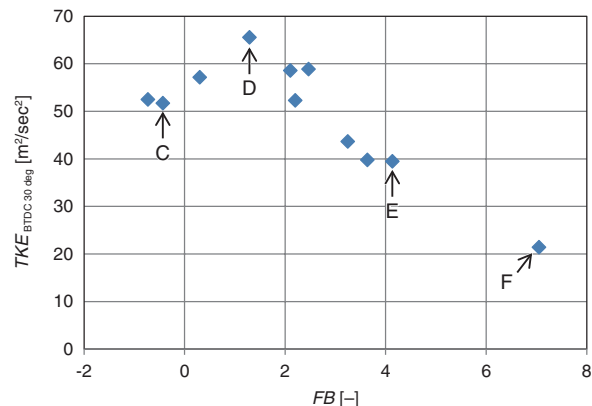


Fig. 8 Relationship between  $TKE_{BTDC\ 30\ deg}$  and  $FB$

speed areas (yellow surfaces expressing 3 m/sec flows and green iso-velocity surfaces expressing 5 m/sec flows).

Results were compared for Specification D, for which *TKE* was the highest, and Specification C, in which *FB* approaches 0 and it is considered that the flow passing the valve is almost straight. Figure 9 shows iso-velocity surfaces for both specifications. The vortex center line becomes straight more rapidly for Specification D than Specification C, and there is minimal deformation or migration of the vortex following this. In other words, a straight flow of intake air from the intake port in the direction of the exhaust port forms a curved tumble vortex. Directing the flow of intake air slightly towards the outside reduces the bending and distortion of this tumble vortex in the combustion chamber.

Figure 10 shows iso-velocity surfaces, flow velocity vectors at the bore center, and the up-down flow velocity

distribution at a position 60 mm below the cylinder head as the position of the center of the tumble vortex at ATDC 150 deg. The results of steady-state flow calculations for Specification C (Fig. 7) showed an area of high-velocity flow at the center of the sleeve on the exhaust port side. In an actual engine, the pistons are beyond this high-velocity area, and the downward flow is transformed into an upward flow. The tumble vortex is formed between these downward and upward flows. As a result, the tumble vortex is formed in a distorted shape, surrounding the area of high-velocity flow.

The direction of the flow passing the valve outwards reduces the flow velocity at the center of the crankshaft direction, and the flow velocity at the sides increases. As a result, the area in which the up-down flow velocity is low becomes a straight line, and the tumble vortex approaches a straight line (Specification D).

Figure 11 shows a conceptual diagram of this flow from the intake valve. Of the flows passing the valve, the flow

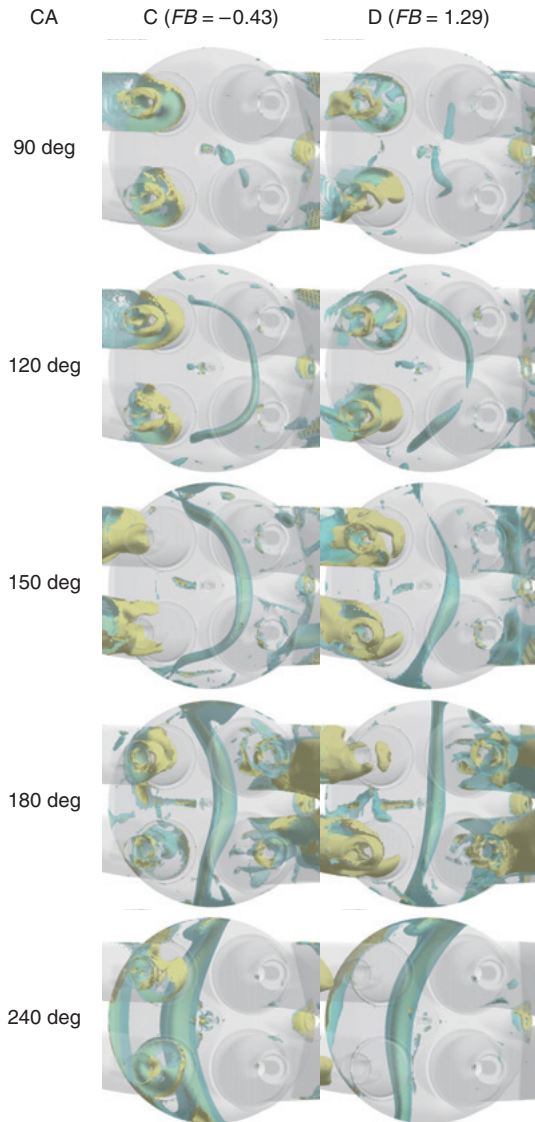


Fig. 9 Generation of curved tumble vortex center line and progression over time

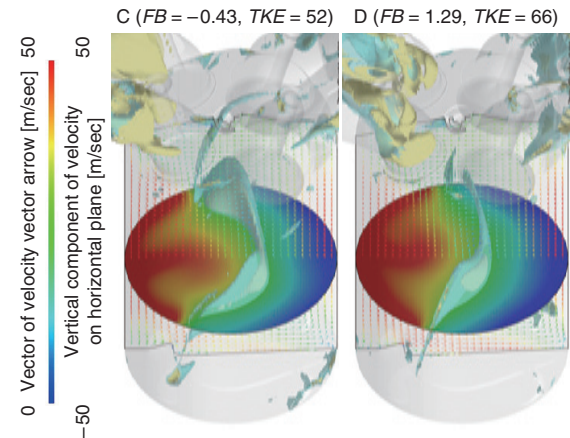


Fig. 10 Spatial relationship between curve of tumble vortex center and downward and upward flows

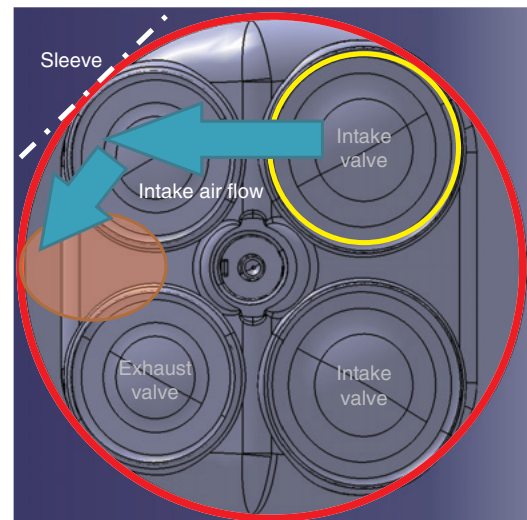


Fig. 11 Conceptual diagram of intake air flow

that runs straight from the intake valve to the exhaust valve is directed inwards when it strikes the bore sleeve because the sleeve is cylindrical, and as in the case of Specification C, the flow concentrates at the center of the exhaust port side of the sleeve. Because of this, the flow being directed outwards to a certain extent is essential to the formation of a linear tumble vortex.

The distortion of the vortex center locally increases the flow velocity, and the velocity gradient steepens. It is normal procedure to treat the air-fuel mixture as a Newtonian fluid; that is, the shear stress acting on the fluid is treated as being proportional to the velocity gradient, and it was considered that a steeper gradient would result in a greater dispersion of kinetic energy.

#### 5.4. Considerations regarding the Relationship between Flow Direction and Stability

Curved tumble vortices were considered next. Figure 12 shows the tumble vortex and flow speed vectors in a cross-section of the bore for Specification C at ATDC 150 deg. The front view shows a normal tumble, but the plan view shows that the tumble vortex is curved. The side view indicates symmetrical vortex motion on the left and right. This phenomenon, known as  $\omega$  tumble, is focused on when discussing the stability of flow motion from the perspective of variations in cycles<sup>(7)</sup>. The formation of the distorted tumble vortex axis creates instability for at least the time intervals termed cycle time. In addition,  $\omega$  tumble

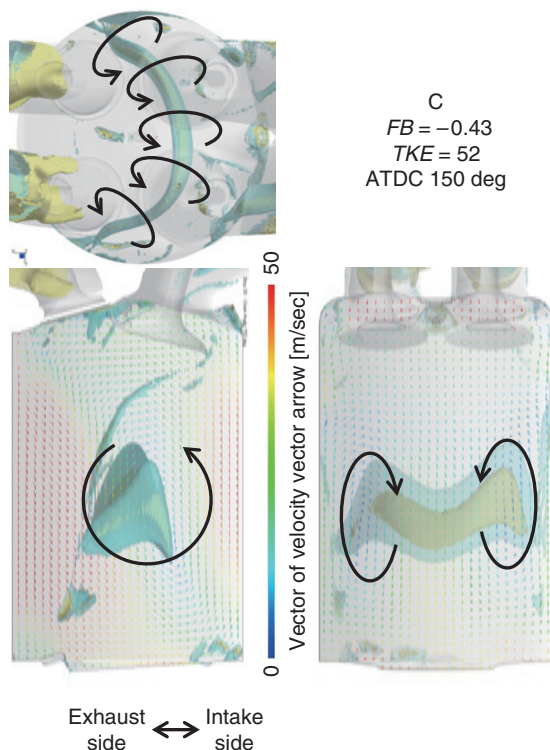


Fig. 12 Curved tumble vortex center line and velocity gradient

is not an independent phenomenon, but occurs as a result of the distortion of the tumble vortex, which is conventionally studied. For this reason, the setting of an appropriate flow balance is an effective means of controlling  $\omega$  tumble.

When the intake valve is closed and no new gas is being introduced, the curved tumble vortex is unstable from the perspective of symmetry, and is repeatedly distorted even within a single cycle. If it is considered that the distortion of the tumble vortex leads to small motions from the perspective of energy cascades, it is possible that there is a decline in the ability of the tumble vortex to retain kinetic energy.

## 6. Application to Intake Port Design

### 6.1. Setting Target $TKE$

Honda's research on the realization of a level of thermal efficiency of 45% in gasoline engines set a main combustion duration within 20 deg at an EGR rate of 30% as targets. However, the main combustion duration of Engine A in Table 1, which employed the intake port of the mass-production base engine, was 35 deg. In addition, it was necessary to advance the position of the combustion center of gravity by 4 deg in order to realize MBT operation. Taking  $S_C$  from Eq. (5) into consideration, a 15 deg reduction in the combustion duration would equate to a difference of 8.33 m/sec. If it were possible to advance the position of the combustion center of gravity by 4 deg, Eq. (9) indicates that  $S_C$  would be increased by 2 m/sec. Subtracting this from the figure of 6.33 m/sec, it would then be necessary to increase  $TKE$  by the amount that would realize a 6.33 m/sec increase in  $S_C$ . From Eq. (10), this would be an increase of 5.71 m/sec in  $\sqrt{TKE}$ . Table 3 shows the tumble ratio,  $FB$ , and  $TKE$  before and after the study of changes in the shape of the intake port. For the base port shape,  $TKE$  is 21.6 m<sup>2</sup>/sec<sup>2</sup>, and  $\sqrt{TKE}$  is 4.65 m/sec. This means that a  $\sqrt{TKE}$  of 10.4 m/sec and a  $TKE$  of 107.2 m<sup>2</sup>/sec<sup>2</sup> are actually necessary. On the other hand, while the correlation is weak, if the relationship between the tumble ratio and  $TKE$  is estimated as

$$TKE \propto 22.5 TR \quad (12)$$

from Fig. 6, then a tumble ratio of over 4 is necessary. However, CFD results suggested that at a tumble ratio in excess of 4, the intake air would tend to flow back into the intake port. Because of this, it was necessary to realize a  $TKE$  of 107.2 m<sup>2</sup>/sec<sup>2</sup> at a tumble ratio of 4 or below. It was therefore necessary to use the concept of flow balance in order to obtain a greater strength of turbulence from the flow at the same tumble ratio.

### 6.2. Selection of Intake Ports using CAE and Results of Engine Tests

Intake port shapes were studied with consideration of the limitations inherent in the manufacture of actual

Table 3 Degree of enhancement of intake flow

|                        | $TR (-)$ | $FB (-)$ | $TKE (m^2/sec^2)$ |
|------------------------|----------|----------|-------------------|
| Base                   | 2.34     | 0.04     | 21.6              |
| Following modification | 3.96     | 0.87     | 113.3             |

engine heads in order to determine a shape producing a tumble ratio of around 4. Table 3 shows the tumble ratio and  $FB$  before and after the study, and Fig. 13 shows the differences in the shapes of the intake ports. Table 3 also shows the results of verification of  $TKE$ . Because  $FB$  was at its maximum when the tumble ratio was close to 4, and the target value of  $107.2 m^2/sec^2$  had been realized for  $TKE$ , the decision was made to manufacture a prototype engine head at this stage, and tests were conducted in order to determine its effectiveness in reducing the duration of combustion. The results are shown in Fig. 14. The results show that the main combustion duration, which had been 35 deg, had been reduced to 20 deg, indicating that the target for reduction of the duration of combustion had been achieved.

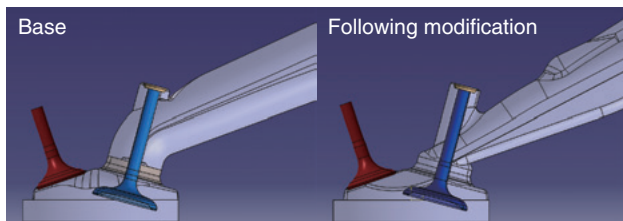
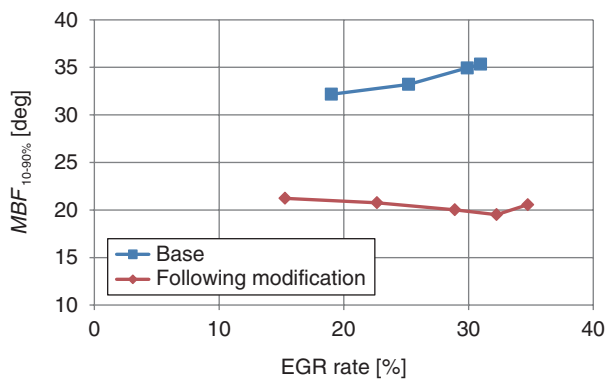


Fig. 13 Intake port shape before and after modification

Fig. 14 Reduction of duration of combustion by high  $TKE$ 

## 7. Conclusion

The research discussed in this paper demonstrated that increasing the  $TKE$  of the unburned air-fuel mixture is effective in reducing the duration of combustion. The results showed that the  $TKE$  finally obtained displayed a

range of values even for the same tumble ratio, and a study of the  $FB$  value indicated that greater strength of turbulence could be obtained from flows at the same tumble ratio. These findings were applied in the development of an intake port of engine A in Table 1, making it possible to reduce the main combustion duration by 15 deg at an EGR recirculation rate of 30%, including the amount realized by increasing the tumble ratio, and contributing to the realization of a level of thermal efficiency of 45%.

## References

- (1) International Energy Agency: Part4 Scenarios and Technology Roadmaps, Chapter13 Transport, Energy Technology Perspectives 2012, p. 443 (2012)
- (2) Tagishi, R., Ikeya, K., Takazawa, M., Yamada, T.: Technology for Realization of Gasoline Engine Brake Thermal Efficiency of 45%, Honda R&D Technical Review, Vol. 27, No. 2, p. 1-11
- (3) JP, 9-105330 (1997)
- (4) JP, 2008-184930 (2008)
- (5) U.S., Patent No. 6,385,530 (2002)
- (6) Hayakawa, A., Miki, Y., Kubo, T., Nagano, Y., Kitagawa, T.: Variations of Turbulent Burning Velocity and Flame Front Shape of Spherically Propagating Premixed Turbulent Flame with Effective Turbulence Intensity, Journal of the Combustion Society of Japan, Vol. 55, No. 172, p. 202-209 (2013)
- (7) Adomeit, P., Jakob, M., Pischinger, S., Brunn, A., Ewald, J.: Effect of Intake Port Design on the Flow Field Stability of a Gasoline DI Engine, SAE Technical Paper, 2011-01-1284 (2011)

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