

Construction of Turbulent Component Decomposition Method Using Time Filtering with High-speed PIV and Application to Hybrid Engine Development

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ABSTRACT

A time filtering method was constructed as a means of decomposing the turbulent component that contributes to promoting engine combustion. This time filtering method used the frequency at which the frequency characteristic of the in-cylinder flow changes as the cut-off frequency, and this frequency was confirmed to correspond to the time scale of the relatively large in-cylinder flow induced by the combustion chamber, intake port, and piston shapes. It was also clarified that the cut-off frequency can be uniquely determined by using the inverse of the integral time scale of in-cylinder flow as the cut-off frequency. The time filtering method was applied to a flow field equipped with a tumble control valve, which is an intake switching device. The obtained turbulence was highly correlated with the combustion duration of a metal engine, which confirmed the reliability of the decomposition method. The time filtering method was implemented in commercial analysis software and applied to hybrid engine development. Turbulence was enhanced by the in-cylinder flow enhancement specification that changed the intake port and piston shape from the early development specification and verified the in-cylinder flow enhancement concept for achieving rapid combustion.

1. Introduction

Social attention is increasingly focused on environmental issues such as global warming due to carbon dioxide (CO₂) emitted by human economic activities. In FY2022, CO₂ emissions from Japan's transportation sector accounted for approximately 18.5% of the country's total, and of that passenger cars accounted for approximately 44.9%⁽¹⁾. To reduce CO₂ emissions from passenger cars, measures such as banning the sale of internal combustion engines and shifting to electrification using batteries or fuel cells as power units are progressing on a global scale. However, batteries require rare metals such as cobalt, the stable supply of which is thought to be an issue due to limited areas for extraction and ore production⁽²⁾. Furthermore, life-cycle assessments that take into account CO₂ emissions produced by battery manufacturing and thermal power generation and other factors suggest that CO₂ emissions from battery electric vehicles may be higher than those of vehicles equipped with gasoline engines in some cases⁽³⁾. Given that background,

efforts are underway to develop fuel compositions suited to gasoline engines in order to achieve carbon neutrality in automotive internal combustion engines^{(4), (5)} and to research synthetic fuels derived from renewable electric power (Erneuerbarer strom - fuel: e-fuel) that assume drop-in enabling use of existing infrastructure as is⁽⁶⁾.

Combustion concepts using lean-burn combustion^{(7), (8)} and massive exhaust gas recirculation (EGR)⁽⁹⁾ in spark ignition (SI) gasoline engines are being studied for the purpose of enhancing thermal efficiency. These combustion concepts add extra air and EGR gas to the air-fuel mixture to reduce pumping loss and heat loss. On the other hand, efforts are being made to enhance the in-cylinder flow by measures such as optimizing the intake port shape to compensate for the decrease in the combustion rate due to a lean or diluted air-fuel mixture⁽¹⁰⁾. Such a method, however, may lead to lower power due to increased intake resistance in the mid-to-high engine speed range. Research and development of intake switching devices is underway to address this issue with the aim of achieving both a higher

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tumble ratio and lower intake resistance according to the engine operating conditions^{(11), (12)}.

2. Turbulent Component Decomposition Method

2.1. Issues with Conventional Methods

Turbulent enhancement is effective for promoting flame propagation combustion in SI gasoline engines, and the cycle average method has conventionally been used to evaluate turbulence in engine cylinders. The cycle average method defines the difference between the cycle-averaged velocity and the instantaneous velocity as the turbulent component⁽¹³⁾⁻⁽¹⁵⁾. It has been pointed out, however, that this method also includes the relatively low-frequency cycle-by-cycle fluctuation of the mean flow component in the turbulent component, so it may overestimate the turbulence. Studies have attempted decomposition using a time filter as another decomposition method. This method used a steady-state time-averaging method to decompose the mean flow component and the turbulent component every cycle based on the velocity near the compression stroke top dead center (TDC). It was assumed that the gas flow can be regarded as a steady flow for a short duration in each cycle near the compression stroke TDC, but a clear physical definition of the averaged time considered to be steady flow is not given^{(16), (17)}. It was also proposed to use the frequency at which the integrated power spectrum of the cycle-averaged velocity was 90% or more as the cut-off frequency, however the cut-off frequency was considered arbitrary^{(18), (19)}. In this way, various methods have been proposed for turbulent component decomposition, but ambiguity remains in the cut-off frequency setting. Therefore, for example, when designing an intake port using the measured turbulent intensity as the criteria, it may be necessary to rework the design because the turbulence is overestimated.

2.2. Time Filtering Method

The authors have proposed a time filtering method

utilizing the high time resolution of high-speed particle image velocimetry (PIV), which is capable of measuring at high speeds, as a new turbulent component decomposition method⁽²⁰⁾. Figure 1 shows the velocity u and the frequency characteristic of the in-cylinder flow from the intake stroke to the compression stroke. (Refer to Fig. 6 hereafter for the velocity direction.) The velocity of an arbitrary cycle fluctuates relative to the cycle-averaged velocity, and the conventional cycle average method treated this fluctuation as the turbulent component. The time filtering method obtains the frequency characteristic by the discrete Fourier transform every cycle, defines the frequency at which the slope of the cycle-averaged characteristic changes as the cut-off frequency f_{c_slope} , and then decomposes the component with a higher frequency than f_{c_slope} as the turbulent component. Here, $\hat{u}$ and $\hat{v}$ are the discrete Fourier coefficients of the u and v velocity components. Figure 2 compares the turbulent kinetic energy k obtained by the cycle average method and the time filtering method. This shows that the turbulent kinetic energy k_c with the cycle average method is overestimated overall compared to the turbulent kinetic energy k_f with the time filtering method. In particular, the turbulent kinetic energy in the latter half of the compression stroke directly affects combustion, so the combustion characteristics may be overestimated. From the above, the time filtering method is characterized by treating the frequency component higher than the cut-off frequency as the turbulent component, so it does not include the cycle variation of the mean flow component (low-frequency component), which is thought to not contribute to promoting combustion, in the turbulent component.

The object of this study is to construct a turbulent analysis method that can be applied to engine in-cylinder flow. First, the effect of the time resolution needed for the time filtering method was confirmed, and the cut-off frequency setting for decomposing the mean flow component and the turbulent component was investigated. Next, the time filtering method was applied to a tumble

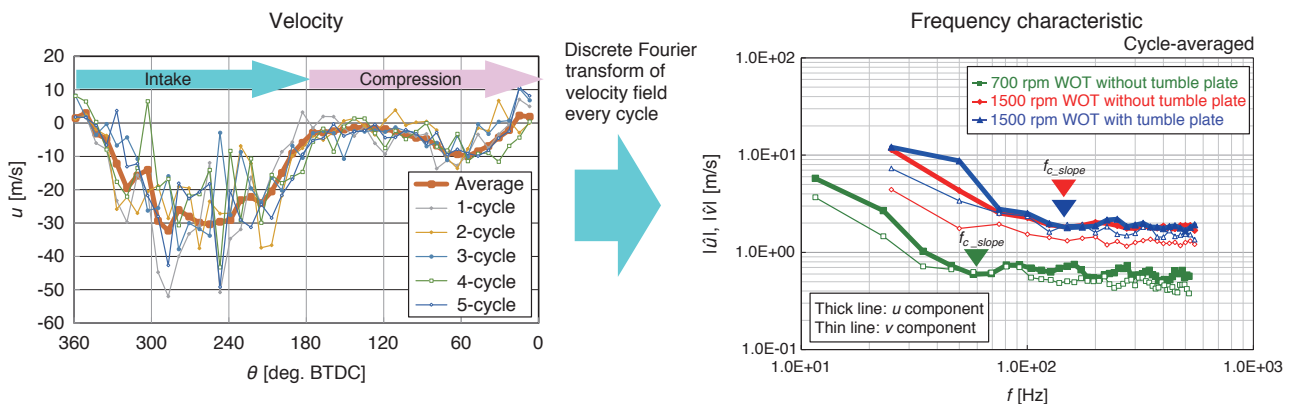


Fig. 1 Velocity and frequency characteristic of in-cylinder flow

control valve (TCV), which is an intake switching device, and the reliability of the turbulent analysis method was verified from the correlation between the obtained turbulence and the combustion characteristics. Finally, the time filtering method was implemented in a commercial PIV auto analysis tool, and the concept of in-cylinder flow enhancement in a hybrid engine was verified. This paper reports these details.

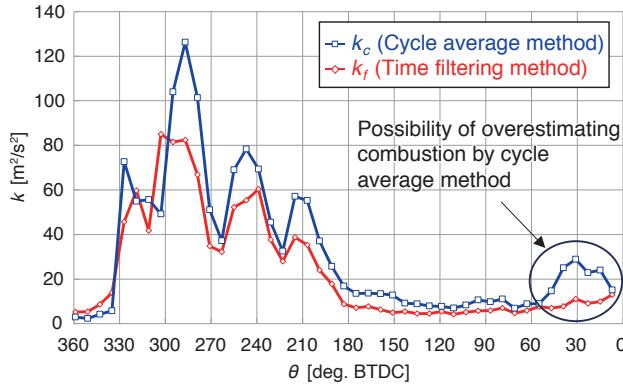


Fig. 2 Comparison of turbulent kinetic energy obtained by time filtering method and cycle average method

3. Construction of Time Filtering Method

3.1. Experimental Technique

3.1.1. Engine specifications

Table 1 shows the main engine specifications, and Fig. 3 shows the appearance of the engine. The engine is an optical full-stroke single-cylinder engine made by Koyama Garage Co., Ltd. It has a bore diameter of 73 mm and a stroke of 89.4 mm and is equipped with two intake and two exhaust valves. A quartz sleeve is installed between the cylinder head and the cylinder block, providing a side view of the inside of the cylinder from the intake stroke to the compression stroke. The top shape of the extended piston can be changed by parts replacement. For optical access from below the piston, it can be changed to a flat shape with a quartz window that provides a view via a 45-degree mirror installed below the extended piston.

3.1.2. Experimental setup

The PIV used for velocity measurement uses a laser or other light source to illuminate tracer particles mixed into the airstream and obtains the velocity from the amount of particle movement in the images captured by a camera. A high-repetition double-pulse Nd:YAG laser (LDP-50MQG made by Lee Laser, Inc.) was used as the light source, and the wavelength of the laser beam was 532 nm. A high-speed CMOS camera (APX-RS made by PHOTRON LIMITED) was used to capture particle images. SiO₂ (Godd Ball B-6C

Table 1 Engine specifications

Engine	Optical single-cylinder engine with quartz sleeve
Bore × Stroke	73 mm × 89.4 mm
Displacement	374 cm ³
Valve train	DOHC 4 valves

made by SUZUKIYUSHI INDUSTRIAL CORPORATION) with an average particle diameter of 3 μm was used as the tracer and was supplied into the engine from the intake chamber located immediately after the throttle. The air pressure of the tracer supply device was adjusted to achieve a tracer particle concentration that enabled imaging of a few particles in the 16 × 16 pixel inspection area for the PIV analysis described hereafter.

Figure 4 shows the crank angle range over which measurement was performed. The intake valve lift curve,

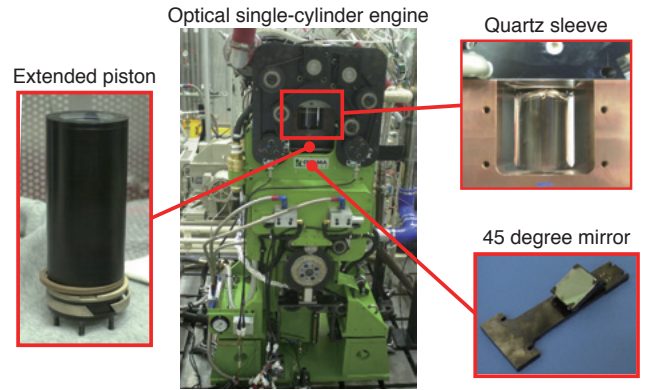


Fig. 3 Appearance of optical single-cylinder engine

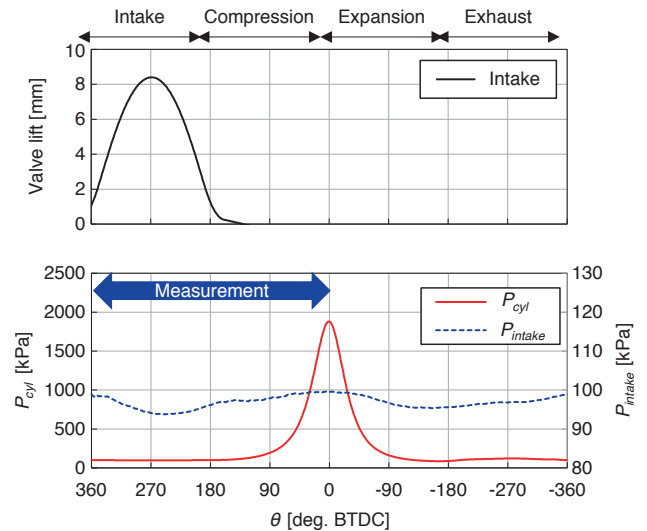


Fig. 4 Measurement range of in-cylinder flow

in-cylinder pressure P_{cyl} , and the intake manifold pressure P_{intake} are added to the figure for reference. The measured crank angle was from the intake stroke TDC crank angle $\theta = 360$ deg. BTDC to the compression stroke TDC crank angle $\theta = 0$ deg. BTDC, and this was measured for multiple cycles. The obtained particle images were analyzed using the commercial PIV analysis software Koncerto ver1.0 made by Seika Digital Image CORPORATION. The analysis algorithm used the recursive FFT cross-correlation method, and the inspection area was varied in three steps from 64×64 pixels to 16×16 pixels, with overlap of 50%. Three test conditions were used: $Ne = 700$ rpm, 1500 rpm, and 2000 rpm, all at wide open throttle (WOT).

3.2. Examination of Cut-off Frequency Setting Method

In this section, the time resolution needed for the time filtering method is clarified, and the method for setting the cut-off frequency and its physical meaning are examined.

Figure 5 shows the layout of the PIV system used in the test, and Fig. 6 shows the measurement region and

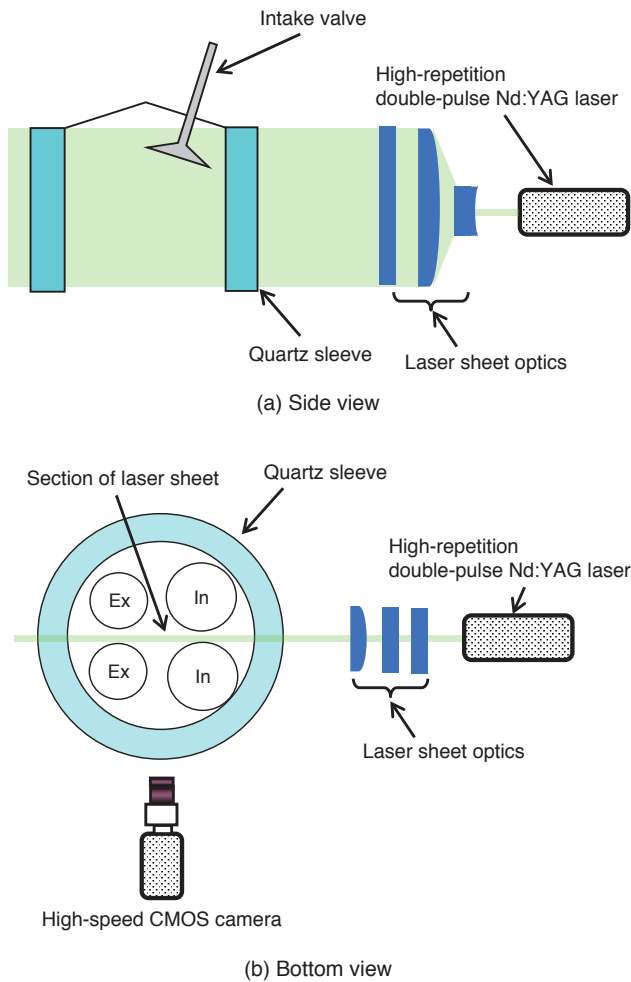


Fig. 5 Layout of PIV system for construction of turbulent analysis method

analysis area. The laser beam was stretched into a 0.5 mm thick sheet by cylindrical lenses, and then irradiated from the side of the engine into the cylinder through a transparent sleeve. The measurement position was the bore center. Imaging was performed with a frame speed of 20 kilo frame per second (kfps), an imaging range of horizontal 40 mm $\times$ vertical 18 mm, and a spatial resolution of 0.077 mm/pixel (512×256 pixel). In addition, a horizontal 36 mm $\times$ vertical 6 mm range at the top of the cylinder was used as the analysis area to enable turbulent analysis up to the vicinity of the compression stroke TDC.

3.2.1. Comparison of time filtering method and cycle average method

Figure 7 compares the cycle-averaged velocity distribution and instantaneous velocity distribution using the in-cylinder flow at $\theta = 300.1$ deg. BTDC in the early stage of the intake stroke as an example. This showed that the instantaneous velocity distribution and the tumble vortex center (red dot) differed in each of the arbitrary six cycles shown in Fig. 7(b). In addition, the cycle-averaged velocity distribution shown in Fig. 7(a) also differed from the velocity distribution in any cycle.

Fig. 8(a) shows the difference in the energy decomposed by the time filtering method and the cycle average method. The energy E on the vertical axis is the mean square value of the u and v velocity components at the analysis point. The time filtering method obtained the frequency characteristic by the discrete Fourier transform every cycle, and the cycle-averaged values were indicated in blue. The frequency f_{c_slope} at which the slope changes

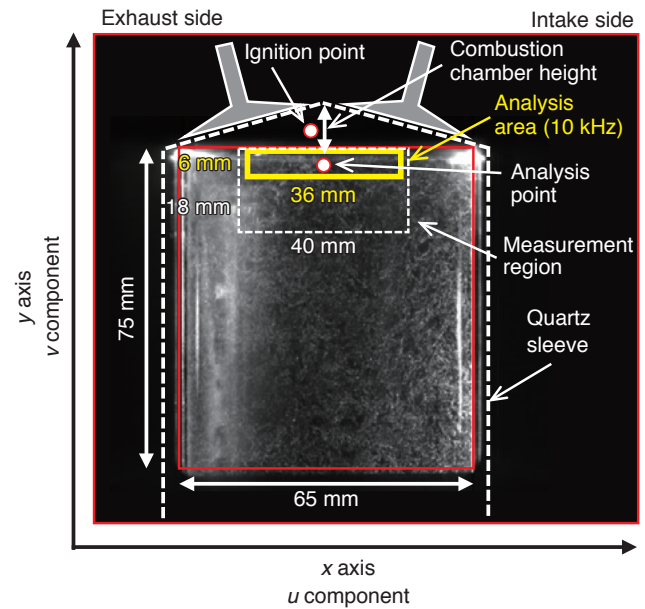


Fig. 6 Measurement region and analysis area for construction of turbulent analysis method

was 360 Hz, and the frequency band higher than that was the turbulent component (blue painted area). This turbulent component corresponds to the mean square value of the mean flow and instantaneous velocity difference shown in Fig. 8(b). The frequency characteristic of the cycle average method was obtained by the discrete Fourier transform from the cycle-averaged velocity and was indicated by the solid red line in Fig. 8(a). The turbulent component of the cycle average method was the red painted area surrounded by the spectra of the time filtering method and the cycle average

method, and the area surrounded by the black dotted line was overestimated compared to the time filtering method. This overestimated area is the mean square value of the velocity difference between the time filtering method and the cycle average method in Fig. 8(b); that is to say, the cycle variation of the mean flow component.

3.2.2. Effect of time resolution

The time resolution is thought to be important with the time filtering method, so it was analyzed. Figure 9 shows

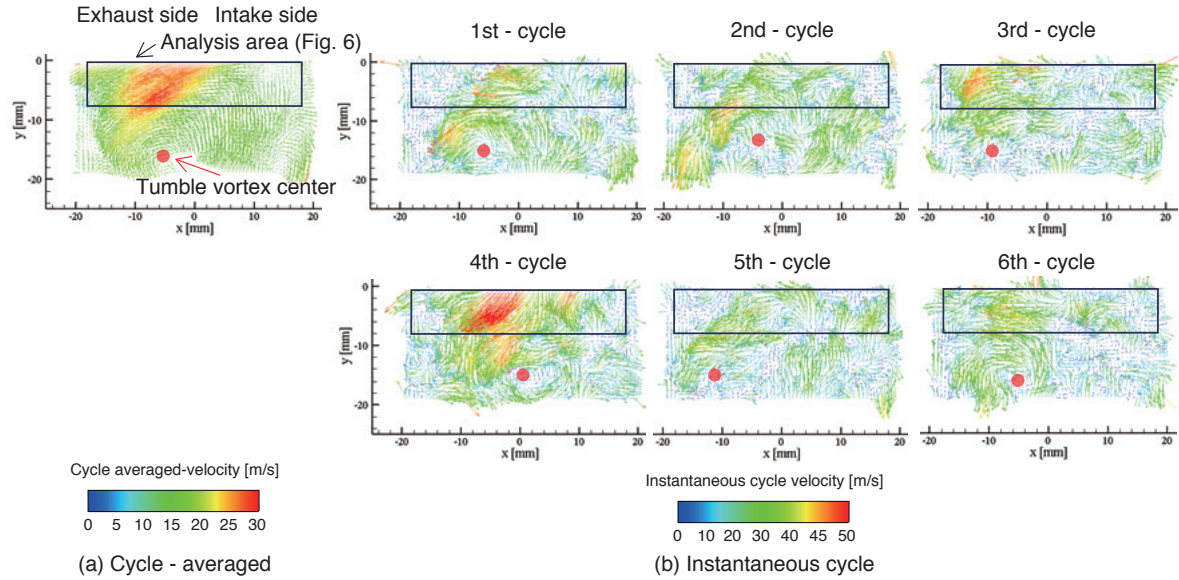


Fig. 7 Comparison of cycle-averaged in-cylinder flow and instantaneous in-cylinder flow for 6 cycles in measurement region, $Ne = 1500$ rpm WOT at $\theta = 300.1$ deg. BTDC

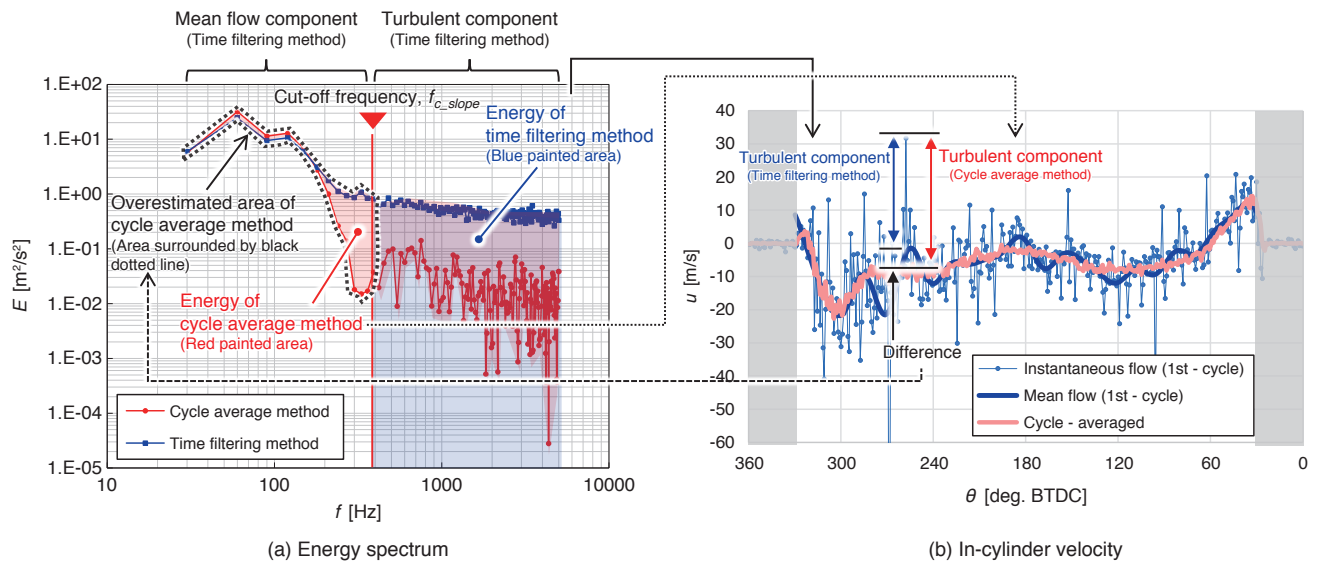


Fig. 8 Comparison of energy obtained by time filtering method and cycle average method at analysis point, $Ne = 1500$ rpm WOT

the effect of time resolution on the energy spectrum E_u of the u velocity component at the analysis point. The time resolution was artificially reduced by thinning of the same measurement data in the crank angle direction. When the time resolution was between around 10 to 2.5 kHz, this did not produce a noticeably large effect on the frequency where the slope changes. The discrete Fourier transform is performed in this analysis, so when a lower time resolution is used, frequencies that differ by an integer multiple of the Nyquist frequency f_N that is half-frequency of the true spectrum and the time resolution are viewed as the same due to aliasing. That is to say, the spectra for each f_N interval overlap the true spectrum, resulting in deviation due to discretization⁽²¹⁾. In this flow field, even when the time resolution is lowered from 10 kHz to 5 kHz, the reduction in energy of 360 Hz or more is approximately 5%, so the effect of aliasing on f_{c_slope} is thought to be small, assuming a time resolution of around 5 kHz.

3.2.3. Cut-off frequency setting method

In this section, the integral time scale of in-cylinder flow τ , the physical meaning of f_{c_slope} , and the cut-off frequency setting are examined.

Figure 10 compares the spatial-averaged integral time scale of in-cylinder flow in the analysis area and the characteristic time of the piston speed τ_p . τ_u was obtained by using Eq. (1) to calculate the autocorrelation coefficient R_{uu} from the instantaneous velocity u , and then integrating with Eq. (2) until the autocorrelation coefficient R_{uu} was 0⁽²²⁾. The integral time scale of in-cylinder flow τ_v of the v velocity component was also calculated in the same manner.

$$R_{uu}(\Delta\theta) = \frac{1}{u(\theta)^2} \sum u(\theta)u(\theta + \Delta\theta) \quad (1)$$

$$\tau_u = \sum R_{uu}(\Delta\theta) \quad (2)$$

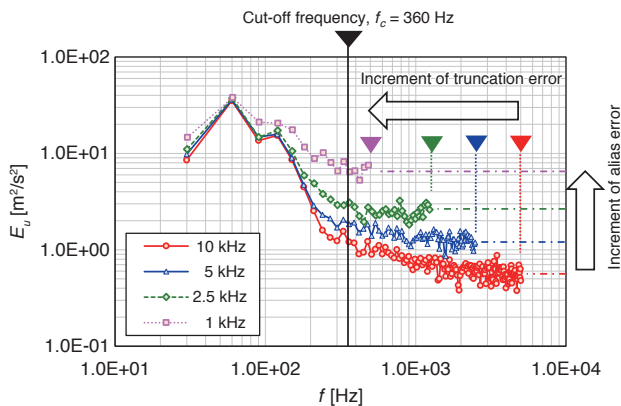


Fig. 9 Effect of time resolution on cycle-averaged in-cylinder energy spectrum at analysis point, $Ne = 1500$ rpm WOT

τ_p was defined as the value obtained by dividing the height of the combustion chamber h_p from the top of the pent-roof combustion chamber to the piston top at bore center by the instantaneous piston speed u_p . h_p at the TDC was 15 mm. Thus, on the integral time scale of in-cylinder flow, particularly the change in τ_v relative to the crank angle exhibited tendencies close to τ_p . In this relationship, the integral time scale of in-cylinder flow is thought to correspond to the relatively large-scale in-cylinder flow induced by the combustion chamber and intake port shapes and the piston speed. The average of τ_u and τ_v at $\theta = 270$ deg. BTDC, which is in the middle of the intake stroke, is used as the integral time scale of in-cylinder flow τ , and its inverse $f_{c_ \theta = 270}$ and f_{c_slope} are compared in Fig. 11. The data of Chapter 4 is also plotted in the figure. As a result, the correlation coefficient between $f_{c_ \theta = 270}$ and f_{c_slope} was 0.936 in Chapter 3, and 0.918 overall with the Chapter 4 data

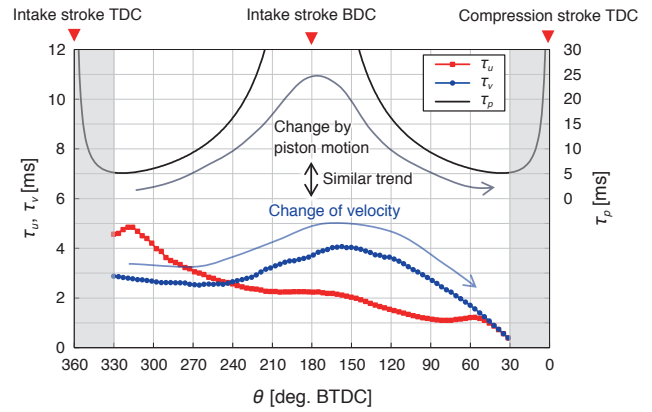


Fig. 10 Comparison of spatial-averaged integral time scale of in-cylinder flow, τ_u , τ_v in analysis area and time scale of piston speed, τ_p . Test condition is $Ne = 1500$ rpm WOT.

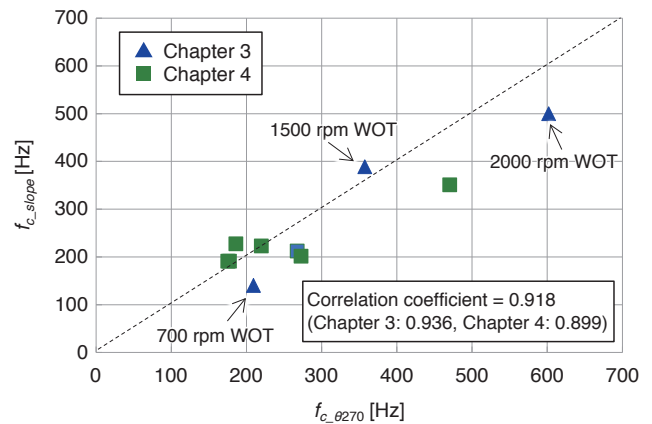


Fig. 11 Relationship between cut-off frequency at changed slope of in-cylinder flow frequency characteristic f_{c_slope} and cut-off frequency obtained from autocorrelation $f_{c_ \theta 270}$

added, indicating a high correlation.

From the above, the frequency f_{c_slope} at which the slope of the frequency characteristic of the in-cylinder flow changes is thought to be correlated with the integral time scale of the in-cylinder flow τ and corresponds to the relatively large-scale in-cylinder flow induced by the combustion chamber and intake port shapes and the piston speed. In addition, the cut-off frequency can be uniquely determined using the inverse of integral time scale of in-cylinder flow, $f_{c_0.270}$ as the cut-off frequency. Unless stated otherwise, $f_{c_0.270}$ is used as the cut-off frequency f_c hereafter.

4. Verification of Reliability of Time Filtering Method

In this chapter, the time filtering method was applied to the intake switching device, TCV, and the reliability of the time filtering method, which is a turbulent component decomposition method, was verified based on the correlation between the obtained turbulence and the combustion characteristics.

Table 2 shows the TCV specifications, and Fig. 12 shows the schematic diagram. When the lower passage of the tumble plate is closed by a flap, air flows through the upper passage of the tumble plate, so this specification is called upper flow and is noted as UF. Conversely, when the upper passage of the tumble plate is closed by a flap, air flows through the lower passage of the tumble plate, so this specification is called lower flow and is noted as LF. The longer the length of the tumble plate L_{plate} , the closer the tip position of the tumble plate to the intake valve stem, and the flow into the combustion chamber is thought to have higher directivity. Therefore, the distance G_{plate} from the tip of the tumble plate to the intake valve stem is used as the tumble plate parameter, and the three specifications of 2 mm, 25 mm, and 50 mm are provided. The flap and tumble plate specifications are noted together hereafter. For example, the case of upper flow and $G_{plate} = 2$ mm is noted as UF 2 mm.

Figure 13 shows the measurement region and the analysis area. The laser sheet beam was reflected upward by the 45-degree mirror and irradiated from the underside of the engine through the piston window and into the combustion chamber. The measurement position was the bore center, and the measurement range was a horizontal $46.6 \text{ mm} \times$ vertical 24 mm region in the combustion chamber. In order to analyze turbulence from ignition by the spark plug to the early stages of combustion, the spark plug was removed and a dummy plug without the outer and center electrodes was installed. The ignition point of the spark plug was hidden by the intake valve in the intake stroke, so a position offset 4.6 mm to the exhaust side (analysis point) was used for the analysis. In addition, when performing spatial averaging, the target was the horizontal $8 \text{ mm} \times$ vertical 4 mm range (analysis area) where intake valve lift did not affect the analysis results. The test condition was $Ne = 1500 \text{ rpm WOT}$.

Table 2 Specifications of tumble control valve

Specifications	Flow part	G_{plate} [mm]	L_{plate} [mm]
UF 2 mm	Upper	2	93
UF 25 mm	Upper	25	70
UF 50 mm	Upper	50	45
LF 2 mm	Lower	2	93
LF 25 mm	Lower	25	70
LF 50 mm	Lower	50	45

4.1. Turbulent analysis of TCV

Figure 14 shows the spatial distributions of the mean flow component and the turbulent kinetic energy k_f for the cases without TCV, TCV UF 2 mm (upper flow), and TCV LF 2 mm (lower flow). The cut-off frequency f_c was 267 Hz, 219 Hz and 470 Hz, respectively. The turbulent kinetic energy k_f was obtained by Eq. (3).

$$k_f = \frac{1}{2} (u'^2 + v'^2) \quad (3)$$

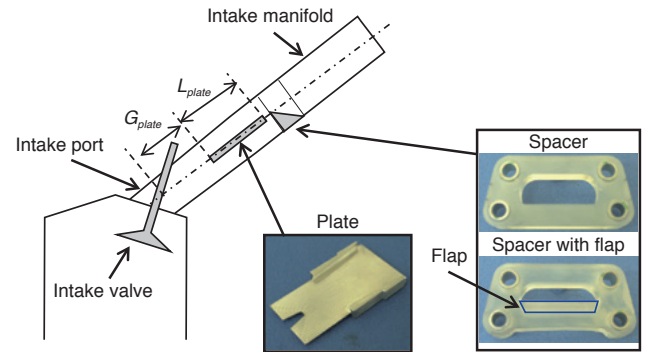


Fig. 12 Schematic diagram of tumble control valve

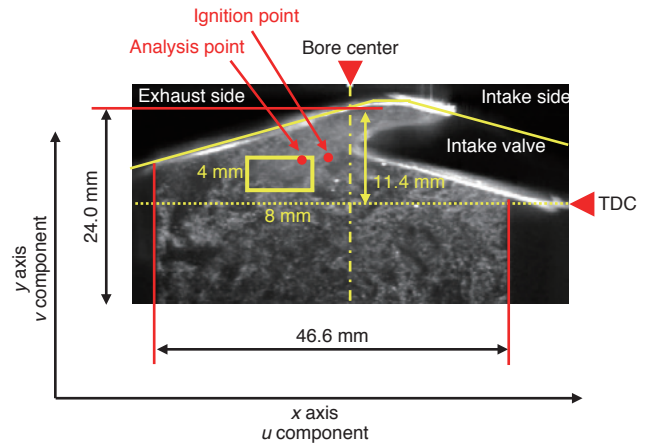


Fig. 13 Measurement region and analysis area for turbulent analysis with tumble control valve

Here, u'_f and v'_f are the turbulent velocity components decomposed by the time filtering method. As shown, the mean flow velocity of TCV UF 2 mm was faster than without TCV, and the vortex center was near the center of the combustion chamber. The mean flow velocity of TCV LF 2 mm indicated almost the same values as without TCV, and there was also no significant difference in the distribution. The k_f of TCV UF 2 mm indicated higher values than without TCV, and the k_f near the center of the combustion chamber where the vortex center of the tumble flow was located indicated particularly high values. On the other hand, the k_f of TCV LF 2 mm indicated almost the same values as without TCV, and there was also no difference in the distribution. The reason for this is that, as shown in the schematic diagram in Fig. 14, the flow into the cylinder from the lower side of the intake port is directed to the exhaust side by the intake valve and does not contribute to enhancing the tumble flow.

4.2. Estimation of turbulent flame structure

The flame structure with a TCV mounted was estimated from the turbulent premixed combustion diagram⁽²³⁾. Figure 15 shows the results at $\theta = 0.4$ deg. BTDC, which

is near the compression stroke TDC. The turbulent intensity u' on the vertical axis used the square root of the turbulent kinetic energy k_f measured in monitoring with the optical engine. The laminar burning velocity S_L was obtained by

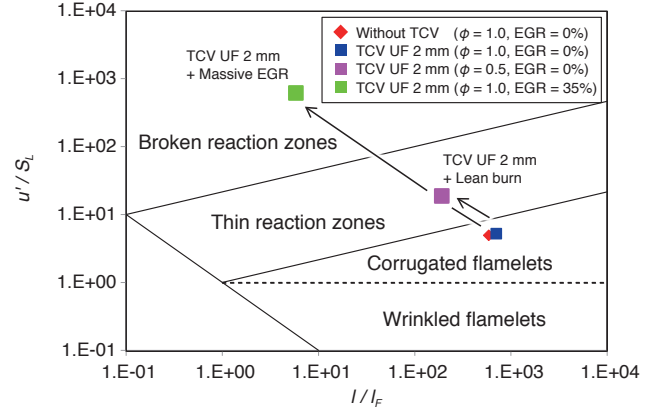


Fig. 15 Effect of enhancement of in-cylinder flow by TCV in analysis area, $Ne = 1500$ rpm WOT at $\theta = 0.4$ deg. BTDC

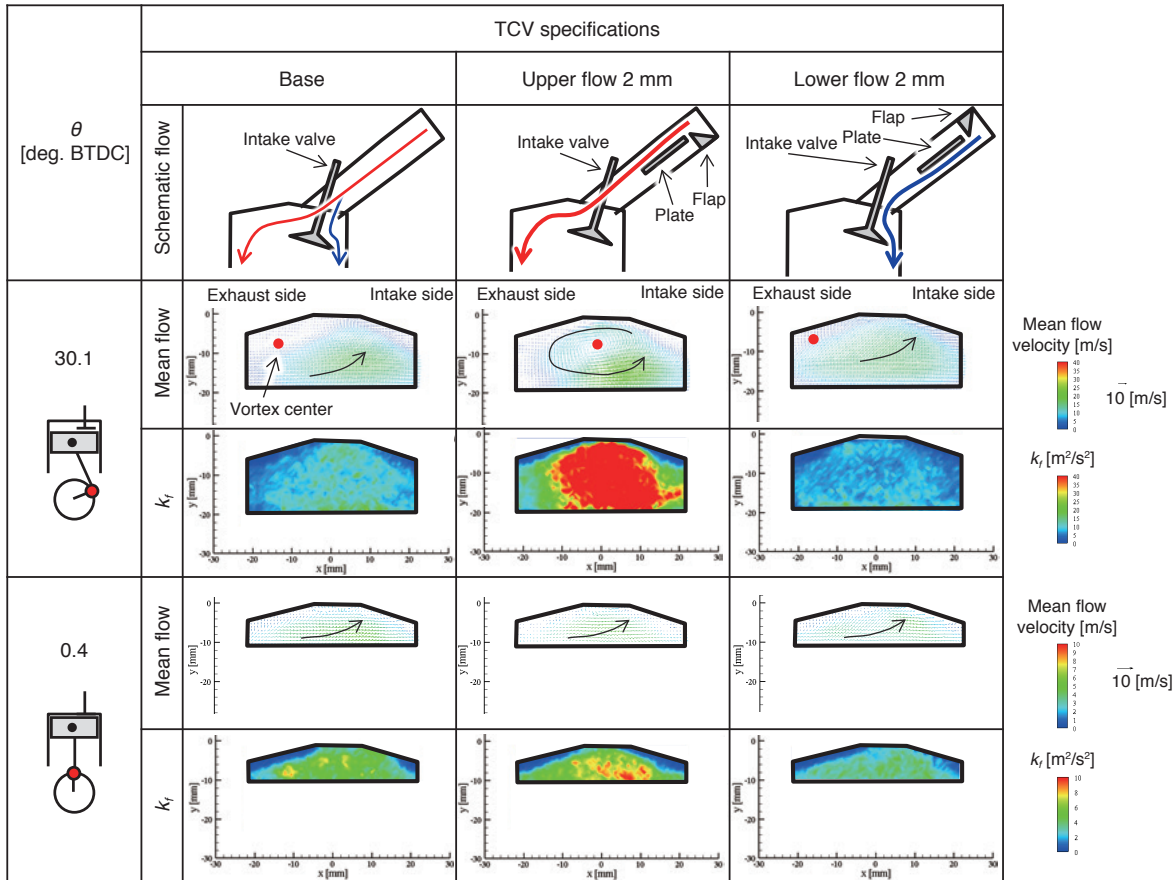


Fig. 14 Distribution of mean flow velocity and turbulent kinetic energy k_f obtained by time filtering method, $Ne = 1500$ rpm WOT in without TCV, upper flow 2 mm and lower flow 2 mm

elementary reaction calculations, assuming that the physical properties of the fluid were an air-fuel mixture with an equivalent ratio of $\varphi = 1.0$ and an EGR ratio of 0%⁽²⁴⁾. The fuel composition was surrogate gasoline (primary reference fuel: PRF), and the conditions were in-cylinder pressure $P_{cyl} = 1270$ kPa and in-cylinder gas temperature $T_{cyl} = 532$ K. The turbulent integral length scale l and the flame thickness l_F on the horizontal axis were respectively obtained by Eq. (4) and Eq. (5).

$$l = u' \times \tau_l \quad (4)$$

$$l_F = \nu_F / S_L \quad (5)$$

Here, τ_l is the integral time scale of the turbulent intensity u' , ν_F was the dynamic viscosity coefficient of the air-fuel mixture, and the fuel composition was obtained by the physical property estimation method⁽²⁵⁾.

As shown, the flame structures of without TCV and TCV UF 2 mm were both classified as corrugated flamelets. The u' and l of TCV UF 2 mm were larger than without TCV, so it was plotted slightly to the upper right of without TCV. In the corrugated flamelets region, wrinkles form on the flame surface, but turbulence is thought to not affect the internal structure of the flame, and the wrinkles on the flame surface are more pronounced than in the wrinkled flamelets region⁽²⁶⁾. That is to say, turbulence is enhanced and the increase in the flame surface area is expected to promote combustion. In anticipation of further enhancement of combustion efficiency, the turbulent flame structures were also calculated for lean burn and massive EGR. When the air-fuel mixture was diluted from an equivalent ratio of $\varphi = 1.0$ to $\varphi = 0.5$, S_L decreased from approximately 42.4 cm/s to 11.5 cm/s, and the flame structure changed to the thin reaction zones. When the EGR ratio increased to 35% at $\varphi = 1.0$, S_L decreased to 0.35 cm/s and the flame structure shifted to the broken reaction zones. These flame structures are thought to have local flame quenching due to turbulence. That is to say, in the case of these flame structures, combustion may not be promoted simply by enhancing the in-cylinder flow. Additional measures are therefore needed, such as enhancement of ignitability by suppressing cycle fluctuation of the mean flow around the spark plug, etc.

4.3. Relationship between turbulence and combustion characteristics

The effect of the TCV on combustion was confirmed using a metal single-cylinder engine with the same specifications (hereafter, metal engine). Figure 16 shows the relationship between the turbulent kinetic energy k_f obtained in monitoring with the optical engine and the combustion duration obtained in metal engine firing. The load of the metal engine was the indicated mean effective pressure high-loop (IMEP.H) from the compression stroke to the expansion stroke, and was 1000 kPa. The duration from ignition to a mass fraction burnt (MFB) of 10% was

defined as the early combustion duration θ_{ig} -MFB10, and k_f used the value at the ignition timing θ_{ig} . As shown, when k_f increased by the TCV, θ_{ig} -MFB10 decreased accordingly. The most effective TCV specifications compared to without TCV were UF 2 mm and UF 25 mm. In contrast, the LF 2 mm specification increased θ_{ig} -MFB10. The correlation coefficient between the turbulent kinetic energy k_f and θ_{ig} -MFB10 was 0.875, and the k_f obtained by the time filtering method exhibited correlation with combustion, which confirmed the reliability of the turbulent analysis method.

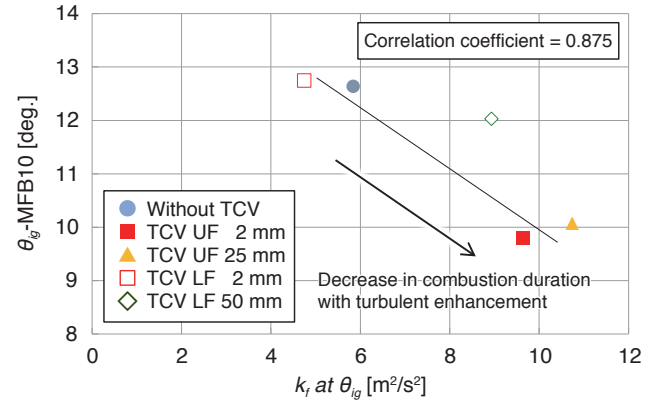


Fig. 16 Relationship between combustion duration on metal single-cylinder engine and turbulent kinetic energy, k_f obtained by time filtering method at ignition timing on optical single-cylinder engine, $Ne = 1500$ rpm IMEP.H = 1000 kPa

5. Application to Hybrid Engine Development

5.1. Standardization of Analysis Tools

To enhance ease of use, the constructed time filtering method was implemented as an in-cylinder flow analysis function in the commercial PIV auto analysis tool FtrPIV-Cruise made by Flowtech Research Inc. This tool features the ability to perform a series of processes such as PIV measurement setup, PIV analysis, statistical processing (turbulent analysis, etc.), and report creation using the same tool⁽²⁷⁾. A function was added to automatically perform turbulent analysis and create reports in web browser format simply by selecting the time filtering method using the graphical user interface (GUI). Simple setup and automation with a GUI allow shortening of the work time needed for turbulent analysis and report creation by approximately 30%, which enables timely feedback of verification results to engine development.

5.2. Verification of In-cylinder Flow Enhancement Concept

The engine specification is an inline 4-cylinder direct

injection engine with displacement of 2.0 L for a 2-motor hybrid. In order to achieve class-leading fuel efficiency, this engine has a compression ratio of 13.9 and is equipped with a cooled EGR system. To achieve quick combustion, the tumble flow is enhanced by the intake port, and a bowl-in piston is adopted that smoothly converts the tumble flow to an upward flow by the piston⁽²⁸⁾.

The intake port and piston shapes were examined using computational fluid dynamics (CFD) to enhance the in-cylinder flow. In order to verify this in-cylinder flow enhancement concept, the in-cylinder flow with early development and in-cylinder flow enhancement specifications were analyzed by PIV. Measurement was performed for 60 cycles at $Ne = 2000$ rpm WOT with a time resolution of 3 deg. BTDC (frame speed 8 kfps). CFD analysis used the commercial software VECTIS made by Ricardo Software, and the standard $k-\varepsilon$ model was applied to the turbulence model. Figure 17 compares the cycle-averaged in-cylinder flow between the early development and in-cylinder flow enhancement specifications. The velocity of the in-cylinder flow enhancement specification obtained by PIV at $\theta = 180$ deg. BTDC was higher than that of the early development specification, and the intake flow from the intake port was smoothly converted to an upward flow. Even at $\theta = 30$ deg. BTDC in the latter half of the compression stroke, the velocity of the in-cylinder flow enhancement specification obtained by PIV was fast, and these tendencies were similar to the CFD analysis results.

Figure 18 compares the turbulent kinetic energy k_f with early development and in-cylinder flow enhancement specifications obtained by PIV. Compared to the early development specification, the k_f with the in-cylinder flow enhancement specification indicated a large value

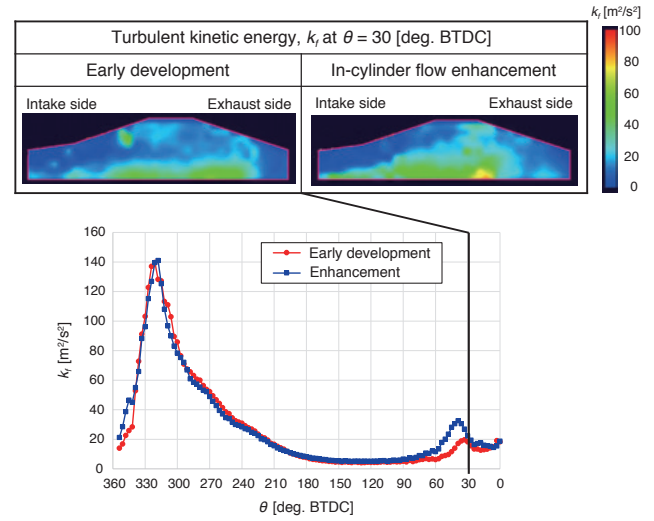


Fig. 18 Comparison of turbulent kinetic energy k_f with early development and in-cylinder flow enhancement specification obtained by PIV, $Ne = 2000$ rpm WOT

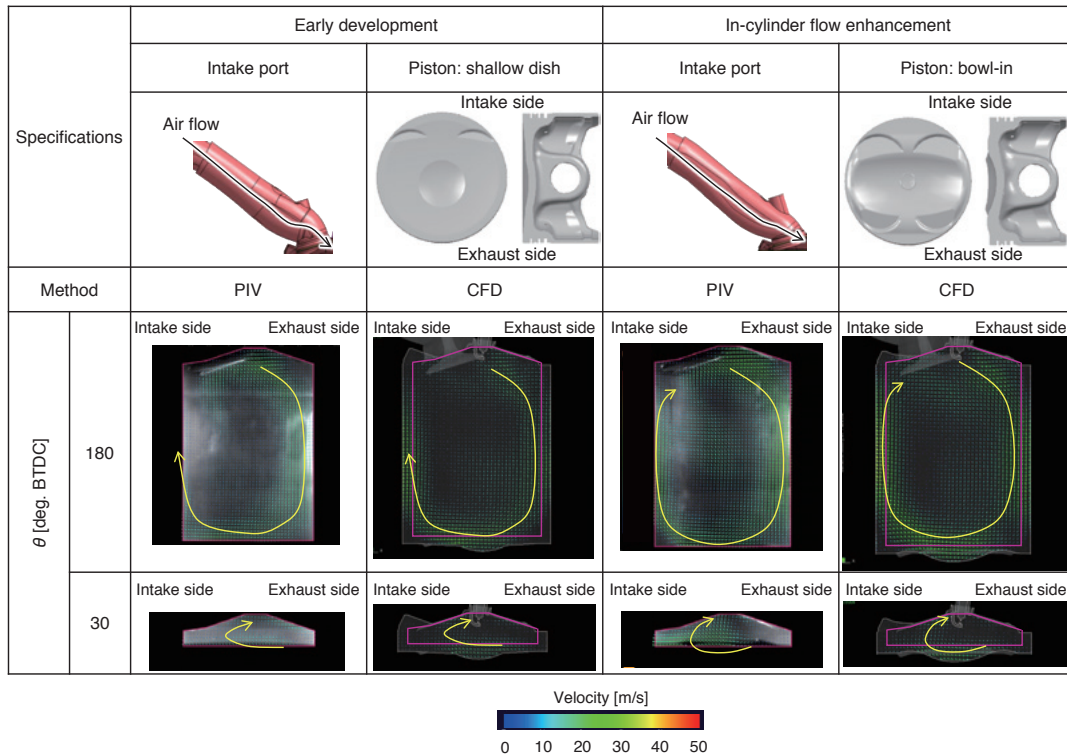


Fig. 17 Comparison of in-cylinder flow with early development and in-cylinder flow enhancement specification obtained by PIV and CFD, $Ne = 2000$ rpm WOT

from $\theta = 60$ deg. BTDC onward. In the k_f distribution at $\theta = 30$ deg. BTDC, the k_f with the in-cylinder flow enhancement specification tended to be higher than that with the early development specification near the center of the combustion chamber.

Figure 19 compares the turbulent kinetic energy k_f with the time filtering method and the cycle average method. With the time filtering method, changing from the early development specification to the in-cylinder flow enhancement specification increased k_f by approximately 21.9%. However, this increase was 24.9% with the cycle average method, so the cycle average method estimated k_f approximately 3% larger.

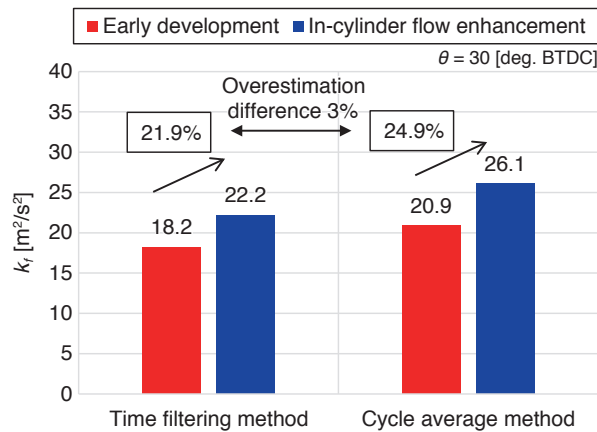


Fig. 19 Effect of time filtering method, $N_e = 2000$ rpm WOT at $\theta = 30$ deg. BTDC

6. Conclusion

A time filtering method that makes use of high time resolution by high-speed PIV was proposed as a new turbulent component decomposition method, and application to engine in-cylinder flow was examined. In addition, the time filtering method was implemented in a commercial PIV auto analysis tool, and the following conclusions were obtained through application to hybrid engine development.

- (1) The time resolution was artificially varied, and the turbulent kinetic energy characteristics were verified. It was found that when the time resolution was 5 kHz or more, there was no great effect on the cut-off frequency used to decompose the mean flow component and the turbulent component in this flow field.
- (2) The frequency at which the slope of the frequency characteristic of the in-cylinder flow changes was found to correspond to the relatively large-scale in-cylinder flow induced by the combustion chamber and intake port shapes and the piston speed. Furthermore, the cut-off frequency can be uniquely determined by using the

inverse of the integral time scale of in-cylinder flow as the cut-off frequency.

- (3) The time filtering method was used to analyze the turbulent characteristics of a tumble control valve, which is an intake switching device. The results confirmed that the flame structure was classified as corrugated flamelets, and combustion promoting effects can be obtained by turbulence. There was high correlation between the turbulent kinetic energy and the metal engine combustion duration, which confirmed the reliability of the turbulent analysis method.
- (4) The time filtering method was implemented in a commercial PIV auto analysis tool, and the analysis tool was standardized to enhance ease of use. This tool was then applied to hybrid engine development and verified the concept of in-cylinder flow enhancement for realizing quick combustion.

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List of Symbols

E	Mean square of u and v velocity components [m^2/s^2]
E_u	Energy of u velocity component [m^2/s^2]
f	Frequency [Hz]
f_c	Cut-off frequency [Hz]
f_{c_slope}	Cut-off frequency at where spectrum slope changes [Hz]
$f_{c_ \theta = 270}$	Cut-off frequency which was calculated from

	inverse of integral time scale of the in-cylinder flow at crank angle 270 deg. BTDC [Hz]	UF	Upper Flow
f_N	Nyquist frequency which is half frequency of time resolution [Hz]	WOT	Wide Open Throttle
G_{plate}	Gap from tip of tumble plate to intake valve stem [mm]		
h_p	Height of combustion chamber from top of pent-roof to piston surface top at bore center [mm]		
k	Turbulent kinetic energy [m^2/s^2] (Subscripts f : time filtering method, c : cycle average method)		
l	Turbulent integral length scale [mm]		
l_F	Flame thickness [mm]		
L_{plate}	Length of tumble plate [mm]		
Ne	Engine speed [rpm]		
P_{cyl}	In-cylinder pressure [kPa]		
P_{intake}	Intake manifold pressure [kPa]		
R_{uu}	Auto correlation coefficient of u velocity component		
S_L	Laminar burning velocity [m/s]		
T_{cyl}	In-cylinder gas temperature [K]		
u	Instantaneous velocity component in x direction [m/s]		
u'	Turbulent intensity [m/s]		
u'_f	Turbulent velocity component in x direction decomposed by time filtering method [m/s]		
u_p	Instantaneous piston speed [m/s]		
v	Instantaneous velocity component in y direction [m/s]		
v'_f	Turbulent velocity component in y direction decomposed by time filtering method [m/s]		
x	Coordinate to bore direction [mm]		
y	Coordinate to stroke direction [mm]		
θ	Crank angle [deg.]		
θ_{ig}	Ignition timing [deg. BTDC]		
ν_F	Dynamic viscosity of mixture [m^2/s]		
τ	Integral time scale of in-cylinder flow [ms] (Subscripts u : x direction, v : y direction)		
τ_i	Turbulent integral time scale [ms]		
τ_p	Characteristic time of piston speed [ms]		
φ	Equivalent ratio [-]		

List of Abbreviations

BDC	Bottom Dead Center
CFD	Computational Fluid Dynamics
e-fuel	Erneuerbarer strom - fuel
EGR	Exhaust Gas Recirculation
fps	frame per second
GUI	Graphical User Interface
IMEP.H	Indicated Mean Effective Pressure High-loop
MFB	Mass Fraction Burnt
LF	Lower Flow
PIV	Particle Image Velocimetry
PRF	Primary Reference Fuel
SI	Spark Ignition
TCV	Tumble Control Valve
TDC	Top Dead Center (Before TDC: BTDC)

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