Further Investigation into the Statistical Properties of Reciprocating Engine Turbulence*

Andrea Emilio CATANIA**, Claudio DONGIOVANNI**
and Antonio MITTICA**

In order to further develop appropriate general methodologies for analyzing the properties of reciprocating engine turbulence and to enhance a fundamental understanding of the in-cylinder turbulent flow phenomena, basic aspects in the correlation and frequency spectral structure of engine turbulence were considered, particularly the estimation of a characteristic time scale, the macro time scale, or simply the time scale, on the analogy of the integral time scale of turbulence for stationary flows. Applications to the study of turbulence properties in an automotive engine under variable swirl flow conditions are presented, including an insight into the probability density function of the engine turbulence. In addition, evaluation of cross terms, which arise in the momentum equation when unconventional averaging is used for the in-cylinder velocity, was provided with reference to the cycle-resolved data reduction procedure previously developed.

Key Words: Engine Turbulence, Autocorrelation, Spectral Structure, Time Scales, Probability Density Function, Non-Reynolds Cross Terms

1. Introduction

Many experimental investigations on engine turbulence have been carried out using different measurement and data reduction methods, and an extensive bibliography was reported by Catania and Mittica^{(1),(2)}. Most of them are Eulerian single-point studies of turbulent flow at different locations in engine cylinders. Attempts have lately been made to measure instantaneous spatial-velocity distributions in engines⁽³⁾⁻⁽⁶⁾. Simultaneous two-point measurements of in-cylinder velocity at various distances were taken by Fraser and Bracco⁽⁷⁾ for direct measurement of turbulence length scales.

However, further research is needed, especially as far as methodological aspects of engine turbulence analysis are concerned, in order to obtain better knowledge of in-cylinder turbulent flow phenomena, and to assess more reliable and effective turbulence models for engine flow application. Therefore, both single-point and simultaneous multiple-point investigations into reciprocating engine turbulence are of primary importance.

More specifically, correlation and spectral structure of engine turbulence generally have not been given due attention, perhaps owing to the lack of procedures for evaluating characteristic temporal and spatial scales of turbulence in nonstationary flows. Furthermore, models for cycle-resolved engine turbulence have not yet been developed, due to the presence of "cross terms" (or non-Reynolds stresses) which are still not well known, in the momentum equation.

A general method for analyzing the time-correlation and frequency-spectral structure of turbulence in internal combustion engines was recently developed by Catania and Mittica⁽²⁾. It is based on an alternative definition of the Eulerian autocorrelation coefficient so as to reduce this to an even function of only the separation time (over which the turbulent fluctuation is correlated), independent of the time instant, within specific correlation periods throughout the engine cycle.

This procedure, easily extensible from temporal to spatial records, was shown to be a refined version of a previous approach used by the same authors^{(1),(8)}, and was proven to be suitable for studying the average statistical properties of segmented nonstationary turbulence sample records. It was compared to the one^{(9),(10)} based on the rough application of the stan-

* Received 19th February, 1991

** Dipartimento di Energetica-Politecnico di Torino
Turin, Italy

dard approach for stationary flows using the conventional definition of the nonstationary autocorrelation coefficient, which generally led to a time-dependent and uneven function of the separation time. The method was applied to the analysis of the micro time scale of turbulence in the cylinder of an automotive engine. The spatial distribution of this parameter and the influence of the cylinder wall on it were also investigated⁽²⁾.

A problem that is still open concerns the more appropriate definition and estimation of another characteristic time scale of engine turbulence, the macro time scale, or simply the time scale, on the analogy of the integral time scale of turbulence defined in stationary flows. The present work addresses this problem in particular as well as other basic aspects in the analysis of correlation and spectral structure of engine turbulence. Specific applications to the study of turbulence properties in an automotive engine under variable swirl flow conditions are reported, including an insight into the turbulence probability density function. Furthermore, an evaluation of cross terms, which arise in the momentum equation when the in-cylinder velocity is filtered, is provided with reference to the cycle-resolved data reduction procedure previously developed⁽¹¹⁾.

2. Turbulence Quantities

2.1 Turbulent velocity fluctuations

Any component of the instantaneous velocity in the cylinder of a reciprocating internal combustion engine can be split into a mean nonstationary part along its direction and a turbulent fluctuation about this, so that, with reference to the engine cycle i ,

$$U_i(t) = \bar{U}_i(t) + u_i(t), \quad (1)$$

where $\bar{U}_i(t)$ is the cycle-resolved mean velocity at time t (corresponding to the crank angle θ) and $u_i(t)$ is the cycle-resolved velocity fluctuation about the mean.

In the conventional approach, Eq. (1) becomes $U_i(t) = U_E(t) + u_{i,E}(t)$, where the mean velocity $U_E(t)$, that is, the ensemble-averaged velocity, is the same in each cycle; the turbulent fluctuation $u_{i,E}(t)$ will be referred to as the instantaneous velocity fluctuation.

The filtering technique here considered to determine the in-cycle nonstationary mean velocity from each single data record is based on the curve fitting of mean value estimates obtained by time-averaging the velocity in discrete crank-angle intervals throughout the engine cycle^{(11),(12)}. It can be succinctly named time-average filtering (TAF), and is able to yield a cycle-resolved turbulent velocity fluctuation with a zero mean value, either as a time-averaged value in each cycle or, virtually, as an ensemble-averaged value

over many cycles⁽¹⁾. This technique is consistent with the present approach of correlation analysis and includes, as limit cases, the Reynolds averaging operators. Therefore, the cycle-resolved turbulent fluctuation referred to in this study was obtained by time-average filtering.

The symbol $\bar{f}_i$ will be used to represent the time average of any variable $f_i(t)$ (or $f_i(\theta)$) in the interval T (or in the crank-angle interval Θ),

$$\bar{f}_i = \frac{1}{T} \int_{t_0-T/2}^{t_0+T/2} f_i(t) dt = \frac{1}{\Theta} \int_{\theta_0-\Theta/2}^{\theta_0+\Theta/2} f_i(\theta) d\theta. \quad (2)$$

One has, within the TAF procedure

$$\tilde{U}_i(t) = I[\bar{U}_i], \quad (3)$$

where the symbol $I[\]$ indicates the interpolation operator of the bracketed values considered at the various instants t_0 in the middle of each interval T , so as to have $\bar{U}_i = \tilde{U}_i^{(1)}$.

Furthermore, in the filtering approach,

$$\tilde{U}_i(t) = U(t) + \tilde{u}_i(t), \quad (4)$$

where $U(t)$ is the ensemble-averaged mean velocity at time t and $\tilde{u}_i(t)$ is the so-called low-frequency cyclic fluctuation of the mean velocity.

The symbol $\langle f_i(t) \rangle$ will be used to represent the ensemble average of the variable $f_i(t)$ at time t over N records (or cycles)

$$\langle f_i(t) \rangle = \frac{1}{N} \sum_{i=1}^N f_i(t). \quad (5)$$

Therefore, within the TAF procedure

$$U(t) = \langle \tilde{U}_i(t) \rangle = \langle I[\bar{U}_i] \rangle = I[\langle \bar{U}_i \rangle]. \quad (6)$$

2.2 Probability density function

For a statistical analysis of nonstationary turbulence properties evaluated by short time-averaging, so as to have negligible distortions due to the unsteadiness of the random process, the probability density function of the turbulent fluctuation in the period T with midpoint at time t_0 can be estimated by

$$p(u, T, t_0) = \lim_{\Delta u \rightarrow 0} \frac{P(u, \Delta u, T, t_0)}{\Delta u} = \lim_{\Delta u \rightarrow 0} \frac{\langle T_i \rangle}{T \Delta u}, \quad (7)$$

where $P(u, \Delta u, T, t_0)$ is the probability that the velocity fluctuation assumes values between $u - \frac{\Delta u}{2}$ and $u + \frac{\Delta u}{2}$ during the time interval T , and T_i is the total portion of T spent by $u_i(t)$ in the range of values $\left[u - \frac{\Delta u}{2}, u + \frac{\Delta u}{2} \right]$. It can be easily verified that Eq. (7) gives a first moment of the cycle-resolved turbulent fluctuation in T equal to $\langle \tilde{u}_i \rangle$, that is, zero within the TAF procedure, and a second moment equal to the square of the turbulence intensity in T : $u' = \sqrt{\langle u_i^2 \rangle}$.

2.3 Cross terms

Non-Reynolds stresses, or cross terms, arise from the products of the turbulent fluctuation and the mean velocity, when ensemble-averaging and time-averag-

ing operations are performed on the instantaneous momentum equation, subsequent to the introduction of the velocity decomposition given by Eq.(1). An insight into the normal Reynolds-like stress (per unit density) $u'^2 = \langle \bar{u}_i^2 \rangle$ and $U_{rms}^2 = \langle \bar{u}_i^2 \rangle$ was given by Catania and Mittica^{(11),(12)}. Here an evaluation of the normal cross term $\langle u_i \bar{U}_i \rangle$ will be presented to give an idea of the relevance which non-Reynolds stresses may have in the time-average filtering approach.

It should be pointed out that in the Gaussian time filtering approach⁽¹³⁾ the product of the mean (or filtered) velocity and turbulent fluctuation, that is, a simple linear approximation of cross terms, is not zero. However, in the time-average filtering approach a similar approximation gives zero as a result.

$$\bar{u}_i \bar{U}_i \approx \bar{U}_i(t_0) \bar{u}_i = 0. \quad (8)$$

If $\bar{U}_i(t)$ is a regular continuous function of time with continuous time derivatives, the following approximation can be easily verified:

$$\langle \bar{u}_i \bar{U}_i \rangle \approx \left. \frac{dU}{dt} \right|_{t_0} \overline{(t-t_0)\Sigma(t)} + O\left[\left. \frac{d^2 U}{dt^2} \right|_{t_0} \times \overline{(t-t_0)^2 \Sigma(t)} \right], \quad (9)$$

where $\Sigma(t) = \langle u_i(t) \rangle$, and the symbol $O[]$ indicates the order of magnitude. $\Sigma(t)$ should be virtually negligible, depending on the number of sample data records⁽¹⁾, and so the cross term may be of a certain significance, even though neglected as an approximation, where $\left. \frac{dU}{dt} \right|_{t_0}$ is very high.

2.4 Autocorrelation and autospectral density coefficients

The Eulerian time autocorrelation coefficient of the turbulent fluctuation $u_i(t)$ in the period T , centered on the instant t_0 , can be defined by⁽²⁾

$$R(T, t_0, \tau) = \frac{1}{u'^2} \left\langle \frac{1}{T-\tau} \int_{t_0-T/2+\tau/2}^{t_0+T/2-\tau/2} u_i(t-\tau/2) \times u_i(t+\tau/2) dt \right\rangle, \quad (10)$$

where the separation-time variable τ in T ranges from zero to $\tau_{max} = T/2$. The factor u'^2 , normalizing the autocorrelation function to unity at $\tau=0$, is the square of the expected value of the time-averaged turbulence intensity in the interval T .

With reference to a collection of sample records $u_i(t)$ from a stationary random process, the right-hand side of Eq.(10) can be verified to be⁽²⁾ an unbiased and consistent estimate of the autocorrelation coefficient $R(\tau)$, independent of T and of t_0 . For a nonstationary random process, $R(T, t_0, \tau)$ might be averaged with respect to t_0 and T in order to be reduced to only a function of τ . However, in this way, information about the time evolution of the process is lost. To analyze the structure of turbulence through-

out the engine cycle, discrete intervals T should be considered, long enough to provide statistical validity of the results, but, at the same time, short enough to have a negligible effect of the turbulence time dependence.

Equation (10) treats the turbulent fluctuation as quasi-stationary in the period T ; namely the statistical character of the fluctuation is considered, on average, as homogeneous with respect to time in T . This equation, in fact, evaluates a time average of the expected value of the correlation function.

The autocorrelation coefficient so defined is an even function of τ , independent of the instant t in the period T . Consequently, the standard approach for stationary flows can be used to determine the micro time scale of turbulence, so as to obtain an average value of this parameter in each correlation interval and, at the same time, its temporal evolution throughout the engine cycle.

Consistent with the conventional ensemble-averaging approach, the following nonstationary autocorrelation coefficient can be introduced, as in Eq. (10), so as to give an even function of τ :

$$R_E(t_0, \tau) = \frac{\langle u_i(t_0-\tau/2) u_i(t_0+\tau/2) \rangle}{u'(t_0-\tau/2) u'(t_0+\tau/2)}, \quad (11)$$

where $u'(t_0-\tau/2)$ and $u'(t_0+\tau/2)$ are the root-mean-square values of the turbulent fluctuations at the instants indicated. This expresses a different time dependence with respect to Eq.(10); in particular, t_0 in Eq.(11) indicates any instant of time, while in Eq. (10) it indicates the position in time of the period over which the expected value of the autocorrelation coefficient is evaluated.

The normalized energy spectral density function of the turbulent fluctuation $u_i(t)$ in the interval T can be obtained from Eq.(10) as follows:

$$E(T, t_0, f) = 4 \int_0^{\tau_{max}} R(T, t_0, \tau) w(\tau) \cos(2\pi f \tau) d\tau, \quad (12)$$

where f is the frequency, and $w(\tau)$ is a window function of the rectangular or Hamming type. It is the one-sided spectral density function given by twice the Fourier cosine transform of the autocorrelation coefficient, provided this is symmetric in time separation⁽¹⁴⁾.

2.5 Eulerian time scales

An estimate of the micro time scale of the turbulent fluctuation in the interval T around t_0 is given by

$$\lambda_\tau(T, t_0) = \frac{1}{\sqrt{2\pi^2 \int_0^{f_{max}} f^2 E(T, t_0, f) df}}, \quad (13)$$

where f_{max} is the maximum frequency for which $E(T, t_0, f)$ is still finite and significant (f_{max} is less than, or at least equal to, the Nyquist frequency associated

with the sample data). By analogy with quasi-steady flows⁽¹⁵⁾, this parameter can be considered to be a statistical measure of the most rapid changes that occur in the fluctuations of $u_i(t)$ during the period T .

Another characteristic turbulence time scale can be evaluated, on the analogy of the integral time scale defined in stationary turbulent flows, by taking the integral of R (for simplicity, the dependence on time variables is omitted in the symbol) over τ from 0 to $\tau_{\max}$, when R has positive values and decays over a sufficiently long period:

$$\Lambda_r(T, t_0) = \int_0^{\tau_{\max}} R(T, t_0, \tau) w(\tau) d\tau. \quad (14)$$

From Eq. (12), $\Lambda_r(T, t_0) = E(T, t_0, 0)/4$.

However, if the autocorrelation coefficient has a negative region and then decays with rapidly damped fluctuations over a characteristic period, the time separation τ over which the fluctuations about the mean velocity remain correlated can be considered as that at which R is a minimum⁽¹⁶⁾:

$$\Delta_r(T, t_0) = \tau|_{R(T, t_0, \tau)_{\min}}. \quad (15)$$

3. In-Cylinder Velocity Measurements

3.1 Experimental system and procedure

The velocity data were acquired in the direct-injection automotive diesel engine previously described⁽¹⁷⁾. The engine, derived from a production engine at Fiat Research Center, had a 75.5-mm bore, an 83.5-mm stroke and a compression ratio of 16. The combustion bowl in the piston was relatively wide and flat (the throat and the maximum depth of the bowl were ≈ 60 mm and ≈ 8 mm, respectively, the clearance height at TDC being 0.7 mm), so a nearly squish-free flow was detected at the end of compression⁽¹⁷⁾ and, consequently, no sensible squish effect on turbulence should be expected⁽¹²⁾. The induction system was made up of two tangential ducts inducing a swirl in the same direction and valves of the same size and lifts. The different swirl flow conditions considered produced by two configurations of the intake system

Fig. 1 Schematic of the intake system

(Fig. 1): with only one duct operating (intake I) and with both ducts operating (intake I & II), at the engine speeds $\omega = 167.6$ rad/s (1 600 rpm) and $\omega = 314.2$ rad/s (3 000 rpm).

The measurements were carried out using an advanced HWA technique, under motored conditions, in the crank-angle interval $\Delta\theta = 0-360$ deg, that is, from the start of the induction stroke (exhaust TDC) to the end of the compression stroke (compression TDC).

Based on the results of previous investigations^{(2),(12),(17)}, one measurement location was considered to be adequate for the purposes of the present study, which is mainly oriented to the data analysis method. The measurement location selected, on the right-hand side in Fig. 1, at a level of ≈ 3.5 mm underneath the flat cylinder head and at a radial distance $r = 18$ mm from the cylinder axis, was of particular interest. In fact, being approximately in the middle of the right-hand section of the combustion chamber at TDC (virtually equidistant from the top of the cylinder and the bottom of the bowl at TDC), it was in the core of the cylinder charge at TDC of compression. Furthermore, it was upstream of the valve in the case of intake I and downstream of the other valve in the case of intake I & II. Thus, in the former case, the sensing wire was directly exposed to the early-stage (or secondary) intake jet flow, which had a strong velocity component in the plane perpendicular to the cylinder axis, while in the latter case, the wire also sensed the main swirling flow during the induction stroke⁽¹⁷⁾.

The sensing wire, $10\ \mu\text{m}$ in diameter and 1.5 mm long, was parallel to the cylinder axis, so as to measure the velocity component in a plane perpendicular to this, that is, the main velocity component for the engine geometry considered. Furthermore, the mean velocity obtained could be considered as approximately equal to the tangential velocity component in the last part of the compression stroke⁽¹⁷⁾. The wire was made up of Pt-Rh(10%) and operated at a constant average temperature of 700°C .

A number of sample data records from 60 to 100 were acquired at a rate of one datum per 0.2 crank-angle degree, which is high enough to give a virtually continuous velocity trace in every cycle.

3.2 Experimental uncertainties

The method used to compute gas velocity from the anemometer output is different⁽¹²⁾ from all of those semiempirical corrections considered by Witze⁽¹⁸⁾. It was validated in nonstationary as well as stationary conditions, at different gas pressures and temperatures and for different wire orientations⁽¹⁷⁾.

Based on the previous findings and on the average

deviations of repeated measurement sets, a maximum uncertainty of ± 10 percent could be ascribed to the computed velocities.

The statistical uncertainty in autospectra estimates depends on the reciprocal square root of the number of data records and could be considered as less than 12 percent in the results presented.

However, it can be expected that the implications of the results and the related conclusions are not influenced by the overall experimental uncertainty.

4. Experimental Results

Figure 2 gives examples of velocity fluctuation patterns versus crank angle obtained from the same set of in-cylinder velocity data records, for three consecutive engine cycles; more specifically, Fig. 2 (a) shows the instantaneous velocity fluctuation, Fig. 2(b) the cycle-resolved turbulent fluctuation and Fig. 2(c) the mean-flow cyclic fluctuation. These two latter fluctuation distributions were obtained using an averaging interval of 12 crank-angle degrees in the TAF procedure^{(11),(12)}.

It is evident from Fig. 2 (a) that the time average of the instantaneous velocity fluctuation is not zero either in the whole crank-angle interval of 360 deg, or in part of it. Larger structures than those evident in cycle-resolved turbulence are apparent in the instantaneous velocity fluctuations. These can be related to the lower-frequency cyclic variations in the mean flow by comparing Figs. 2 (a) and (c). No sensible squish

effect on the velocity fluctuations appears from Fig. 2.

A modified version of the algorithm developed by Rader⁽¹⁹⁾ and by Rabiner et al.⁽²⁰⁾ was used for correlation and spectrum function estimation. Crank-angle correlation intervals of 60 deg were considered throughout the intake and compression strokes for analysis.

Figure 3 shows patterns of autocorrelation and

(b) Cycle-resolved turbulent fluctuations

(a) Instantaneous velocity fluctuations

(c) Mean-velocity cyclic fluctuations

Fig. 2 Velocity fluctuations in individual cycles

autospectral density coefficients obtained for the values of the crank angle θ_0 (corresponding to the time t_0) and of the crank-angle correlation interval Φ (corresponding to the period T) indicated. The intake system configuration and the engine speed ω , are also evidenced in the figure. The autocorrelation coefficients are plotted versus crank-angle separation

$\theta - \theta_0$ (corresponding to time separation τ), and the autospectra are plotted versus frequency f .

More specifically, Figs. 3(a) and (d) compare the two definitions of autocorrelation coefficient, given by Eqs.(10) and (11), through their application to the cycle-resolved turbulent fluctuation ($R[u_i]$ and $R_E[u_i]$, respectively) as well as to the instantaneous

Fig. 3 Autocorrelation (a, d) and autospectral density (b, c, e, f) coefficients

velocity fluctuation ($R[u_{i,E}]$ and $R_E[u_{i,E}]$, respectively) derived from the same sample data records. A continuously decaying pattern is shown by $R[u_{i,E}]$ in Fig. 3(d); a similar trend can be observed in Fig. 3(a) for the same quantity. $R[u_i]$ shows a decaying pattern with rapidly damped fluctuations in both Figs. 3(a) and (d). More complex decaying patterns are presented by $R_E[u_i]$ and $R_E[u_{i,E}]$, as a general results. The normalized energy spectral density functions obtained from the autocorrelation coefficients of Figs. 3(a) and (d) are reported in ordinary (Figs. 3(b) and (e)) and logarithmic scales (Figs. 3(c) and (f)) to give emphasis to the lower or to the higher frequency components, respectively. The spectra E (for simplicity, the dependence on time and frequency variables is omitted in the symbol) are nonnegative for all values of f , as can be easily demonstrated. However, meaningless negative and fluctuating values may be present in $E_E^{(2)}$, as shown by $E_E[u_i]$ in Figs. 3(b) and (e) and also by $E_E[u_{i,E}]$ in Fig. 3(b) (E_E represents the normalized energy spectral density function obtained by Eq.(12) using Eq.(11) instead of Eq.(10) for the autocorrelation coefficient). Furthermore, $E_E[u_i]$ indicates the presence of a significant low frequency content in the energy spectrum of the cycle-resolved turbulent fluctuation, whereas most of the low-frequency energy should have been removed by time-average filtering, as shown by $E[u_i]$. This may be ascribed to the different time dependence expressed by Eq.(11) with respect to Eq.(10) for nonstationary flows. These results confirm the physical consistency of the proposed autocorrelation coefficient definition and, at the same time, the limits of the nonstationary one in the standard approach for time-frequency spectral analysis of engine turbulence.

It is interesting to observe in Figs. 3(c) and (f) that the tails of the spectra $E[u_i]$ and $E[u_{i,E}]$ show virtually the same decaying pattern vs frequency, which means that the dissipative components of cycle-resolved turbulence and instantaneous velocity fluctuation have similar energy distributions. For analogy with wave number energy spectra, the slopes $-5/3$, -3 and -4 are marked in these figures. It is worth pointing out⁽²¹⁾ that the Kolmogorov $k^{-5/3}$ inertial range of the wave number spectrum was deduced within the energy cascade concept and verified in the small scales of a flow with a sufficiently high Reynolds number; on the other hand, a Kraichnan k^{-3} and a Saffman k^{-4} wave number spectra can be predicted within the "enstrophy**1" cascade concept⁽²¹⁾ which applies to environments with high local vorticity gradients and thus a strong eddy interaction, implying a high flux of vorticity into the small scales. The results of Fig. 3(c) refer to the early stage of induc-

tion and those of Fig. 3(f) to the last stage of compression, in the case of intake I, at 3000 rpm. Similar results were obtained for other crank angles, swirl flow conditions, and also measurement locations along a radial direction at the same axial level considered. Carrying out measurements at different levels, sufficiently distant from the cylinder head, was not possible due to the shallow bowl (which would have required flow disrupting modifications in the piston to accommodate the probe). However, this is a challenging undertaking for engine designs with deeper bowls and a subject for our continuing work in this research field.

Figure 4 illustrates the dependence of the autocorrelation coefficient and time scales on the correlation interval length for both cycle-resolved (Fig. 4(a)) and conventional (Fig. 4(b)) turbulent fluctuations. The autocorrelation curve R_E is also reported for comparison. The macro time scale was evaluated as separation time Δ_τ at which R is minimum for u_i and as integral time scale Λ_τ for $u_{i,E}$. In general, $R[u_i]$ was almost insensitive to Φ (Fig. 4(a)), whereas $R[u_{i,E}]$ showed a more sensible dependence on this parameter (Fig. 4(b)). This led to a slight dependence of Δ_τ on Φ (top right of Fig. 4(a)), and to a nonnegligible variation of Λ_τ with Φ (top right of Fig. 4(b)). In many cases Λ_τ displayed a wide maximum at $\Phi=60$ deg. It can be inferred from the figure that the micro time scale λ_τ is virtually independent of Φ for both u_i and $u_{i,E}$, as is the obscuring parabola in the vertex of the correlation curves⁽¹⁵⁾. This confirms both the choice of $\Phi=60$ deg for correlation interval in the present study and the validity of the results on the small-scale structure of engine turbulence previously reported using $\Phi=30$ deg^{(2),(12)}.

The same deductions can be drawn from Fig. 5 which reports the autospectral density coefficients corresponding to the correlations of Fig. 4.

Figures 6-9 compare the results obtained for the different swirl flow conditions considered to show any influence of the induction system configuration (that is, the closure of one intake duct) and of the engine speed on the structure of the cycle-resolved turbulence (Figs. 6(a), 7(a), 8(a) and 9(a)), and the fluctuating motion in its more conventional statistical sense (Figs. 6(b), 7(b), 8(b) and 9(b)), including the mean-flow cyclic variation.

Figures 6 and 7 show the autocorrelation coefficients and their respective Fourier cosine transforms evaluated in the correlation interval of 60 degrees before compression TDC, centered on $\theta_0=330$ deg. Taking into account the fact that the squish

**1 The enstrophy is defined as the variance of the vorticity⁽²¹⁾.

effect on turbulence is virtually negligible due to the relatively wide and shallow combustion bowl⁽¹²⁾, it can be inferred from Figs. 6(a) and 7(a) that the cycle-resolved turbulence structure at the last stage of

compression is more sensitive to the engine speed than to the intake system configuration. The same conclusion can be reached for the instantaneous velocity fluctuation, though a larger dispersion is shown by

Fig. 4 Autocorrelation coefficients of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

Fig. 5 Normalized energy spectral density functions of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

Fig. 6 Autocorrelation coefficients of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

Figs. 6(b) and 7(b).

Similar results were obtained throughout the induction and compression strokes, as can be deduced from the distributions of the time scales reported in Figs. 8 and 9.

At both engine speeds the fine structure of the cycle-resolved turbulence was shown to be virtually uniform from the start of induction to the end of compression (Fig. 8(a)). A similar pattern was displayed by the macro time scale $\Delta\tau$ at the higher

Fig. 7 Normalized energy spectral density functions of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

Fig. 8 Micro time-scale distributions of cycle-resolved turbulence (a) and instantaneous velocity fluctuation (b)

Fig. 9 Macro time-scale distributions of cycle-resolved turbulence (a) and instantaneous velocity fluctuation (b)

Fig. 10 Normalized probability density function of cycle-resolved turbulence

Fig. 11 Cross-term distributions

speed, while at the lower speed it exhibited an increasing trend, particularly for intake I (Fig. 9(a)). This evolution in time may be due to the fact that smaller structures decay more rapidly than larger ones, which is more evident at the lower speed when the process lasts longer.

The time scales of the unfiltered turbulent fluctuation Figs. 8(b) and 9(b) show an increasing trend on the main part of compression, though with more uniform distributions at the higher speed, probably due to the eddy stratification in the cylinder charge; however, it has to be recalled that cycle-to-cycle variations in the mean flow are also included in the instantaneous velocity fluctuation. These scales show a decreasing tendency at the end of compression, owing to a higher repeatability of the in-cycle mean flow and to an average eddy size reduction in the shallow combustion bowl at that stage. Low-frequency larger structures are present in the early part of induction and before the end of compression.

Figure 10 reports the normalized probability density function of cycle-resolved turbulence in the periods T (same length as that used for time-average filtering) at the crank angles θ_0 indicated. For comparison, the standardized Gaussian probability density function⁽¹⁴⁾ is plotted with a dashed line. This is fairly well approached from the induction to the end of

compression, with negligible distortions due to the unsteadiness of the flow.

Figure 11 presents, for the engine conditions considered, the pattern of $I[\langle u_i \tilde{U}_i \rangle]$ versus crank angle (for simplicity, the notation $u_i \tilde{U}_i$ is indicated in the figure) normalized by the squared mean piston speed V_p . The values of the cross term are negligible during compression and significant only at the early stage of induction. In fact, strong mean-velocity gradients occur there, mainly for intake I at 167.6 rad/s, due to an intense secondary jet flow with velocities up to 100 m/s⁽¹⁷⁾. In any case, this term is everywhere as low as at least one order of magnitude, or more, less than the normalized second moment of the cycle-resolved turbulent fluctuation⁽¹²⁾.

5. Conclusions

The general method previously developed for autocorrelation and autospectra estimation of non-stationary turbulence, which gives information on the average statistical properties of segmented turbulent fluctuation sample records, was shown to be suitable for the analysis of the macro time scale, or simply the time scale, of reciprocating engine turbulence.

This parameter, being a measure of the characteristic time separation over which the fluctuation about the nonstationary mean velocity remain correlated on average, was estimated as a stationary-flow integral-time-scale analog, for the fluctuating motion in its more conventional statistical sense, and as time separation at which the autocorrelation coefficient is a minimum, for the cycle-resolved turbulent fluctuation. In fact, for the instantaneous velocity fluctuation the autocorrelation coefficient generally showed a continuous decaying pattern over a sufficiently long period, due to the presence of the so-called low-frequency mean-flow cyclic variations, whereas, for the cycle-resolved turbulent fluctuation it initially decreased to negative values, (implying a tendency of the fluctuation to be in opposite directions more often than in the same direction,) and then decayed with rapidly damped fluctuations over a relatively long period. A

60-deg crank-angle interval can be considered as a good compromise for a sufficiently long period in engine flows.

The application of the method to the analysis of the turbulence structure in an automotive engine under variable swirl flow conditions, as produced by deactivating one of the two intake ducts and/or by changing the engine speed in a range of practical interest, gave reasonable and physically consistent results. The time-frequency spectral structure of the in-cylinder turbulence was shown to be more sensitive to the engine speed than to the intake system configuration.

The normalized probability density function, for statistical analysis of short time-averaged turbulence properties, was very close to the standardized Gaussian distribution, with negligible distortion due to the unsteadiness of the flow. This supported the validity of the correlation definition proposed.

The time-average filtering technique used to separate the in-cycle mean flow from turbulence, consistent with the present approach of spectral analysis, is expected to yield negligible cross terms which arise in the momentum flow equation when non-Reynolds averaging procedures are considered.

Acknowledgments

Financial support for this work was provided by M.U.R.S.T. (Ministero dell'Università e della Ricerca Scientifica e Tecnologica).

References

- (1) Catania, A. E. and Mittica, A., Extraction Techniques and Analysis of Turbulence Quantities from In-Cylinder Velocity Data, *Trans. ASME, J. Eng. Gas Turbines Power*, Vol. 111 (1989), p. 466.
- (2) Catania, A. E. and Mittica, A., Autocorrelation and Autospectra Estimation of Reciprocating Engine Turbulence, *Trans. ASME, J. Eng. Gas Turbines Power*, Vol. 112 (1990), p. 357.
- (3) Ikegami, M., Shioji, M. and Nishimoto, K., Turbulence Intensity and Spatial Integral Scale During Compression and Expansion Strokes in a Four-Cycle Reciprocating Engine, *SAE Paper No. 870372* (1987).
- (4) Dinsdale, S., Roughton, A. and Collings, N., Length Scale and Turbulence Intensity Measurements in a Motored Internal Combustion Engine, *SAE Paper No. 880380* (1988).
- (5) Glover, A. R., Hundleby, G. E. and Hadded, O., An Investigation into Turbulence in Engines Using Scanning LDA, *SAE Paper No. 880378* (1988).
- (6) Reuss, D. L., Adrian, R. J., Landreth, C. C., French, D. T. and Fansler, T. D., Instantaneous Planar Measurements of Velocity and Large-Scale Vorticity and Strain Rate in an Engine Using Particle-Image Velocimetry, *SAE Paper No. 890616* (1989).
- (7) Fraser, R. A. and Bracco, F. V., Cycle-Resolved LDV Integral Length Scale Measurements Investigating Clearance Height Scaling, Isotropy, and Homogeneity in an I. C. Engine, *SAE Paper No. 890615* (1989).
- (8) Catania, A. E. and Mittica, A., Cycle-by-Cycle, Correlation and Spectral Analysis of I. C. Engine Turbulence, in: *Flows in Internal Combustion Engines-III*, FED-Vol. 28, T. Uzkan, W. G. Tiederman, and J. M. Novak, eds., ASME, New York (1985).
- (9) Witze, P. O., Measurements of the Spatial Distribution and Engine Speed Dependence of Turbulent Air Motion in an I. C. Engine, *Trans. SAE, Paper No. 770220* (1977).
- (10) Liou, T.-M. and Santavicca, D.A., Cycle Resolved LDV Measurements in a Motored IC Engine, *Trans. ASME, J. Fluids Eng.*, Vol. 107 (1985), p. 232.
- (11) Catania, A. E. and Mittica, A., A Contribution to the Definition and Measurement of Turbulence in a Reciprocating I. C. Engine, *ASME Diesel & Gas Engines Symposium, ETCE, Paper No. 85-DGP-12* (1985).
- (12) Catania, A. E. and Mittica, A., Induction System Effects on Small-Scale Turbulence in a High-Speed Diesel Engine, *Trans. ASME, J. Eng. Gas Turbines Power*, Vol. 109 (1987), p. 491.
- (13) Aldama, A. A., Filtering Techniques for Turbulent Flow Simulation, *Lecture Notes in Engineering*, Brebbia, C. A. and Orszag, S. A., eds., (Springer Verlag, 1990).
- (14) Bendat, J. S. and Piersol, A. G., *Random Data: Analysis and Measurement Procedures*, 2nd ed., (John Wiley & Sons, 1986).
- (15) Hinze, J. O., *Turbulence*, (McGraw-Hill, 1975).
- (16) Tritton, D. J., *Physical Fluid Dynamics*, (Van Nostrand-Reinhold, 1977).
- (17) Catania, A. E., Induction System Effects on the Fluid-Dynamics of a D. I. Automotive Diesel Engine, *ASME Diesel & Gas Engines Symposium, ETCE, Paper No. 85-DGP-11* (1985).
- (18) Witze, P. O., A Critical Comparison of Hot-Wire Anemometry and Laser Doppler Velocimetry for I. C. Engine Applications, *Trans. SAE, Paper No. 800132* (1980).
- (19) Rader, C. M., An Improved Algorithm for High Speed Autocorrelation with Applications to Spectral Estimation, *IEEE Trans. Audio Electroacoustics*, Vol. AU-18, No. 4 (1970) p. 439.
- (20) Rabiner, L. R., Schafer, R. W. and Dlugos, D., Correlation Method for Power Spectrum Estimation, *IEEE Programs for Digital Signal Processing* (1979).
- (21) Lesieur, M., *Turbulence in Fluids*, (Martinus Nijhoff Publishers, 1987).