Further Investigation into the Statistical Properties of Reciprocating Engine Turbulence

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ABSTRACT

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 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.

INTRODUCTION

Many experimental investigations on engine turbulence have been performed using different measurement and data reduction methods and an extensive bibliography was reported by Catania and Mittica (1989, 1990). 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 (Ikegami et al., 1987; Dinsdale et al., 1988; Glover et al., 1988; Reuss et al., 1989). Simultaneous two-point measurements of in-cylinder velocity at various distances were taken by Fraser and Bracco (1989) for direct measurement of turbulence length scales.

However further research is needed, especially in methodological aspects of engine turbulence analysis, to gain a 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 have not generally 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 IC engines was recently developed (Catania and Mittica, 1990). 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 instant of time 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 (1985b, 1989), and was proved to be suitable for studying the average statistical properties of segmented nonstationary turbulence sample records. It was compared to the one (Witze, 1977; Liou and Santavicca, 1985) based on the rough application of the standard approach for stationary flows using the conventional definition of the nonstationary autocorrelation coefficient, which led generally 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 wall on it were also investigated (Catania and Mittica, 1990).

A problem that is still open concerns the more appropriate definition and estimation of another characteristic time scale of engine turbulence, the macro 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 and 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 filtering the in-cylinder velocity, was provided with reference to the cycle-resolved data reduction procedure previously developed (Catania and Mittica, 1985a).

TURBULENCE QUANTITIES

Turbulent velocity fluctuations

The instantaneous velocity in the cylinder of a reciprocating IC engine is generally split into a mean nonstationary component and a turbulent fluctuation about this, so that, with reference to the engine cycle i

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

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

In the conventional statistical approach, the mean velocity, that is the ensemble-averaged velocity, is the same in each cycle: $\tilde{U}_i(t) = U_E(t)$; the turbulent fluctuation will be called instantaneous velocity fluctuation too: $u_i(t) = u_{i,E}(t)$.

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 (Catania and Mittica, 1985a, 1987). It can be shortly named time-average filtering (TAF), and yields 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 (Catania and Mittica, 1989). 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.

Using the symbol $\overline{f}_i$ 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))

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

it is, within the TAF procedure

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

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 (Catania and Mittica, 1987).

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.

Using the symbol $<f_i(t)>$ to represent the ensemble average of the variable $f_i(t)$ at time t over N records (or cycles)

$$<f_i(t)> = \frac{1}{N}\sum_{i=1}^{N} f_i(t) \quad (5)$$

it is

$$U(t) = <\tilde{U}_i(t)> = <I[\overline{U}_i]> = I[<\overline{U}_i>] \quad (6)$$

Probability density function

For a statistical analysis of nonstationary turbulence properties evaluated by short time-averaging, so as to have negligible distorsions 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

$$p(u,T,t_0) = \frac{<T_i>}{T\,du} \quad (7)$$

where T_i is the total portion of T spent by $u_i(t)$ in the range of values [u, u + du]. It can be easily verified that equation (7) gives a first moment of the cycle-resolved turbulent fluctuation in T equal to $<\overline{u}_i>$, that is zero within the TAF procedure, and a second moment equal to the square of the turbulence intensity in T: $u' = \sqrt{<\overline{u_i^2}>}$.

Cross terms

By ensemble-averaging and short time-averaging operations performed on the instantaneous momentum equation, subsequent to the introduction of the velocity decomposition (1), non-Reynolds stresses or cross terms arise from the products of the turbulent fluctuation and the mean velocity. An insight into the normal Reynolds-like stresses (per unit density) $u'^2 = <\overline{u_i^2}>$ and $U_{rms}^2 = <\overline{\tilde{u}_i^2}>$ was given by Catania and Mittica (1987, 1989). Here an evaluation of the normal cross term $<\overline{u_i\tilde{U}_i}>$ 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 (Aldama, 1990) 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

$$\overline{u_i\tilde{U}_i} \approx \tilde{U}_i(t_0)\,\overline{u}_i = 0 \quad (8)$$

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

$$<\overline{u_i\tilde{U}_i}> \approx \left.\frac{dU}{dt}\right|_{t_0}\overline{(t-t_0)\Sigma(t)} + O\left[\left.\frac{d^2U}{dt^2}\right|_{t_0}\overline{(t-t_0)^2\Sigma(t)}\right] \quad (9)$$

where $\Sigma(t) = <u_i(t)>$, and the symbol $O[\;]$ indicates the order of magnitude. $\Sigma(t)$ should be virtually negligible, depending on the number of sample data records (Catania and Mittica, 1989) 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.

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 (Catania and Mittica, 1990)

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

where the separation-time variable τ in T ranges from zero to τ_{max} = T/2. The factor u'^2, normalizing the autocorrelation function to unity at τ = 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 equation (10) is an unbiased and consistent estimate of the autocorrelation coefficient $R(\tau)$ independent of T and of t_0 (Catania and Mittica, 1990). For a nonstationary random process, $R(T, t_0, \tau)$ should be averaged with respect to t_0 and T in order to be reduced to only a function of τ. In this way, information about the time evolution of the process is lost. To analyze the structure of turbulence throughout 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) assumes a quasi-steadiness of the turbulent fluctuation, namely the statistical character of the fluctuation is considered, on average, as homogeneous with respect to time in the period 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, similarly to equation (10), so as to give an even function of τ

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

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 equation (10); in particular, t_0 in equation (11) indicates any instant of time, while in equation (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 equation (10) as follows

$$E(T,t_0,f)=$$

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

where f is the frequency, $w(\tau)$ is a window function of 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 (Bendat and Piersol, 1986).

<u>Eulerian time scales</u>

An estimate of the Eulerian dissipation, or micro, time-scale of turbulence in the interval T is given by

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

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 to the sample data). This parameter is a measure of the most rapid changes that occur in the fluctuations of $u_i(t)$ on average in 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[1] over τ from 0 to τ_{max}, when R has positive values and decays over a sufficiently long period

$$\Lambda_\tau(T,t_0)=\int_0^{\tau_{max}} R(T,t_0,\tau)w(\tau)d\tau \qquad (14)$$

From equation (12), $\Lambda_\tau(T,t_0)=E(T,t_0,0)/4$.

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 (Tritton, 1977)

$$\Delta_\tau(T,t_0)=\tau\,|_{R(T,t_0,\tau)_{min}} \qquad (15)$$

IN-CYLINDER VELOCITY MEASUREMENTS

<u>Experimental system and procedure</u>

The velocity data, for reduction and analysis, were acquired in the direct-injection automotive diesel engine previously described (Catania, 1985). 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 induction system was made up of two equiverse swirl tangential ducts, and valves of the same size and lifts. The measurements were taken under motored conditions for the two configurations of the intake system (Fig. 1): with only one duct operating (intake I) and with both ducts operating (intake I and II), at the engine speeds of 167.6 rad/s (1600 rpm) and 314.2 rad/s (3000 rpm). One measurement location was considered along the radial direction on the right hand side in Fig. 1, at a level of ≈ 3.5 mm underneath the flat cylinder head. It was at radius r = 18 mm (from the cylinder axis), that is approximately in the middle of the right hand section of the shallow combustion bowl at TDC.

The measurements were carried out in the crank-angle interval θ = 0-360 deg, that is from the start of the induction stroke to the end of compression.

A number of from 60 to 100 sample data records were acquired using an advanced HWA technique (Catania, 1985), with the sensing wire parallel to the cylinder axis, at a rate of one data per 0.2 crank-angle degree, which is high enough to give a virtually continuous velocity trace in every cycle.

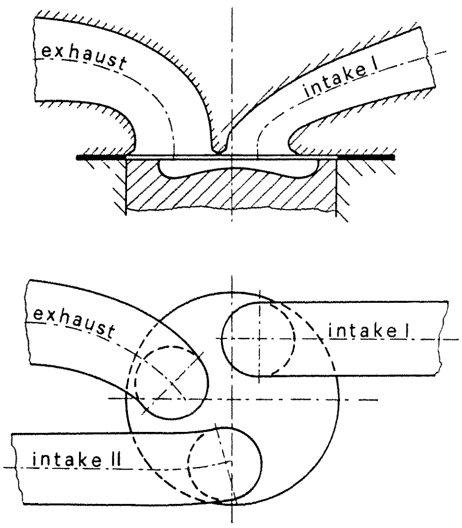

Fig. 1 - Schematic of the intake system.

<u>Experimental uncertainties</u>

The method used to compute gas velocity from the anemometer output is different from all of those semiempirical corrections considered by Witze (1980) (Catania and Mittica, 1987). It was validated in nonstationary as well as stationary conditions, at different gas pressures and temperatures and for different wire orientations.

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.

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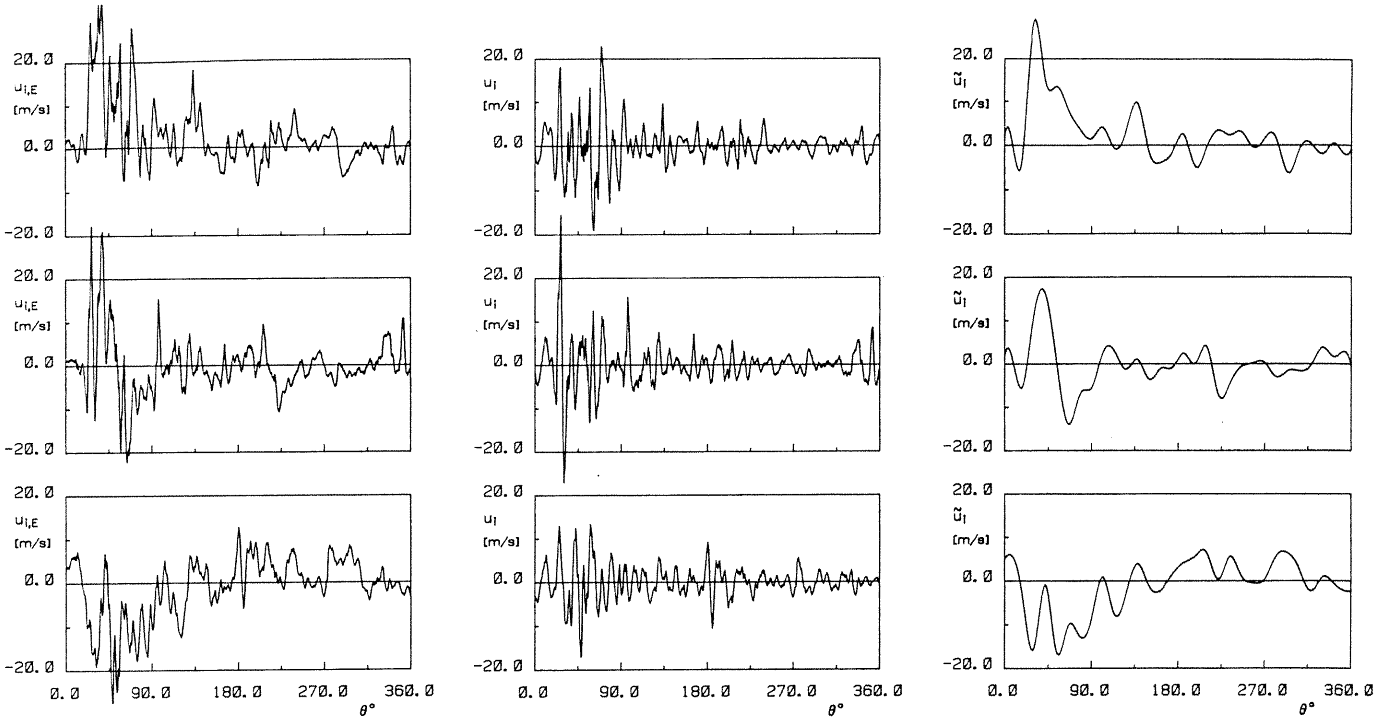
1 The symbol R indicates the autocorrelation coefficient in general.



a - Instantaneous velocity fluctuations

b - Cycle-resolved turbulent fluctuations

c - Mean-velocity cyclic fluctuations

Fig. 2 - Velocity fluctuations in individual cycles.

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.

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 (Catania and Mittica, 1985a, 1987).

It is evident from Fig. 2(a) that the time average of the instantaneous velocity fluctuations 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 cab be related to the lower frequency cyclic variations in the mean flow by comparing Figs. 2(a) and 2(c).

A modified version of the algorithm developed by Rader (1970) and by Rabiner et al. (1979) 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 compares the two definitions of autocorrelation coefficient, given by equations (10) and (11), by their application to the cycle-resolved turbulent fluctuation u_i and to the instantaneous velocity fluctuation $u_{i,E}$ derived from the same sample data records, and using the results in equation (12) for autospectra estimation. The crank angle θ_0 (corresponding to time t_0), the crank-angle correlation interval Φ (corresponding to the period T), together with the intake system configuration and the engine speed ω, are indicated in the figure. A continuously decaying pattern (Fig. 3(a)) is shown by $R[u_{i,E}]$ versus time separation (corresponding to the crank angle $\theta - \theta_0$) as would be expected for unfiltered velocity data records.

The normalized energy spectral density functions corresponding to the autocorrelation coefficients of Fig. 3(a) are reported in ordinary (Fig. 3(b)) and logarithmic scales (Fig. 3(c)) 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 symbols) are nonnegative for all values of f, as can be easily demonstrated. However, meaningless negative and fluctuating values may be present in E_E (Catania and Mittica, 1990), as evidenced by $E_E[u_i]$ (Fig. 3(b)). Furthermore, this spectrum indicates the presence of a significant low frequency component in the cycle-resolved turbulent fluctuation, whereas low frequencies should have been removed by filtering, as shown by $E[u_i]$. This may be ascribed to the different time dependence expressed by equation (10) with respect to equation (11) 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 Fig. 3(c) that the tail of the spectra $E[u_i]$ and $E[u_{i,E}]$ shows virtually the same decaying law with frequency, which means that the dissipative components of cycle-resolved turbulence and instantaneous velocity fluctuation have similar energy distributions. This was confirmed by other results not reported here. For analogy with wave number energy spectra, the slopes -5/3, -3 and -4 are marked in the figure. It is worth pointing out (Lesieur, 1987) 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 when the Reynolds number is sufficiently high; on the other hand a Kraichnan k^{-3} and a Saffman k^{-4} wave number spectra can be predicted within the enstrophy cascade concept which applies to environments with high local vorticity gradients

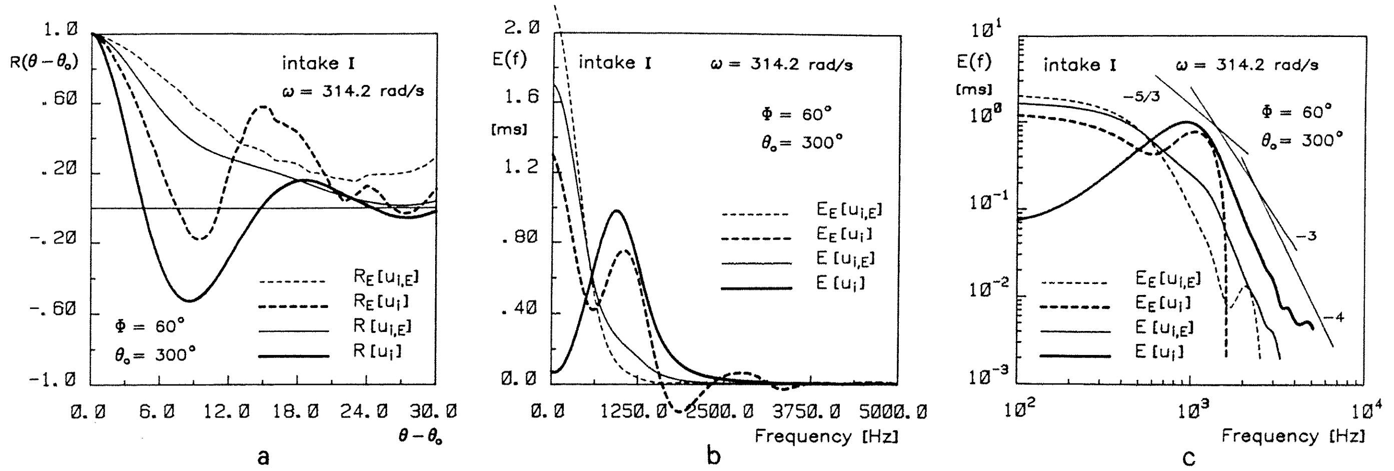


Fig.3 - Autocorrelation (a) and autospectral density (b,c) coefficients

and so a strong eddy interaction, implying a high flux of vorticity into the small scales.

Figure 4 illustrates the dependence of the autocorrelation coefficient and time scales on the correlation interval length for both cycle-resolved (a) and conventional (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 a 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 relevant 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 Φ = 60 deg. It can be inferred from the figure that the micro time scale is virtually independent of Φ for both u_i and $u_{i,E}$, as the obsculating parabola in the vertex of the correlation curves is (Hinze, 1975). This confirms both the choice of Φ = 60 deg for correlation interval and the validity of the results on the engine turbulence structure previously reported using Φ = 30 deg (Catania and Mittica, 1987, 1990).

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 examine the influence of the induction system partialization (that is the closure of one intake duct) and of the engine speed on the structure of the cycle-resolved turbulence (a) and the fluctuating motion in its more conventional statistical sense (b), including the mean-flow cyclic variation.

Figures 6 and 7 show the autocorrelation coefficients and their respective Fourier cosine transforms at the end of compression. It can be inferred from Figs. 6(a) and 7(a) that the cycle-resolved turbulence structure 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 evident in Figs. 6(b) and 7(b).

Similar results were obtained throughout the induction and compression strokes, as evident 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 Δ_τ at the higher 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

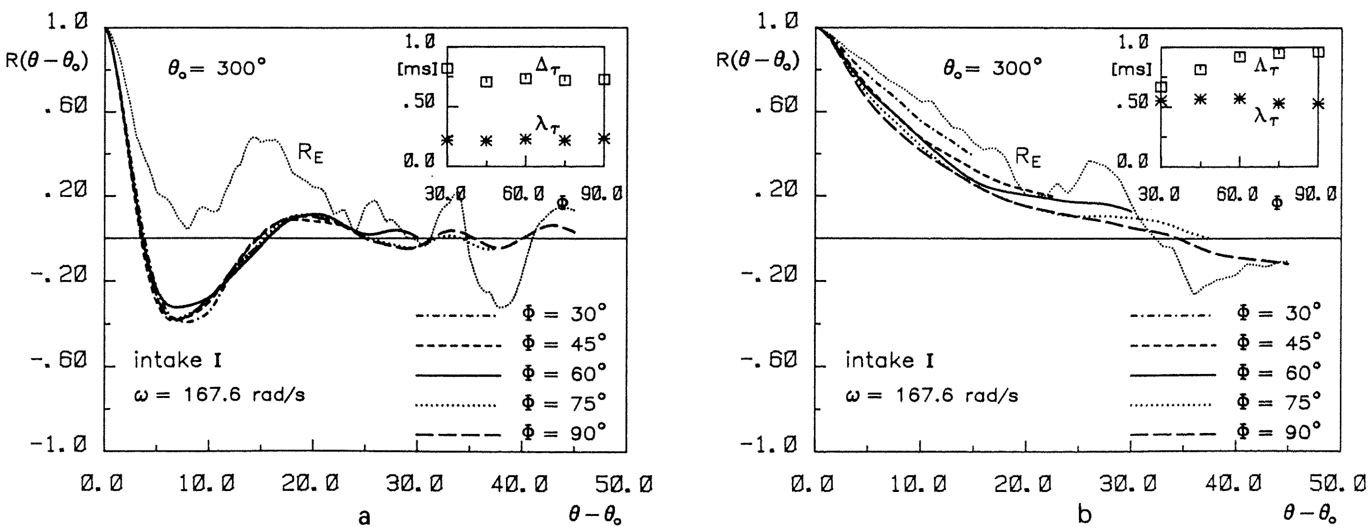


Fig. 4 - Autocorrelation coefficients of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

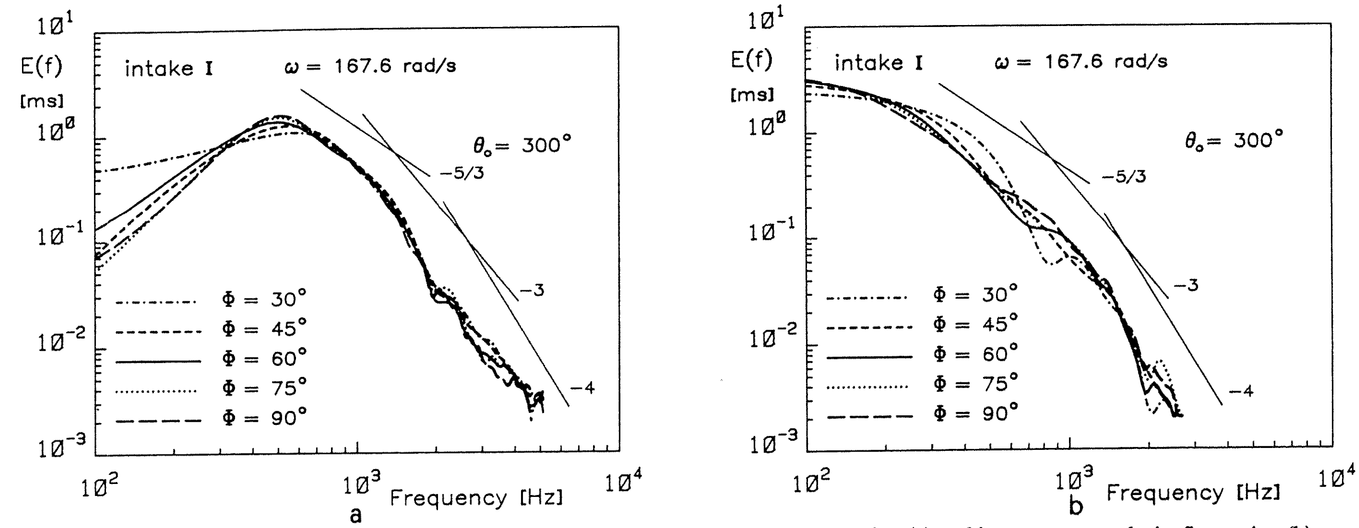


Fig. 5 - Normalized energy spectral density function of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

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 on 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 is plotted with dashed line. This is approached closely and gradually from the induction to the end of compression, with negligible distorsions due to the unsteadiness of the flow.

Fig. 11 presents, for the engine conditions considered, the pattern of $I[<u_i \overline{U}_i>]$ versus crank angle (for simplicity, the

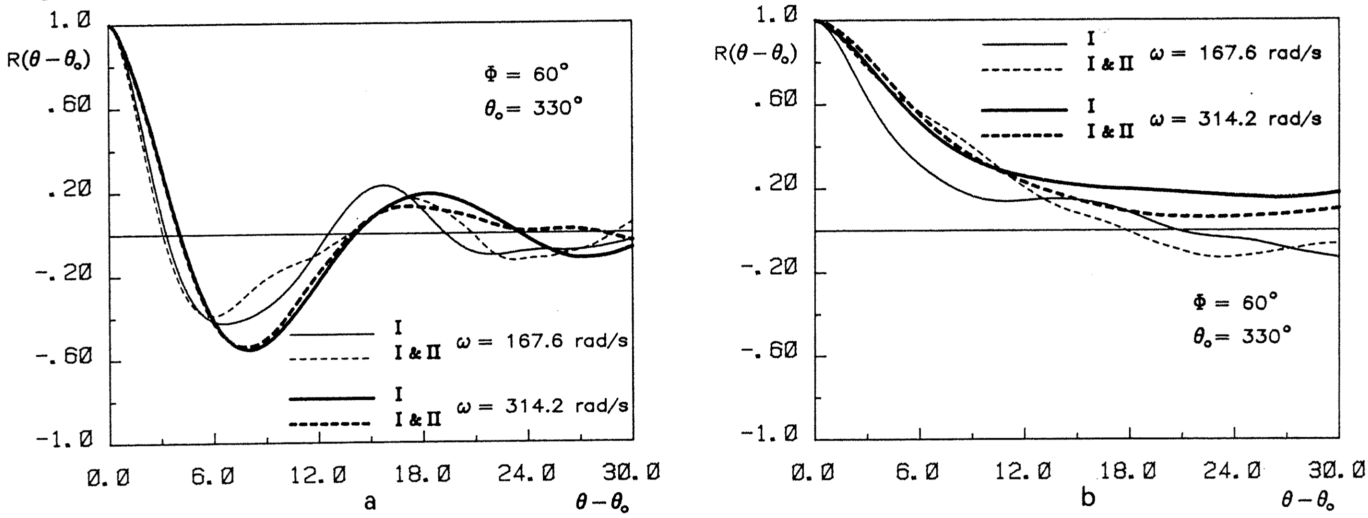


Fig. 6 - Autocorrelation coefficients of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

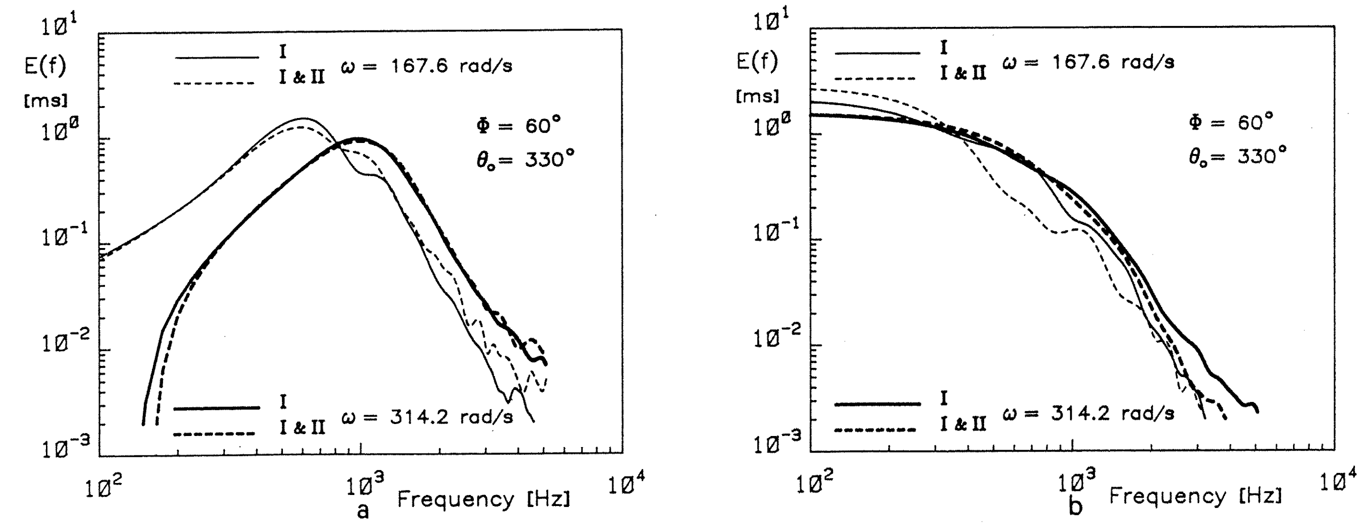


Fig. 7 - Normalized energy spectral density functions of cycle-resolved turbulent fluctuation (a) and instantaneous velocity fluctuation (b)

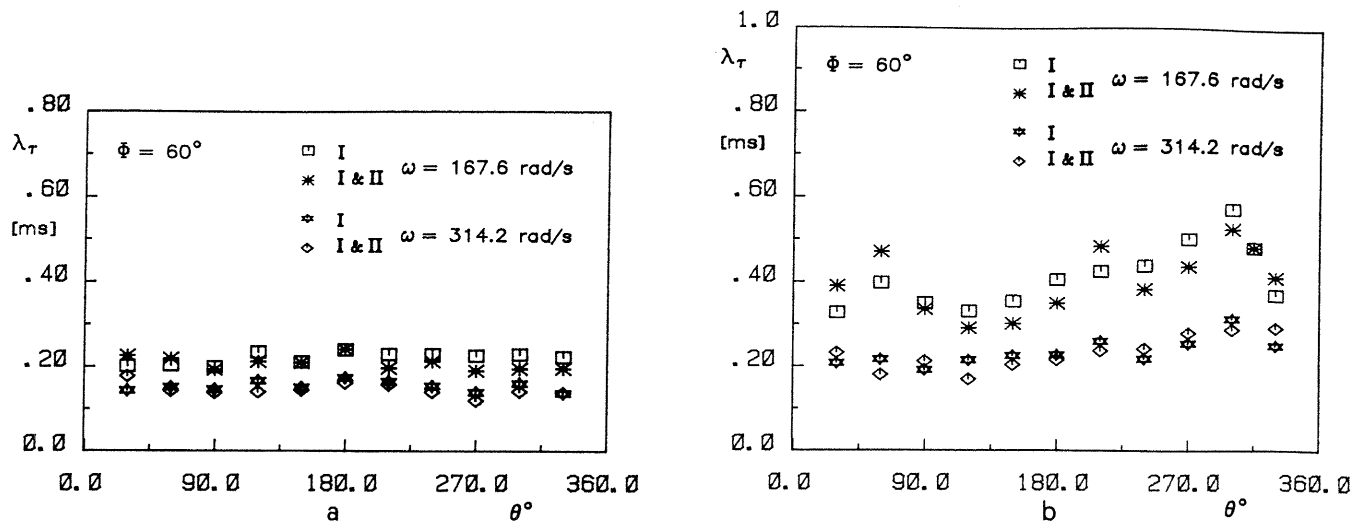


Fig. 8 - Micro time-scale distributions of cycle-resolved turbulence (a) and instantaneous velocity fluctuation (b)

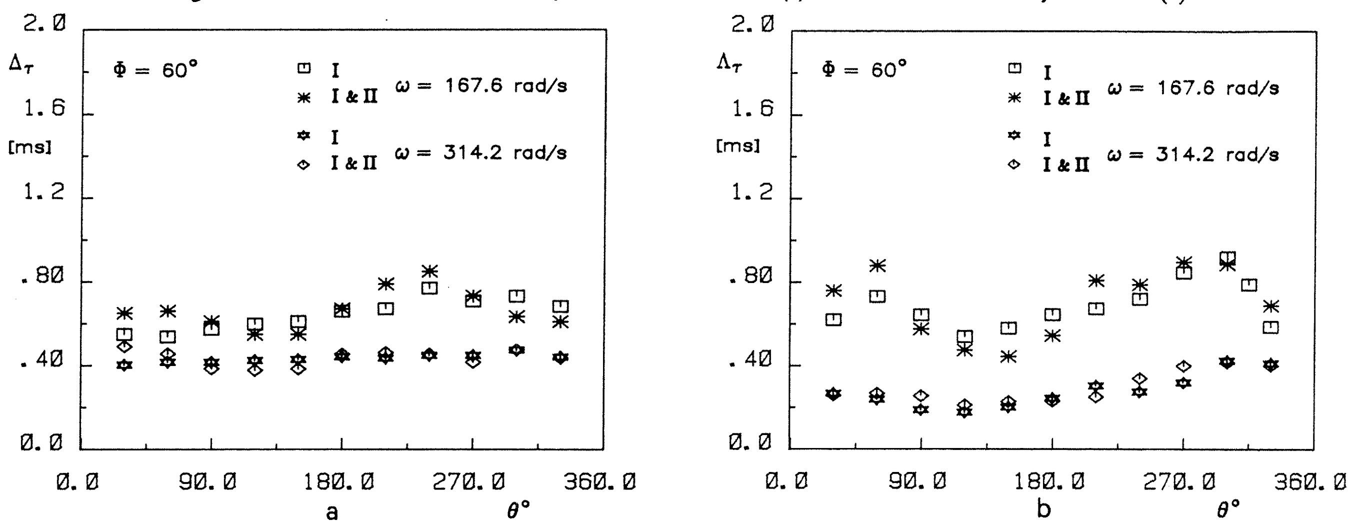


Fig. 9 - Macro time-scale distributions of cycle-resolved turbulence (a) and instantaneous velocity fluctuation (b)

notation $u_i\tilde{U}_i$ is indicated in 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 (Catania, 1985). 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 (Catania and Mittica, 1987).

CONCLUSIONS

The general method previously developed for autocorrelation and autospectra estimation of nonstationary turbulence, giving 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 stationary-flow integral time-scale analog, for the

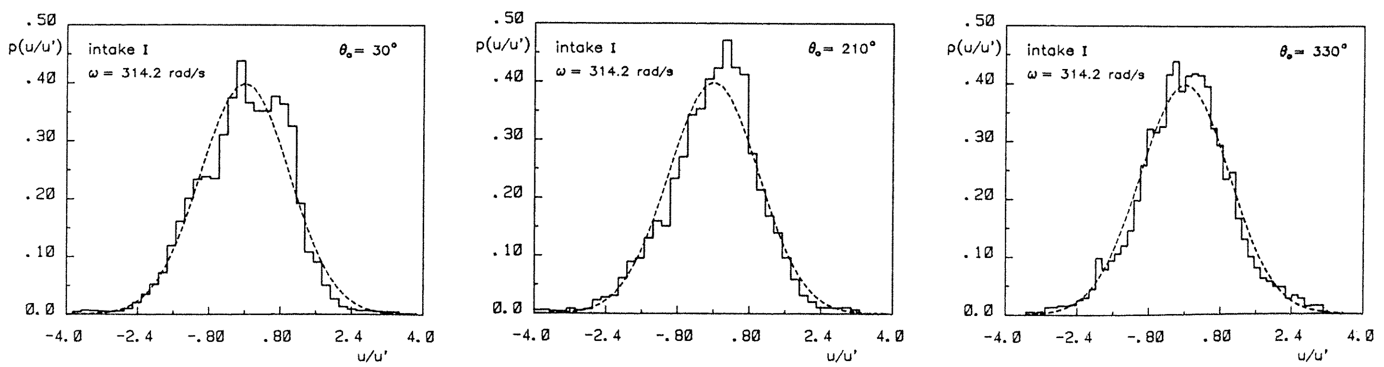


Fig. 10 - Normalized probability density function of cycle-resolved turbulence

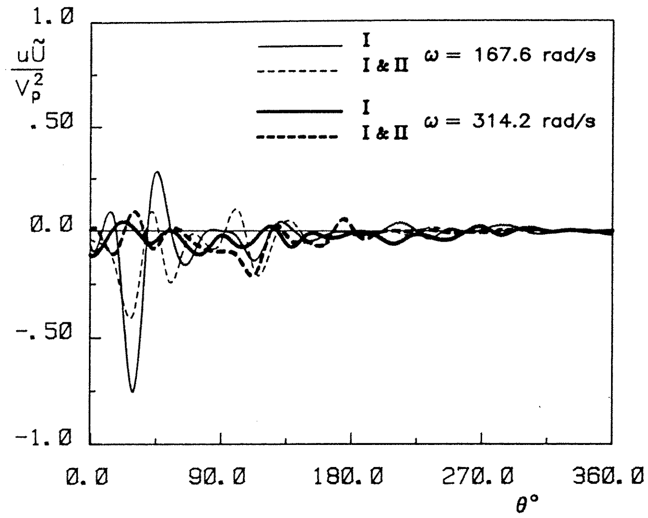


Fig. 11 - Cross term distributions

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 showed generally 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 displayed a negative region, implying that the fluctuation tend to be in opposite directions more 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 partialization of a two-duct intake system 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 turbulence was shown to be more sensitive to the engine speed than to the intake system configuration.

The normalyzed probability density function, for statistical analysis of short time-averaged turbulence properties, was very close to the standardized Gaussian distribution, with negligible distorsion due to the unsteadiness of the flow. This confirmed 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 analysis, is expected to yield negligible cross terms which arise in the momentum flow equation when non-Reynolds averaging procedures are considered.

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