

Turbulence Models Validation by LDA in an Internal Combustion Engine

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ABSTRACT

In the present paper both cycle resolved and ensemble averaged LDA measurements were carried out in the combustion chamber of a real D.I. diesel engine with a compression ratio of 18:1 running at 1000 rpm.

Indirect measurements of turbulence length scales were performed during the compression stroke. Data reduction technique based on FFT filtering procedure is discussed. The experimental findings were employed to validate four different variants of the $k-\epsilon$ turbulence model with compressibility terms installed in 3D KIVA code.

It was found a reasonable agreement between computed mean swirl velocity and measured one while all turbulence models slightly underestimate turbulence intensity at TDC. Measured longitudinal length scales are of same order of magnitude of turbulent dissipation length scale obtained by different variants of $k-\epsilon$ model with the exception of Watkins model.

INTRODUCTION

The most used turbulence models for I.C. engine flow are the $k-\epsilon$, in various formulations, and the Large Eddy Simulation one (LES) developed by Los Alamos school and implemented in the first version of the Kiva code [1]. The $k-\epsilon$ model must be properly derived for strongly compressible engine flows, that are typical of i.c. engine applications. The principal formulations of this model are due to Watkins [2], Reynolds [3], Morel and Mansour [4] and finally El Thary [5]. In a recent work Ahmady-Brefui and Gosman gave an assesment of the different variants of $k-\epsilon$ model for strongly compressible flows [6].

They compared the behaviour of the different models with the experimental measurements of mean velocity and turbulence intensity in the axisymmetric disc-chamber model engine operating at 200

rpm and compression ratios of 3.5 and 6.7:1 respectively. Even if many experimental data of in cylinder flow are available both with LDA and HWA techniques the majority, as in the case of ref. [6], are referred to low engine speed and/or low compression ratio value. In addition the comparison between measurements and computations in literature are always referred to the ensemble averaged measurements of mean velocity and turbulence intensities inside principally premixed charge engines and model engines [13-16].

Recently A.O. Zur Loye, D.L. Siebers and others [7] showed a comparison between cycle resolved measurements and 2D computations performed by Kiva code, during the compression stroke of a quiescent chamber diesel engine operating at 300 and 600 rpm with a compression ratio of 10:1. They justified the need of cycle resolved measurements because this kind of combustion system produces in cylinder flow with low turbulence intensity and large cycle variations.

However in previous experiences [10] we found that, also in presence of moderate rate of swirl and squish motions, the cyclic variation can reach values close to the ensemble averaged turbulence intensities. Because the cyclic variation doesn't influence turbulent mixing the choice of comparing the cycle resolved measurements and calculations may be appropriate.

Thus, in the present work measurements of mean velocity and turbulence intensity both ensemble averaged and cycle resolved were performed in a real single cylinder four stroke diesel engine.

The cycle resolved techniques allowed also an estimate of turbulent length scale. The different variants of $k-\epsilon$ model, installed in the Kiva code, were tested, during the compression stroke, starting from initial conditions experimentally fitted at 90 CA before TDC compression. The comparison between experimental and numerical results were performed on the basis of mean swirl velocity, turbulence intensity and macro-length scale calculations along a diameter posed at a

distance of 4. mm from the cylinder head. In the following an analysis of behaviour obtained from different turbulence models adoption is also given.

EXPERIMENTAL SET-UP

The measurements were performed on a single-cylinder Ruggerini four-stroke, direct injection Diesel engine of 100 mm bore, 95 mm stroke with optical access, fig.1. A shrouded intake valve allowed to run under variable swirl conditions.

To reduce the problem of optical window fouling, two large lubricating bronze-impregnated teflon piston rings were used allowing the engine to run without cylinder lubrication. LDV data were acquired under motoring conditions at engine speed of 1000 RPM in the horizontal plane of the cylinder at the depth of 4. mm from the engine head. A toroidal combustion chamber, fig.1, with an Equivalent Aspect Ratio (EAR) of 3.6 and a squish area of 72% was tested.

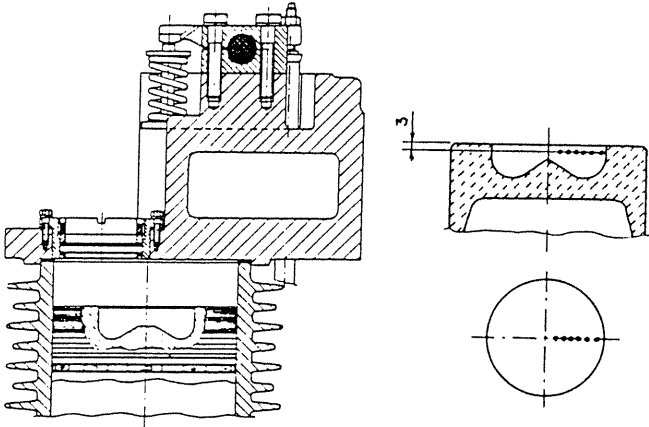


Fig.1 Engine with optical access and toroidal combustion chamber tested.

The instantaneous tangential component was measured during compression stroke using a crank angle window of .6 degree. The cycle that had a number of validated measurements lower 150 was refused according to the cycle-resolved analysis rule that requires valid measurements at least every 1-2 degrees [8].

The LDV data rate at 1000 rpm was about 1.4 KHz with a validated percentage about 60 to 70%. Finally the consecutive cycles number used for the LDA processing were about 20 for schematic configuration of LDA system described in more details in a previous paper [9].

DATA REDUCTION ANALYSIS

In stationary turbulent flows is valid the Reynold's decomposition splitting the instantaneous velocity into mean and fluctuating components:

$$U(t) = \bar{U} + u(t)$$

In cylinder flow field the fluid motion is time dependent and so the mean velocity is a time function. An ensemble averaging procedure is commonly used to define the mean velocity, denoted $U_c(\theta)$, as a crank angle function:

$$U_c(\theta) = \frac{1}{N} \sum_{J=1}^N U(\theta, J)$$

where $U(\theta, J)$ is the instantaneous velocity at θ crank angle and J -cycle; N is the total number of cycles.

Because the fluctuations are random, their ensemble mean, $\langle u(\theta) \rangle$, vanishes, so:

$$\langle u(\theta) \rangle = 0$$

A measure of the intensity of the velocity fluctuations is provided by the rms about ensemble mean velocity:

$$u'(\theta) = \left\{ \frac{1}{N} \sum_{J=1}^N [U(\theta, J) - U_c(\theta)]^2 \right\}^{1/2}$$

This definition of turbulence intensity includes all fluctuations from the ensemble average so to overcome this problem a cycle-resolved-velocity analysis is necessary to estimate the individual cycle-mean velocity $U_f(\theta, J)$.

We adopte a low-pass digital filtering to estimate the in-cycle mean velocity $U_f(\theta, J)$ by means a square moving-window average in the time domain.

The moving-window average is a simple technique where: $U_f(\theta, J)$ is estimated by taking a weighted sum over $N = 2L+1$ input values $U(\theta, J)$ obtained at equally spaced time simmetrically disposed about θ . So the filtered velocity $U_f(\theta, J)$ is given by:

$$U_f(\theta, J) = \sum_{k=-L}^L U(\theta - k, J) \cdot w(k)$$

where $w(k)$ is an appropriate weight function which lenght determines the effective cut-off frequency, F_c .

Theoretically it should be choice from characteristics scales of physical process but in practice there is arbitrariness in the choice of the exact cut-off frequency.

There we select our choice of F_c at the end of the first sharp decay of the ensemble power spectral density using a FFT procedure chosing a square window function.

The result of filtering produces a $U_f(\theta, J)$ and a high-frequency fluctuation component relative to filter cut-off frequency:

$$uhf(\theta, J) = U(\theta, J) - U_f(\theta, J)$$

while the intensities of the high-frequency fluctuations $u'hf(\theta, J)$ are characterized by their rms about $U_f(\theta, J)$:

$$u'hf(\theta, J) = \left\{ [U(\theta, J) - U_f(\theta, J)]^2 \right\}^{1/2}$$

An ensemble high frequency fluctuations, $u'_{ehf}(\theta)$ can be defined as:

$$u'_{ehf}(\theta) = \left\{ \frac{1}{N} \sum_{J=1}^N [Uf(\theta, J) - U_{ef}(\theta)]^2 \right\}^{1/2}$$

where $U_{ef}(\theta)$ is the ensemble average of the filtered velocity:

$$U_{ef}(\theta) = \frac{1}{N} \sum_{J=1}^N Uf(\theta, J)$$

The filtering produces a digital function $uhf(\theta, J)$ time-invariant so we can define time autocorrelation function, $R(t, J)$, defined in terms of the fluctuation about the mean:

$$R(t, J) = \frac{\langle uhf(\theta, J) \cdot uhf(\theta + t, J) \rangle}{u'_{hf}(\theta, J) \cdot u'_{hf}(\theta + t, J)}$$

and an ensemble autocorrelation function $R(t)$:

$$R(t) = \frac{1}{N} \sum_{J=1}^N R(t, J)$$

it can be shown [11] that the micro time scale of turbulence, λ , is given by:

$$\frac{1}{\lambda^2} = -\frac{1}{2} \left[\frac{\partial^2 R(t)}{\partial t^2} \right]_{t=0}$$

Here the Eulerian time scale is a measure of the most rapid change that occurs in the fluctuations $uhf(\theta, J)$.

Other characteristic time scale, L_t , can be estimate taking the integral of $R(t)$ from 0 to t_{max} when $R(t)$ has positive values or decay over a long period; if $R(t)$ obtains negative values, the delay time over which the fluctuations remain correlated is that at which $R(t)$ has a minimum [12]. If Taylor's hypothesis is true then we can estimate the longitudinal integral length scale, L_g , from the characteristics decay time of $R(t)$ and the ensemble-mean filter tangential velocity:

$$L_g(\theta) = L_t \cdot U_{ef}(\theta)$$

Finally we compute the $u'_{elf}(\theta)$ terms from the ensemble average of the filtered velocity:

$$u'_{elf}(\theta) = \left\{ \frac{1}{N} \sum_{J=1}^N [Uf(\theta, J) - U_{ef}(\theta)]^2 \right\}^{1/2}$$

We relate low frequency ensemble rms fluctuation intensity to the cyclic variation although filtering cannot rigorously separate the instantaneous velocity into mean velocity, cyclic variation and turbulent fluctuations.

SELECTED LDA RESULTS

In the figs.2, 3 and 4 the evolution of mean velocity and turbulence intensities during compression stroke are shown. The measurements were taken in the locations of 11,15 and 19 mm from cylinder axis along a

diameter of the combustion chamber (points C,B,A respectively). In the top part of the diagrams the ensemble averaged velocities and the filtered ones are compared while in the bottom part ensemble averaged rms and cycle resolved turbulence intensities are reported. It's easy to note that the cyclic variation can assume values comparable with the turbulence intensity. However, as well known, these results are strongly influenced by the choice of the cut-off frequency. Moreover in order to formulate and calibrate the turbulence models the characterization of turbulence scales in engine environment is of fundamental importance.

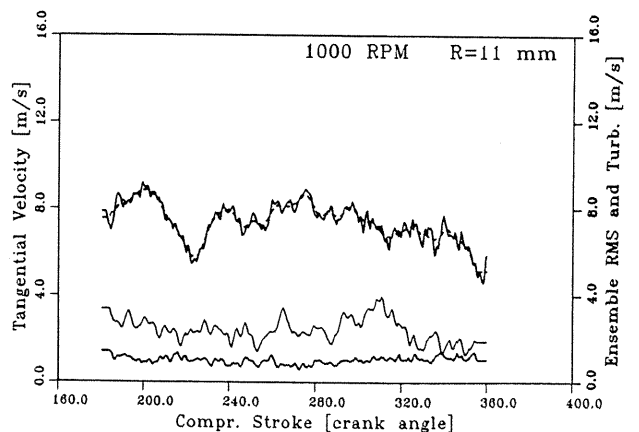
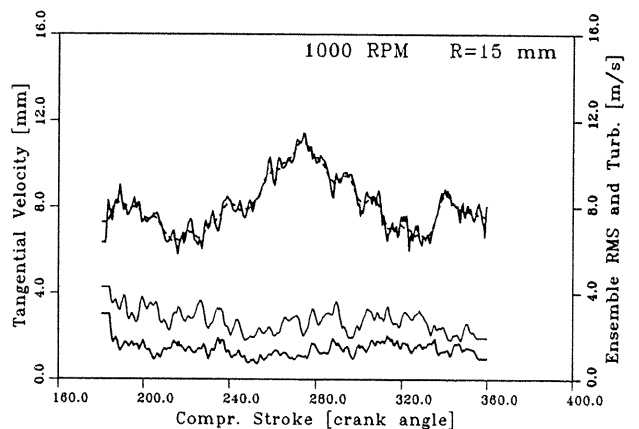
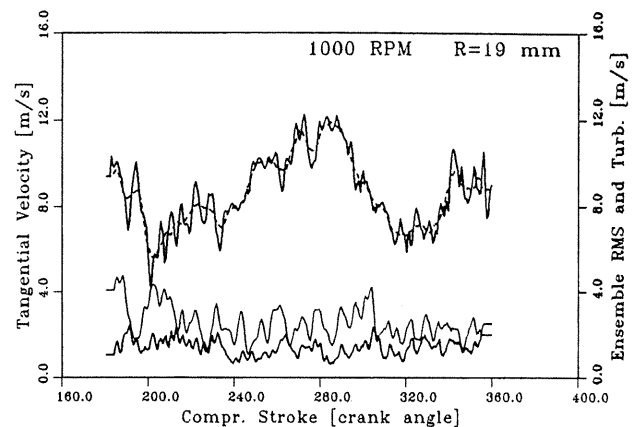


Fig.2 Tangential velocity (top) and Fig.3 turbulence (bottom) profile vs. Fig.4 compression stroke.

In the present paper it will be discussed only the indirect measurements of length scales. The measurements, as previously shown, are based on the Taylor's hypothesis. During the compression stroke the Taylor hypothesis is approximately valid. In fact the flow field is not stationary, not exactly homogeneous even near TDC and the mean velocity often can assume the same value of turbulence intensity. Thus direct measurements of engine length scale will be certainly more incisive. However, despite previous considerations, in the ref. [17] it's shown that the direct and indirect measurements give the same order of magnitude in similar engines.

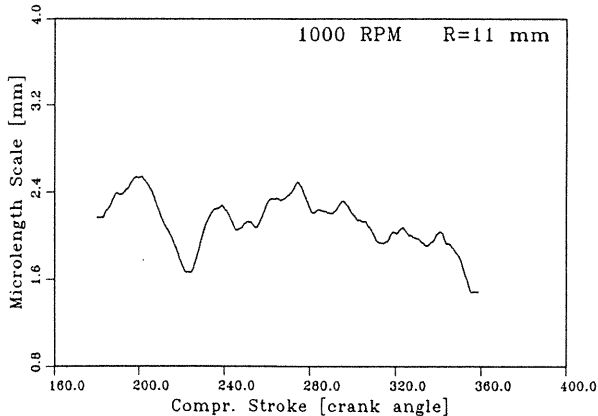
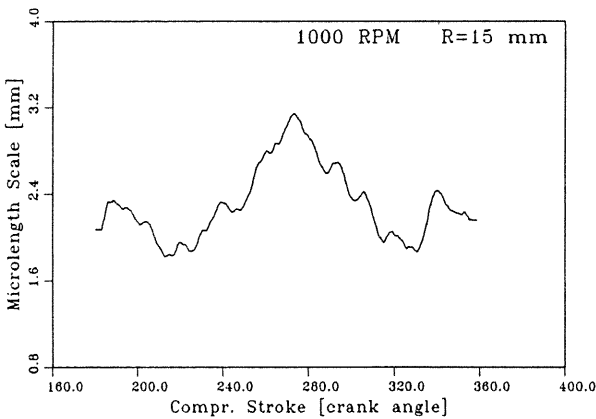
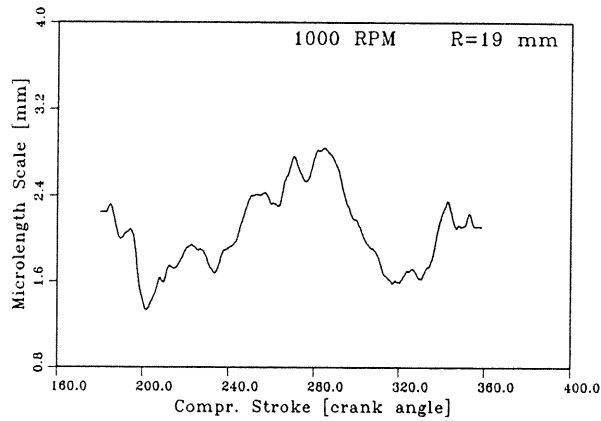


Fig.5 Micro length scale profile vs. compression stroke
Fig.6
Fig.7

The indirect evaluation of length scales from single point measurements, very easy to perform applying the cycle resolved technique, can be proposed for comparative evaluations.

In the figs.5,6 and 7 the evolution of longitudinal micro-length scales in the locations A,B,C is shown. The length scales assume a maximum at half compression stroke and then falls down. When the piston reach the TDC, the length scale rises again. Only in the location near to the chamber axis the pattern is more flat assuming at the end of compression stroke very small values. Few data are reported in literature on indirect measurements of length scales with speed and compression ratio close to our. In the work of Ball, Pettifer and Waterhouse [18] a Ricardo E16 diesel engine with open chamber combustion system was used with 120.6 mm bore and compression ratio of 16.3:1. Even if the combustion chamber was larger than our, the swirl component measurements in certain position in the bowl are very similar to those measured in our experiment at TDC at the same engine speed (about 1000 rpm). The length scales obtained at TDC are reported in the Tab. I, ref. [18].

TAB. I

Speed (rev/sec)	16	24	40
Macroscale (mm)	6.2	9.9	7.4
Microscale (mm)	3.6	5.4	5.3

These values are slightly higher than those measured in the present work but it is reasonable taking in to account the larger bore and valve ports of the Ricardo engine.

NUMERICAL CALCULATIONS

The numerical calculations were performed with the first version of the Kiva code with the k- ϵ turbulence model in different variants [6]. As well known the transport equation for the turbulent kinetic energy is the same for the all models and can be written in cartesian tensor notation as:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho U_j k) = \frac{\partial}{\partial x_j} \left(\frac{\mu_T}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + 2\mu_T S_{ij} S_{ij} - \frac{2}{3}(\mu_T D^2 - \rho k D) - \rho \epsilon \quad (1)$$

where ρ is the density, μ_T is the turbulent viscosity, $S_{ij} = \frac{1}{2} \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right)$, $D = S_{ii}$ is the velocity tensor divergence, σ_k is the turbulence Prandtl number.

The transport equation for ϵ may be represented by:

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_j}(\rho U_j \epsilon) = \frac{\partial}{\partial x_j} \left(\frac{\mu_T}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right) + C_4 \frac{\rho \epsilon}{\mu} \frac{\partial \mu}{\partial t} + \frac{\epsilon}{k} \left[2C_{1\mu} \mu_T S_{ij} S_{ij} - \frac{2}{3}(C_{1\mu} \mu_T D^2 + C_{1\mu}'' \rho k D) \right] + C_{3\rho} \rho \epsilon D - C_2 \frac{\rho \epsilon^2}{k} \quad (2)$$

For a very deep analysis of the different terms of equation (2) see the ref. [6]. In tab. II also derived from [6] the numerical value of the constant for ϵ equation is shown.

TAB. II

Name Author	C ₁	C' ₁	C'' ₁	C ₂	C ₃	C ₄
Turb2 Watkins	1.44	1.44	1.44	1.92	1.	0.
Turb3 Reynold	1.44	1.44	1.44	1.92	.372	0.
Turb4 Mor. Man.	1.44	1.32-1.44	3.5-4.5	1.92	1.	0.
Turb5 El Thary	1.44	1.44	1.44	1.92	-0.33	1.

It can be noted that the constant C₁ and C₂ are function of the strain field for the model Turb4; the model Turb5 due to El Thary includes an additional compressibility term in the form:

$$C_4 \frac{\rho \epsilon}{\mu} \frac{\partial \mu}{\partial t}$$

where μ is the molecular viscosity. This term can be rearranged in the form C₄ $\rho\epsilon\nu$ and the coefficient C₄ can vary in the range -0.15 to -0.25 for the temperature variations typical of engine compression process.

Because our version of the KIVA code is not, at moment, capable of calculating the intake process, the calculations were initiate 90 CA before TDC compression on the basis of experimental measurements obtained by LDA. In particular it was assumed an experimental swirl velocity profile measured on the top of the piston and axial velocity close to the piston speed.

From the measurements of turbulence intensity and of turbulent length macroscale the initial value of k and ϵ was computed. In fig.8 the 3D computational mesh with 19x15x18 nodes is shown at 90 CAD BTDC.

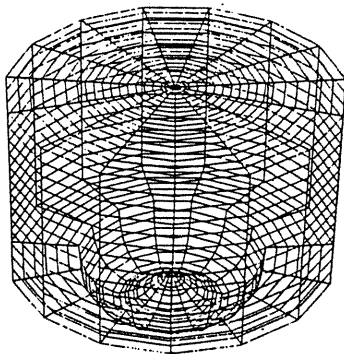


Fig.8 3D computational mesh at 90 cad btcdc.

COMPARISON BETWEEN COMPUTATIONS AND MEASUREMENTS

The following air flow field parameters were selected for the comparison between computations and measurements:

- mean swirl component U
- turbulence intensity $u' = (2/3K)^{1/2}$
- dissipation length scale $l = C\mu^{3/4}k^{3/2}/\epsilon$.

However it can be noted that the dissipation length scale is related but not identical with the energy containing scales and consequently with the integral length scales. Then the computed values can, only, be referenced to the prediction trends and to orders of magnitude.

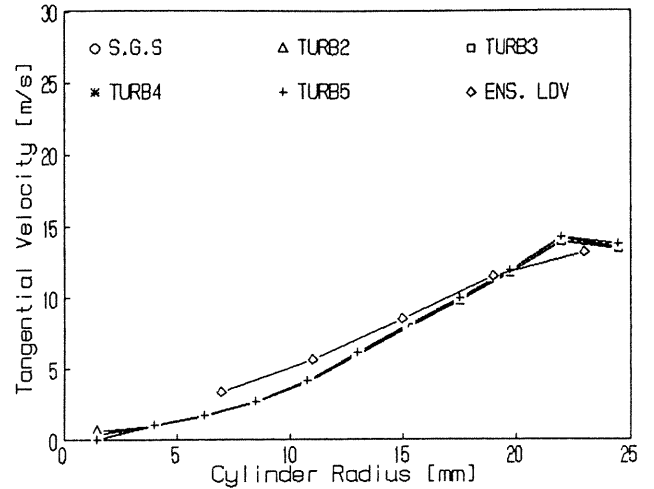


Fig.9 Measured and computed mean tangential velocity at 20 cad btcdc.

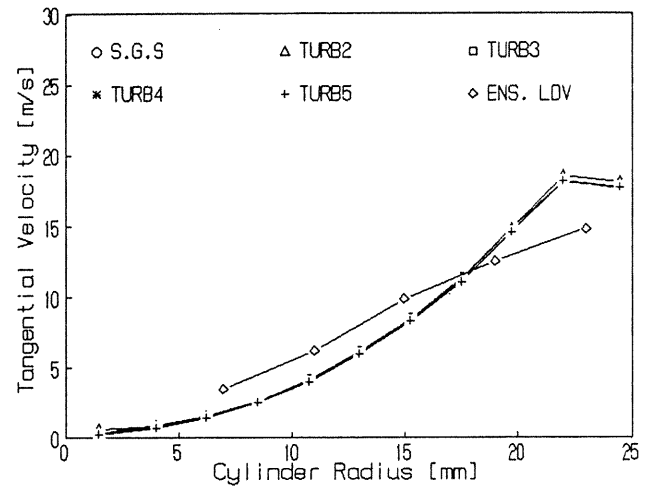


Fig.10 Measured and computed mean tangential velocity at 10 cad btcdc.

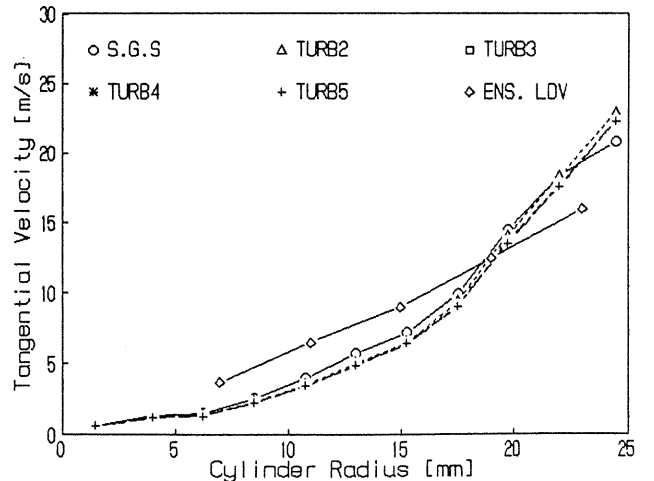


Fig.11 Measured and computed mean tangential velocity at tdc.

In the diagram of the figs.9,10 and 11 the measured and computed mean swirl velocity along cylinder radius are compared at 20 CAD BTDC, 10 CAD BTDC and at TDC respectively. The calculation performed with the four variants of the k-ε model, previously described, and with the LES model give quite similar results. The computed velocities underestimate the measurements near cylinder axis and overestimate near the combustion chamber wall. The overall agreement between computation and measurements seems acceptable.

The measured turbulence intensities both by ensemble averaged and cycle resolved techniques are compared with those computed by different turbulence models in the diagrams of figs.12,13 and 14. All the models give lower values of turbulence intensity than measured ones. The Watkins model (Turb2) predicts higher values than others and seems to fit well the cycle resolved turbulence intensities near cylinder axis while all models give similar results at the periphery of the bowl.

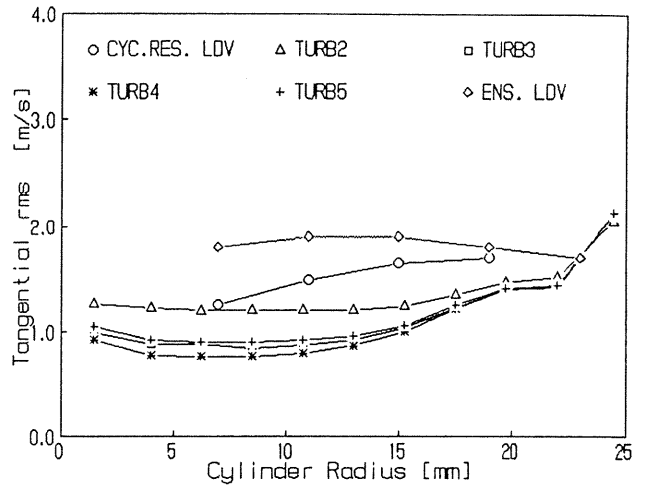


Fig.14 Measured and computed turbulence intensities at tdc.

In the diagram of the figs.15 and 16 the computed dissipation length scale versus cylinder radius at 10 CAD BTDC and at TDC are shown. All the turbulence models give similar values at the periphery of the bowl but the Turb2 model strongly differs from the others at the locations near cylinder axis.

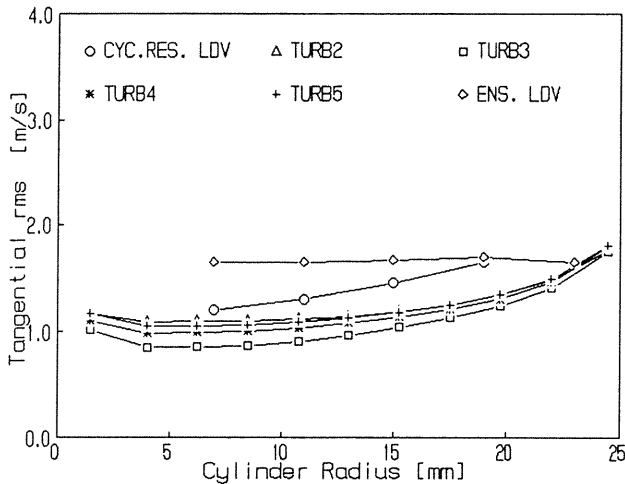


Fig.12 Measured and computed turbulence intensities at 20 cad btcd.

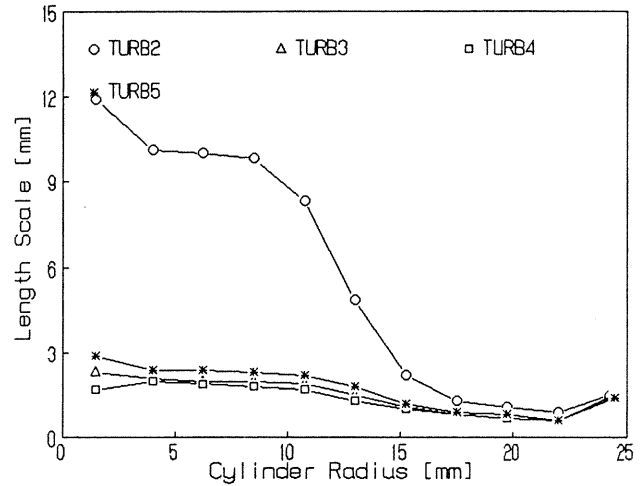


Fig.15 Computed dissipation length scale vs. cylinder radius at 10 cad btcd.

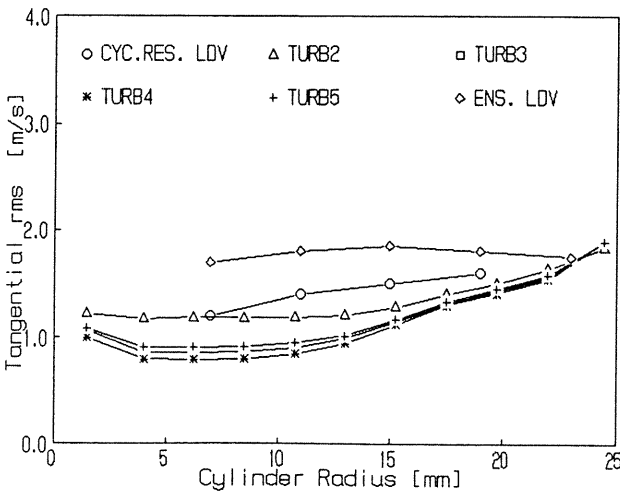


Fig.13 Measured and computed turbulence intensities at 10 cad btcd.

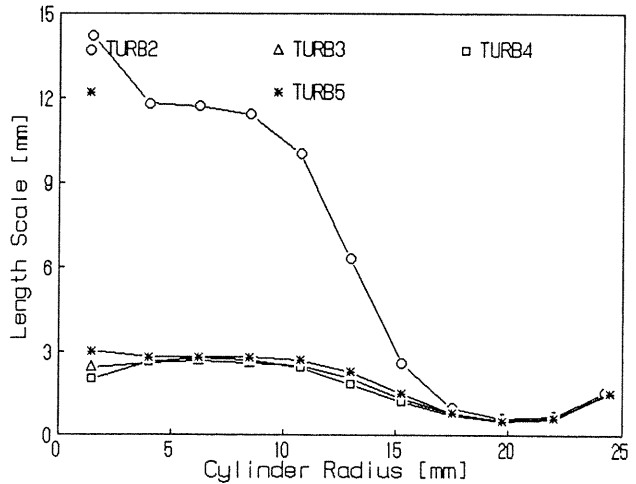


Fig.16 Computed dissipation length scale vs. cylinder radius at 10 tdc.

In the diagram of the fig.17 the macrolength scales, derived from cycle resolved single point LDA measurement, are shown. The order of magnitude of the measurements agree with the predictions of all variants of k- ϵ model with the exception of Watkins model that strongly overestimates the measured values near cylinder axis.

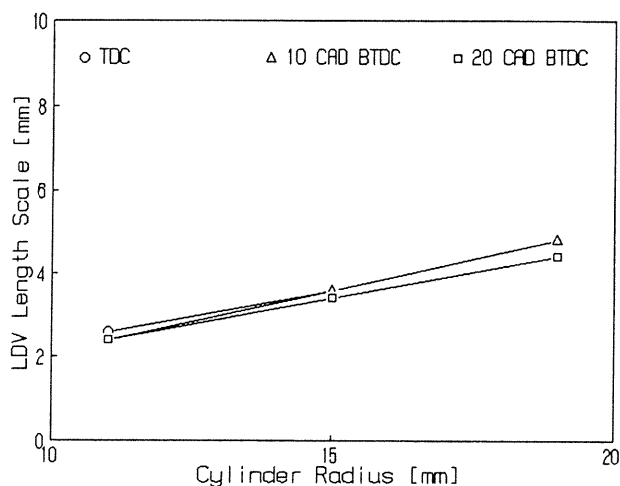


Fig.17 Macrolength scale derived from LDA measurement at 1000 rpm.

In conclusions the different variants of the k- ϵ model fit acceptably the measurements of mean swirl velocity and turbulence intensity. However the Watkins model predicts unacceptable values of the dissipation length scale during the compression stroke. Thus it appears unsuitable for engine applications.

These results fully confirm those obtained in ref.[6] at low engine speed and compression ratio. The discrepancies from measured and computed values can be due both to the experimental uncertainty of cycle resolved measurements and the choice of initial values for the calculations.

FINAL REMARKS

The cycle resolved measurements carried out in the combustion chamber of a single cylinder d.i. diesel engine by LDA have allowed to determine at high compression ratio the cyclic variation and longitudinal length scales of turbulence. Because the cyclic variation, determined by FFT filtering procedure, assumes about the same values of the absolute turbulence intensities, the choice of taking the cycle resolved measurements for the comparison with the computations is well appropriated. Despite the large approximations deriving from single point measurements, the order of magnitude of the length scales match

with the flow data present in the literature.

With a proper choice of the initial conditions a 3D model can fit with sufficient accuracy the mean swirl velocity evolution during the compression stroke. The different variants of k- ϵ model underestimate the cycle resolved turbulence intensities at TDC and the predicted values are very similar from a model to another.

The discrepancies between measurements and computations can be due to the uncertainties in cycle resolved data processing and to the choice of initial conditions for the calculations. Measured longitudinal length scales are of same order of magnitude of computed dissipation length scales with the exception of Watkins model. Thus this model appears unsuitable for engine applications.

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