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Hierarchical 1D/3D approach for the development of a turbulent combustion model applied to a VVA turbocharged engine. Part I: turbulence model

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Abstract

It is widely recognized that air-fuel mixing, combustion and pollutant formation inside internal combustion engines are strongly influenced by the spatial and temporal evolution of both marco- and micro- turbulent scales. Particularly, in spark ignited engines, the generation of a proper level of turbulence intensity for the correct development of the flame front is traditionally based on the onset, during the intake and compression strokes, of a tumbling macro-structure.
Recently, in order to both reduce pumping losses due to throttling and develop advanced and flexible engine control strategies, fully variable valve actuation systems have been introduced, capable of simultaneously governing both valve phasing and lift. Despite the relevant advantages in terms of intake system efficiency, this technology introduces uncertainties on the capability of the intake port/valve assembly to generate, at low loads, sufficiently coherent and stable structures, able therefore to promote adequate turbulence levels towards the end of the compression, with relevant effects on the flame front development.
It is a common knowledge that 3D-CFD codes are able to describe the evolution of the in-cylinder flow field and of the subsequent combustion process with good accuracy; however, they require too high computational time to analyze the engine performance for the whole operating domain. On the contrary, this task is easily accomplished by 1D codes, where, however, the combustion process is usually derived from experimental measurements of the in-cylinder pressure trace (Wiebe correlation). This approach is poorly predictive for the simulation of operating conditions relevantly different from the experimental ones. To overcome the above described issues, enhanced physical models for the description of in-cylinder turbulence evolution and combustion to be included in a 1D modeling environment are mandatory.
In the present paper (part I), a 0D (i.e. homogeneous and isotropic) phenomenological (i.e. sensitive to the variation of operative parameters such as valve phasing, valve lift, intake and exhaust pressure levels, etc.) turbulence model belonging to the K-k model family is presented in detail. The model is validated against in-cylinder results provided by 3D-CFD analyses carried out

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with the Star-CD™ code for motored engine operations. In particular, a currently produced small turbocharged VVA engine is analyzed at different speeds, with valve actuations typical of full load and partial load (EIVC) operations, as well.
The proposed turbulence model shows the capability, once tuned, to accurately estimate the temporal evolution of the in-cylinder turbulence according to the engine operating conditions. In the subsequent part II of the same paper, the developed turbulence model will be employed within a quasi-dimensional fractal combustion model.

Keywords: internal combustion engine; turbulence model; K-k; 0D/3D modeling.

Nomenclature

A	Area
B	Bore
c_{pkap}, c_{pkey}	Tuning constants of the mean and turbulent flow kinetic energy production
$c_{in.}$ c_{inv}, c_{ex}	Tuning constants of kinetic energy production term during the intake and exhaust phases
c_{tumb}	Constant related to the compression dependent turbulent kinetic energy and mean flow production
D	Turbulent kinetic energy dissipation term
$h_{\max}$	Maximum valve lift
k	Turbulence kinetic energy
K	Mean flow kinetic energy
L_I	Integral length scale
L_{Ij}	Tuning constants for the integral length scale reconstruction
m	Mass
$\dot{m}$	Mass flow rate
P	Turbulent kinetic energy production rate
U, u,v,w	Velocity magnitude, velocity components
u'	Turbulence fluctuation
U_f	Mean flow velocity
t	Time, characteristic time scale
V	Volume
Greeks	
$\Delta\theta$	Tuning constant for the integral length scale reconstruction
ε	Specific turbulent kinetic energy dissipation rate
ρ	Density
Subscripts	
cyl	Related to the cylinder
ex	Related to the exhaust
k	Related to turbulent kinetic energy
K	Related to mean flow kinetic energy
in	Related to the intake
pist	Related to the piston
valve	Related to a valve
Acronyms	
BDC, TDC	Bottom, Top dead center
BSFC	Brake specific fuel consumption
EIVC	Early intake valve closure
VVA	Variable valve actuation
TKE, MKE	Specific turbulent and mean kinetic energies

1. Introduction

The development of a modern internal combustion engine (ICE) has to comply with several simultaneous requirements. On one hand, in fact, customers are becoming increasingly sensitive to fuel consumption levels, mainly because of the uprising fuel costs and carbon taxes. On the other hand, car manufacturers are requested not to reduce, or even to increase, power/torque performance, in order to guarantee the desired level of drivability and fun-to-drive. Furthermore, both tasks have to comply with increasingly stringent pollutants emission limitations. To fulfill the above listed aims, often conflicting, the architecture of the ICEs is becoming more and more complex.

Concerning spark ignition ICEs (SI ICEs), advanced technical solutions involving the combination of a fully flexible actuation system of the intake valves (Variable Valve Actuation - VVA), a turbocharging system and gasoline direct injection are under continuous research and development. Particularly, VVA allows the designers to combine a fine intonation of the pressure waves in the intake system at full load (maximum cylinder filling) and a flexible load control without throttling the intake system (thus reducing fuel consumption). The use of either a turbocharging or a supercharging system aims at further reducing brake specific fuel consumption at low- and mid-loads through the reduction of engine displacement (downsizing), while preserving (or even increasing) peak engine torque at high load / low engine speed and peak engine power at high load / high engine speed. Finally, gasoline direct injection allows for both a wider valve overlapping (de-throttling) at low loads, and a lower knock risk [1] at high loads. For some applications, it can proficiently be used to generate a proper air/fuel charge stratification. The increased number of degrees of freedom related to the previously cited additional sub-systems makes the engine calibration a challenge of increasing complexity, to be experimentally accomplished over the whole engine operating space. For this reason, virtual design tools and numerical models can be successfully employed to guide and limit the usually more expensive experimental activities [2], [3], [4].

Nowadays, 0D/1D codes are used during all the stages of the engine development, from concept design to powertrain control development and calibration. However, they are generally based on empirical sub-models [5], [6] and often suffer from a lack of predictive capability, especially concerning the in-cylinder phenomena. To overcome this issue, 0D phenomenological models have been developed to describe in-cylinder turbulence evolution and combustion on a more robust physical basis. They, once tuned, are able to guarantee an adequate consistency also in absence of experimental data. Consequently, they can be successfully integrated in automatic procedures to perform a “virtual calibration” of the engine to achieve the required targets [2] (maximum power/torque at full load, minimum fuel consumption and/or pollutants emissions at part load, etc.).

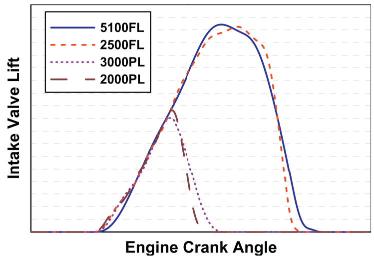
A 0D turbulence model aims at synthesizing very complex 3D phenomena based on the assumption that turbulence is both homogenous (i.e. equally distributed in the space) and isotropic (i.e. without a preferential direction in space). Several 0D turbulence models are available in the technical literature. The so called K-k model family describes the energy cascade mechanism from the average flow field kinetic energy (K) to the turbulence fluctuation kinetic energy (k) by an ordinary derivative two-equation system [7]. In other cases, macro-scale kinetic energy is imposed [8], while balance equations are solved for the turbulence fluctuation kinetic energy and the dissipation rate [9], [10]. Further approaches require the computation of the ordinated flow fields (swirl and tumble), apart from the turbulent kinetic energy and the dissipation rate [11], [12].

The attitude of the intake system to create turbulent structures is usually experimentally investigated for simplified steady operations of the system: a steady flow bench is used, and measurements are carried out for fixed valve positions / fixed pressure drops and with no piston. The use of optically accessible engines to investigate the actual unsteady process is much more unusual and is limited to research laboratories. As a consequence, CFD analysis of the intake and compression strokes becomes a very powerful tool to analyze the in-cylinder flow structure evolution under actual engine operations, as well as to address the influence of port design on the in-cylinder flow field [13].

In this paper a hierarchical 1D/3D approach is employed: information from 3D-CFD analyses, in terms of temporal evolution and spatially averaged in-cylinder flow fields, both mean and turbulent [14], are utilized to set-up and calibrate an in-house developed 0D turbulence model. The latter is included as a user routine in the well known commercial code GT-Power^{TM}. Calculations are performed with reference to a small twin-cylinder VVA engine, whose main features are listed in Table 1 below. Motored operation is analyzed at different engine speeds and valve lift strategies typical of full-load (labeled as 2500FL and 5100FL) and part-load (2000PL and 3000PL), reported aside Table 1.

Table 1. Engine main features and valve lift profiles used in the following analysis

Model	2 cyl., 8 valves, intake VVA, turbocharged
Displacement	875 cm^3
Bore / Stroke	86 mm / 80.5 mm
Connecting rod length	136.85 mm
Compression ratio	9.9
Power @ rpm	64.6 KW @ 5500 rpm
Torque @ rpm	146.1 Nm @ 2500 rpm



The methodology is based on a preliminary 1D simulation, performed to provide realistic boundary conditions for the subsequent 3D-CFD analyses. Related results are then utilized to identify the set of tuning constants required to calibrate the 0D turbulence model. In the following, the developed turbulence model is described in detail. Then, the set-up of the 3D-CFD calculations and their findings are illustrated. Finally, the comparison between 3D and 0D turbulence modeling is presented.

2. 0D Turbulence model

The model describes the temporal rates of the mean and turbulent kinetic energies, equations (1), (2):

$$K=\frac{1}{2}m_{cyl}U_{flow}^2 \qquad \frac{dK}{dt}=\left(\frac{dK}{dt}\right)_{valve}-P_K+c_{tumb,K}K\frac{\dot{\rho}_{cyl}}{\rho_{cyl}}-K\frac{\dot{m}_{ex}}{m_{cyl}}+K\frac{u_{pist}}{\bar{u}_{pist}}\frac{du_{pist}}{\bar{u}_{pist}} \qquad (1)$$

$$k=\frac{3}{2}m_{cyl}u'^2 \qquad \frac{dk}{dt}=\left(\frac{dk}{dt}\right)_{valve}+P_k+c_{tumb,k}k\frac{\dot{\rho}_{cyl}}{\rho_{cyl}}-k\frac{\dot{m}_{ex}}{m_{cyl}}-D \qquad (2)$$

The first term in both equations accounts for the kinetic energy produced by the flow through the valves (both intake and exhaust). P_K and P_k represent the K decay and the k production term, respectively. In both the equations, the third term accounts for the kinetic energy production due to the piston motion close to compression TDC; in particular, in equation (1), this term mimics the tumble macro-vortex compression responsible for the increase of the mean-flow velocity, while, in equation (2), it simply describes an isotropic compression according to the theory [15], [16]. The compression terms are multiplied by the constants, $c_{tumb,K}$ and $c_{tumb,k}$, respectively. They depend on a single tuning constant, c_{tumb}, and on the maximum valve lift, $h_{\max}$ according to:

$$c_{tumb,k}=c_{tumb}*h_{\max} \qquad c_{tumb,K}={}^{1}\!/_{3}\,c_{tumb,k} \qquad (3)$$

The fourth terms accounts for the convective flow through the exhaust valves. Last term in equation (1) takes into account the effects of piston motion, at velocity u_{pist}, on the mean flow field. As shown below, this contribution is relevant especially during the expansion stoke, while in the remaining portion of the cycle it is masked by the other effects. The D term in equation (2) is the dissipation rate of k into heat, derived as follows:

$$D=m_{cyl}\varepsilon \qquad \varepsilon=0.09^{3/4}\frac{k^{3/2}}{L_I} \qquad (4)$$

L_I being the integral length scale. The production term related to the flow through the valves in equation (1) is calculated as:

$$\left(\frac{dK}{dt}\right)_{valve} = \left(c_{in}\frac{1}{2}\dot{m}_{in}u_{in}^2 + c_{ex}\frac{1}{2}\left|\dot{m}_{ex}\right|u_{ex}^2\right) + c_{inv}\frac{1}{2}\dot{m}_{ex}u_{cyl}^2 \quad (5)$$

being $\dot{m}_{in}$, $\dot{m}_{ex}$, u_{in}, and u_{ex} the mass flow rates and the flow speeds through the intake and the exhaust valves, respectively. The above terms are taken into account only when the gas flows from the ports towards the cylinder (i.e. direct intake flow and reverse exhaust flow). Direct flow through the exhaust valve is indeed considered in the last term, with reference to an exhaust flow related mean velocity u_{cyl}, evaluated as follows:

$$u_{cyl} = \frac{\dot{m}_{ex}}{\rho_{cyl}A_{pist}} \qquad A_{pist} = \frac{\pi B^2}{4} \quad (6)$$

Concerning k equation (2), the valve related production term is more simply computed as:

$$\left(\frac{dk}{dt}\right)_{valve} = \frac{k}{K}\left(c_{in}\frac{1}{2}\dot{m}_{in}u_{in}^2 + c_{ex}\frac{1}{2}\left|\dot{m}_{ex}\right|u_{ex}^2\right) \quad (7)$$

c_{in}, c_{ex} and c_{inv} parameters in equations (5) and (7) are tuning constants; the only one given as an input is c_{in}, while the others are derived from:

$$c_{ex} = 0.1 * c_{in} \qquad c_{inv} = c_{in} \quad (8)$$

The terms P_K and P_k are calculated according to:

$$P_K = c_{pkap}\frac{K}{t_K} \qquad P_k = c_{pkey}\frac{k}{t_k} \quad (9)$$

c_{pkap} and c_{pkey} being additional tuning constants. t_K and t_k are the characteristic time scales of K decay and of k production, evaluated as:

$$t_K = \frac{L_I}{U_{flow}} \qquad t_k = \frac{L_I}{u'} \quad (10)$$

Previous investigations [17],[18] showed that the time evolution of the integral length scale L_I is only slightly depending on engine operating conditions. It can be hence prescribed as an assigned sequence of Wiebe-like functions (see paragraph 6). The control points of these equations are given as an input; they represent additional tuning constants for the turbulence model.

3. Hierarchical 3D-1D approach

Geometrical data provided by the engine manufacturer are used to build the 1D model of the whole engine within the GT-Power environment and to carry out the preliminary 1D simulations in motored operations. Then, a multi-cycle 3D analysis, based on 1D computed time dependent pressure and temperature boundary conditions, is performed. Detailed information about the in-cylinder mean and turbulent flow fields are extracted from the 3D analyses over a whole engine cycle, to be supplied to the 1D code. In particular, the mass averaged integral length scale, specific turbulent kinetic energy and mean flow kinetic energy, eq. (11) are calculated all over the 3D domain.

$$L_I = \frac{\sum_{CYL-CELL} \rho_i V_i L_{I,i}}{\sum_{CYL-CELL} \rho_i V_i} \qquad TKE = \frac{\sum_{CYL-CELL} \rho_i V_i k_i}{\sum_{CYL-CELL} \rho_i V_i} \qquad MKE = \frac{\sum_{CYL-CELL} \frac{1}{2} \rho_i V_i U_i^2}{\sum_{CYL-CELL} \rho_i V_i} \tag{11}$$

ρ_i, V_i, U_i, k_i, and $L_{I,i}$ are the density, the volume, the mean flow velocity, the specific turbulent kinetic energy and the integral length scale within the i-th cell, respectively. $L_{I,i}$ is defined as a function of the local dissipation rate ε_i, while the mean flow velocity accounts for its three local components, u_i, v_i, w_i:

$$L_{I,i} = 0.09^{3/4} \frac{k_i^{3/2}}{\varepsilon_i} \qquad U_i = \sqrt{u_i^2 + v_i^2 + w_i^2} \tag{12}$$

In order to compare 3D and 0D results, the following 3D-derived mean flow and turbulent velocities are defined:

$$u'_{3D} = \sqrt{\frac{2}{3} TKE} \qquad U_{flow,3D} = \sqrt{2MKE} \tag{13}$$

4. 3D model setup

Fully transient in-cylinder simulations are carried out using Star-CD, licensed by CD-adapco, for different engine speeds and valve lift strategies under motored conditions. The investigated cases are listed in Table 2.

Table 2. Tested engine operations

Case Number	Engine speed, rpm	Intake valve closure angle, deg after CTDC	Label
1	2500	560	2500FL
2	5100	548	5100FL
3	2000	462	2000PL
4	3000	448	3000PL

The computational domain covers the combustion chamber together with the port portions within the engine head. The mesh is mainly made of hexahedral cells, which are progressively distorted, activated and deactivated in order to take into account the piston and valve motion. The near-valve cell size is chosen in order to guarantee mesh uniformity during the intake and exhaust strokes. All the operating conditions are analyzed using the k-ε turbulence model in its high-Reynolds formulation. The latter provides the local values of the specific turbulent kinetic energy and dissipation rate, that are used in the equations (11). The solution of the Reynolds Averaged Navier-Stokes equations (RANS) indeed supplies the local velocity components u_i, v_i, w_i.

In order to take into account the turbulence model near-wall requirements, a single prismatic layer of 0.3 mm in height is used. The resulting grid, whose characteristics are kept fixed for all the investigated cases, is made up of nearly 560.000 and 320.000 cell at BDC and TDC respectively.

Figure 1 shows a detail of the resulting grid along two plane sections passing through the intake and exhaust valves axes respectively, both at maximum valve lift positions and for the 2500FL case. For all the calculations, time dependent pressure and temperature boundary conditions are derived from the 1D model of the whole engine. Spatially uniform initial conditions for the very first engine cycle are imposed. Heat transfer through the combustion chamber walls is accounted for by setting fixed temperatures on the different wall regions and adopting the heat transfer model by Angelberger. All the analyses start from the exhaust stroke and cover a full engine cycle. Variable time steps according to the cycle events are set. Particularly, the adopted time steps range from 0.1 CAD down to 0.025 CAD during the valve openings and closures, in order to accomplish the Courant–Friedrichs–Lewy (CFL) condition throughout the simulation. The monotone advection and reconstruction 2nd order differencing scheme is used for each of the mass, momentum, energy, turbulence and scalar transport equations. As mentioned in the previous section, in-cylinder data post-processing is performed in terms of time-history of both mean quantities (mass, pressure, temperature, velocity magnitude) and turbulent ones (turbulence intensity, length scale).

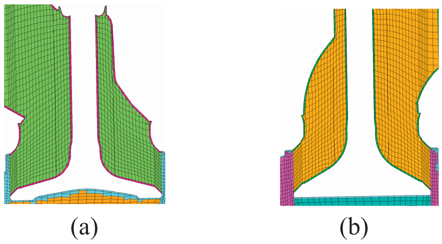


Figure 1. Detail of the intake (a) and exhaust (b) valve curtain grid. 2500FL case

5. 3D results

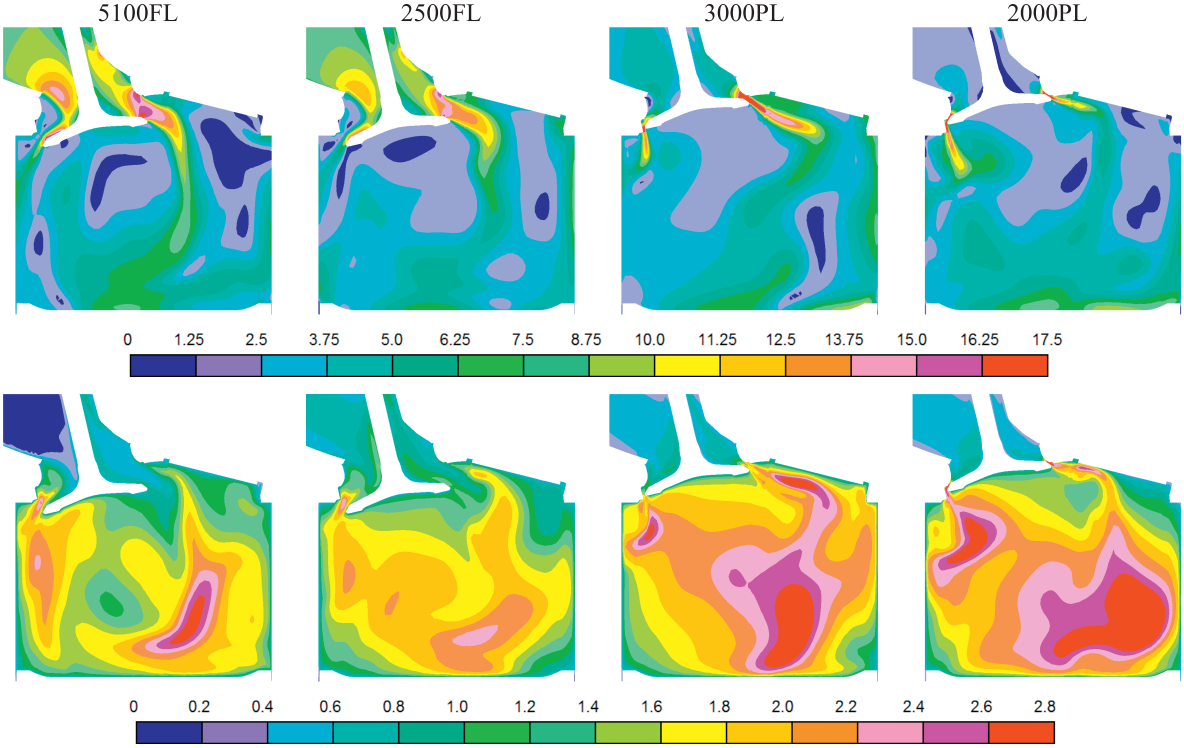


Figure 2. Normalized mean (above) and turbulent (below) velocity @450 CAD

As said before, data exchange between the 1D and the 3D software is preceded by a 3D multi-cycle analysis, in order to obtain initial condition-independent in-cylinder data. To reach a satisfactory periodic convergence, a minimum of three subsequent engine cycles are simulated for each operation. Figure 2 shows a comparison between all the considered operations in terms of velocity magnitude (a) and velocity fluctuation (b), normalized with the mean piston speed, a few crank angle degrees before the intake valve closure position for the 2000PL case. The reported view is obtained on a plain parallel to the cylinder axis and passing through the axis of one of the intake valves. As visible from the pictures, different intake valve actuation strategies lead to massively different in-cylinder flow patterns in terms of both distribution and intensity. Despite the huge differences, the analysis of in-cylinder flow structures during the compression stroke shows a progressive realignment of the calculated fields. For example, Figure 3 shows the integral length scale contours on a plain section normal to the cylinder axis, at a z-coordinate position 1.5 mm above the squish height. As visible, similar patterns (in both intensity and distribution) can be observed for all the cases. This reveal in advance the possibility to prescribe a case-independent L_I evolution in the 0D model.

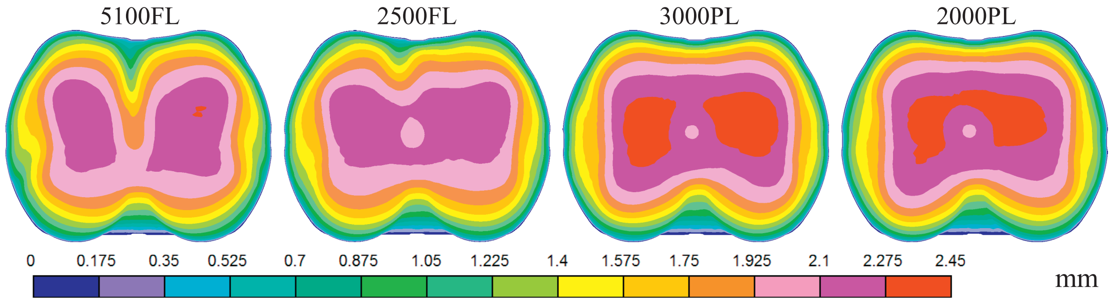


Figure 3. Integral Length scale @720 CAD

6. 0D turbulence model validation

As stated earlier, the turbulence model is tuned against 3D-CFD motored findings for different speeds and intake valve closure angles typical of full and part load operations, as well (Table 2). First of all, a reliable time evolution of the integral length scale must be assigned. This is done by employing a sequence of Wiebe-like functions passing through specified control points. In particular, 3D results show that the mass averaged L_I (equation 11), starting from a very low TDC level, L_{I3}, continuously increases during the intake stroke and reaches its maximum during the compression stroke. The location of the above maximum is prescribed in the 0D model in terms of angular position, $\Delta\theta$, and height, L_{I2}. Then, the integral scale reduces, reaching a minimum at the compression TDC (L_{I1}). During the expansion and exhaust strokes, L_I mainly follows the in-cylinder height above the piston, increasing once again up to the L_{I2} level at the expansion BDC and reducing down to the L_{I3} level at the valve overlap TDC. The different phases previously described are connected through a sequence of Wiebe-like functions, as shown in Figure 4. Parameters L_{Ij}, (j=1,3), and $\Delta\theta$ represent of course, further unknown constants.

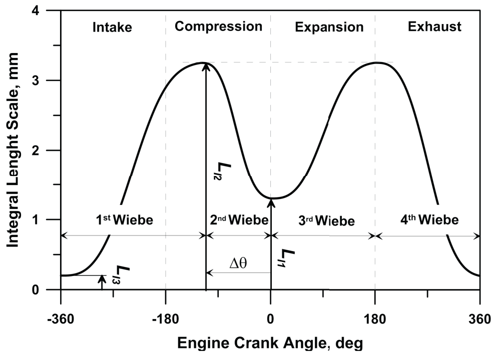


Figure 4. Integral length scale reconstruction.

Table 3. Optimal values of turbulence model constants

L_{I1}	1.3 mm	L_{I2}	3.25 mm	L_{I3}	0.2 mm	$\Delta\theta$	120 deg
c_{in}	0.7	c_{tumb}	0.19	c_{pkey}	0.03	c_{pkap}	0.05

Once the integral length scale is formally identified, each tuning constant of the 0D model has to be set to fit the actual results provided by the 3D-CFD calculations. Through a simple trial and error procedure, a single set of optimal constants is selected as listed in Table 3. It is not worthless to emphasize that, for all the considered cases, the tuning constants are kept unchanged to demonstrate the generality of the proposed model.

Figure 5 shows the comparison between 3D and 0D results, for the integral length scale and for all the operating points. For sake of clarity, the attention is focused on the intake, compression and half expansion stroke portion of the cycle. The agreement is satisfactory for all the cases. It is confirmed that the trends of 3D computed integral length scale depend only slightly on the engine operating conditions, especially in the neighbourhood of the compression TDC. This allows to use a unique L_I reconstruction for all considered operating points.

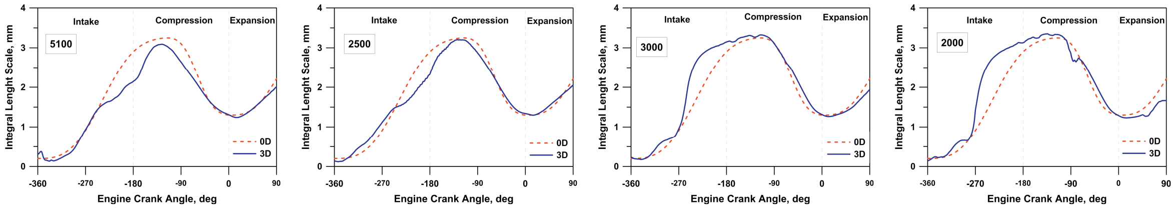


Figure 5. Integral length scale 0D/3D comparison

Figure 6 highlights that the proposed model is able to forecast the in-cylinder turbulence levels during the combustion relevant portions of the cycle, i.e. intake, compression and expansion phases. This mainly depends on the model capability to sense the different durations of the intake phases for the considered cases. The different turbulence levels at the compression TDC are very well predicted thanks to the compression term in equation (2), which properly accounts for the different valve lift profiles, mainly through the maximum lift correction term included in the definition of the tuning constant $c_{tumb,k}$. The turbulence fluctuation (Fig. 6) and the mean flow velocity (Fig. 7) are normalized with the piston mean speed.

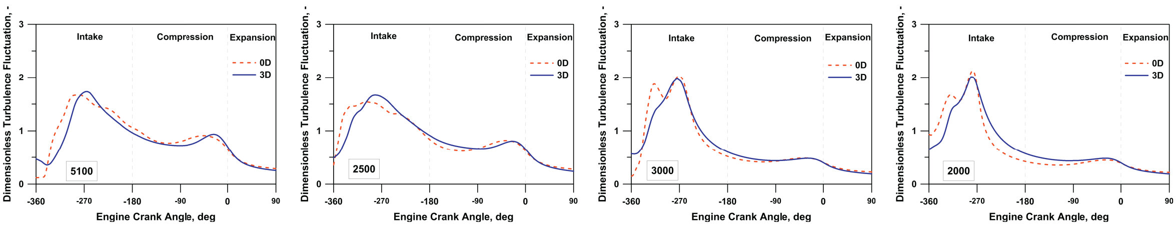


Figure 6. Dimensionless turbulence fluctuation 0D/3D comparison

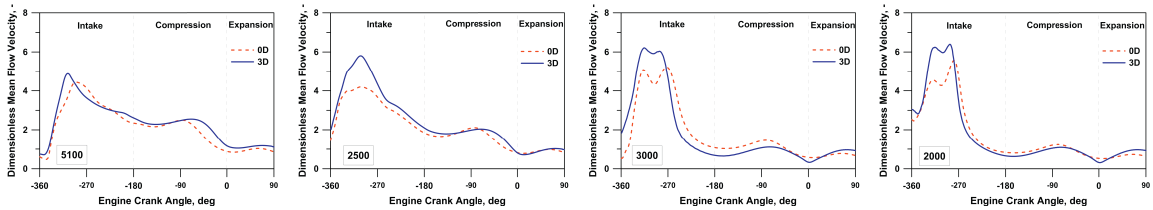


Figure 7. Dimensionless mean flow velocity 0D/3D comparison

Since equations (1) and (2) are strongly coupled, it is also important to verify the accuracy in the prediction of the mean flow velocity, reported in Figure 7. It can be appreciated that a good agreement is reached, especially during the compression and expansion strokes. Concerning the compression stroke, the density-dependent term of equation (1) is able to capture the local maximum before TDC, while the local maximum during the expansion is described by the term related to the piston movement. Simultaneously looking at Figures 5, 6 and 7 it is demonstrated that the 0D model successfully describes the energy cascade mechanism from the mean to the turbulent flow and it represents the basis for an accurate development of the combustion model, discussed in part II.

7. Conclusions

In the present paper, part I, a 0D turbulence sub-model to be included in a phenomenological combustion model is presented. The model belongs to the K-k family and describes the energy cascade from the mean flow scale to the turbulence scale. Numerical analyses on a small twin-cylinder VVA turbocharged engine are presented. In order to validate the turbulence model, 3D-CFD simulations of the motored flow field inside the cylinder are carried out for different engine speeds and largely different intake valve lift strategies. Boundary conditions for the 3D simulations are preliminary derived from the 1D code. Without any case-dependent tuning, the 0D turbulence model is able to satisfactory fit the 3D findings for all the considered operating conditions, especially during the compression and expansion strokes. This demonstrates the capability of the model to sense both different flow conditions through the valves (according to the engine speed) and different strategies for the intake valve actuation.

The proposed methodology represents a successful example of integration of a refined 3D approach and a simplified 0D one. The latter is used to reliably evaluate the engine performance, through a fractal combustion model described in part II of the paper.

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