

Development of a Phenomenological Turbulence Model through a Hierarchical 1D/3D Approach Applied to a VVA Turbocharged Engine

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ABSTRACT

It is widely recognized that spatial and temporal evolution of both macro- and micro- turbulent scales inside internal combustion engines affect air-fuel mixing, combustion and pollutants formation. Particularly, in spark ignition engines, tumbling macro-structure induces the generation of a proper turbulence level to sustain the development and propagation of the flame front.

As known, 3D-CFD codes are able to describe the evolution of the in-cylinder flow and turbulence fields with good accuracy, although a high computational effort is required. For this reason, only a limited set of operating conditions is usually investigated. On the other hand, thanks to a lower computational burden, 1D codes can be employed to study engine performance in the whole operating domain, despite of a less detailed description of in-cylinder processes. The integration of 1D and 3D approaches appears hence a promising path to combine the advantages of both.

In the present paper, a 0D phenomenological mean flow and turbulence model belonging to the $K-k$ model family is presented in detail. The latter is implemented in the GT-Power™ software under the form of “user routine”. The model is tuned against in-cylinder results provided by 3D-CFD analyses carried out by the Star-CD™ code at two engine speeds under motored operation. In particular, a currently produced twin-cylinder turbocharged VVA engine is analyzed. The 0D model is then validated against further 3D results at various engine speeds and intake valve lifts, including early closure strategies, both under motored and fired operation.

The proposed 0D mean flow and turbulence model shows the capability to accurately estimate the temporal evolution of the incylinder turbulence for all the considered operating conditions, without requiring any case-dependent tuning, proving its generality and reliability.

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INTRODUCTION

The development of a modern Internal Combustion Engine (ICE) has to comply with several simultaneous requirements. On one hand, 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 provide the desired level of drivability and fun-to-drive. Furthermore, both tasks have to comply with increasingly stringent pollutants emission limitations [1]. To fulfill the above listed aims, often conflicting, the architecture of the ICEs is becoming more and more complex.

Concerning spark ignition ICEs, advanced technical solutions such as Variable Valve Actuation (VVA) and/or Variable Valve Timing (VVT), turbocharging, and gasoline direct injection, are under continuous research and development. Particularly, VVT 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 with a reduced throttling of the intake system (thus reducing fuel consumption) [2,3]. The use of a turbocharging system aims at further lowering brake specific fuel consumption at low- and mid- loads through the reduction of the engine displacement (downsizing) [4], while preserving (or even increasing) peak torque at low speed and peak power at high speed. Finally, gasoline direct injection allows a

wider valve overlapping (scavenging) at low loads, and a lower knock risk [5,6] at high loads. For some applications, it can be proficiently employed to generate a proper air/fuel charge stratification.

Nowadays, 1D codes are used during all the stages of the engine development, from concept design to powertrain control development and calibration. However, they generally rely on empirical models [7,8] and often suffer from a lack of predictive capability, especially concerning the in-cylinder phenomena. To overcome this issue, 0D phenomenological combustion models have been developed with the aim to describe the burning rate on a more robust physical basis. Once tuned, they have the potential to provide an adequate consistency also in absence of experimental data. Current literature is plenty of papers describing quasi-dimensional combustion models, such as eddy burn-up models [9,10,11], flame surface density approaches [12], and fractal models [13,14]. Of course, whichever is the employed approach, results are always strictly dependent on the reliability of the turbulent sub-model coupled to the combustion one.

The development of a 0D mean flow and turbulence model is a very challenging task, since it aims to synthesize very complex 3D phenomena. A 0D approach, indeed, automatically requires the assumption of in-cylinder homogenous (i.e. evenly distributed in the space) and isotropic (i.e. without a preferential direction in space) mean and turbulent distributions.

Under the above hypotheses, several 0D models have been developed in the past. The so-called *K-k* model family describes the energy cascade mechanism from the macroscale kinetic energy (*K*) to the turbulent kinetic energy (*k*) by an ordinary derivative two-equation system [15]. In other cases, macro-scale kinetic energy is imposed [12], while balance equations are solved for turbulent kinetic energy and dissipation rate [16,17,18]. Further approaches require in addition the characterization of structured flow fields, such as swirl and/or tumble motion [19,20]. Developed models, however, lack in an extensive validation [12], and/or require a case-dependent tuning [17]. At authors' knowledge, extension to VVA engines, where turbulence production and dissipation rates also depend on the actuated valve strategy, is moreover absent.

This lack of predictive capabilities is sometimes recovered by the employment of experimental data, such as the tumble and swirl ratios, which synthesize the attitude of the intake system to create ordinate flow structures. The measurement of the above data is carried out by steady flow benches, for fixed valve positions / fixed pressure drops, and with no piston. However, their employment in a 1D unsteady simulation is questionable, since, as an example, those data do not provide any direct information about the turbulence production related to the tumble vortex collapse towards the end of the compression stroke.

Consequently, unsteady 3D-CFD analyses of the intake and compression phases become a very powerful tool to better analyze the in-cylinder flow structure evolution under actual engine operations, as well as to address the influence of port layout, combustion chamber design, and valve strategy, on the in-cylinder flow and turbulence fields [21].

With the aim of improving the accuracy of available 0D turbulence models, in this paper a hierarchical 1D/3D approach is followed: information from 3D-CFD analyses, in terms of temporal evolution of both mean and turbulent flow fields, are utilized to set-up, calibrate and validate an in-house developed 0D phenomenological model. 3D simulations are set up in Star-CD™ framework, while 1D simulations are performed in GT-Power™ environment. Analyses refer to a small twin-cylinder turbocharged VVA engine. The main features of the considered engine are listed in Table 1. This engine is equipped with a fully flexible actuation system for the intake valves, which is able to properly adjust the intake lift profile according to the engine operating conditions. The consistency and generality of 0D model here presented is hence tested with reference to different valve strategies, under both motored and fired operation.

Table 1. Engine main characteristics

Model	2 cylinders, 8 valves, intake VVA, waste-gated turbocharger
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

HIERARCHICAL 1D/3D APPROACH

The proposed hierarchical 1D/3D approach is synthesized in the flowchart of Figure 1. First, geometrical data provided by the engine manufacturer are used to implement the 1D schematization of the whole engine in the GT-Power environment. A reduced number of engine operating conditions are selected (in the present work, two operating points) and for each of them, a preliminary 1D simulation is carried out under motored operation. Then, a 3D schematization of a cylinder and of the related port portions within the engine head is setup in Star-CD environment. Hence, a multi-cycle 3D analysis, based on 1D-computed time-dependent pressure and temperature boundary conditions, is performed. Detailed information about the incylinder distribution of mean and turbulent flow fields are extracted from the 3D calculations over the whole engine cycle and then synthesized in representative mass-averaged data. The latter are hence used to tune the corresponding 0D phenomenological model, implemented in GT-Power under the form of “user routine”.

It is important to underline that the outlined approach refers to motored operation for three main reasons:

1. a 3D analysis under motored operation is much less time consuming than the corresponding fired simulation;
2. boundary conditions derived by the 1D model under motored operation are unique, not depending on the tuning of a combustion model; hence a continuous loop is not required to reach consistency between 1D and 3D approaches;
3. as showed in the following, mass-averaged mean flow and turbulent fields are only slightly affected by combustion and flame propagation phenomena.

Once tuned as described in Figure 1, the 0D model will be validated under fired operation and for various valve strategies, so proving the reliability and generality of the proposed approach.

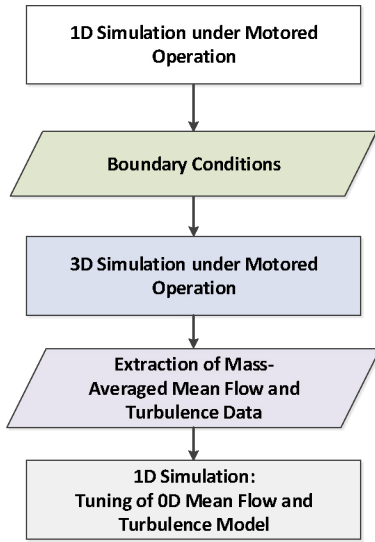


Figure 1. Flowchart of hierarchical 1D/3D approach

3D MODEL SETUP

The computational domain for the engine in Table 1 is built in Star-CD 4.22, licensed by CD-Adapco. It covers the combustion chamber together with the port portions within the engine head. The mesh is mainly made of hexahedral cells (nearly 750.000 and 330.000 cell at BDC and TDC respectively), which are progressively distorted, activated and deactivated in order to take into account both piston and valve motions. A single prismatic layer of 0.3 mm in height is used to model the near-wall region. The cell size in the valve-curtain zone is chosen in order to guarantee mesh uniformity during the intake and exhaust strokes. The $k-\varepsilon$ RNG turbulence model is employed for all the considered operating points.

As said, for all calculations time-dependent pressure and temperature boundary conditions are derived from the 1D simulations of the whole engine. Spatially uniform initial conditions for the 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 at least three full engine cycles, until a periodic solution is reached. 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 condition throughout the simulation. The monotone advection and reconstruction 2nd order differencing scheme is used for mass, momentum, energy, turbulence and scalar transport equations. At the end of each calculation, in-cylinder data post-processing is performed in terms of time-history of both mean quantities (mass, pressure, temperature, velocity magnitude) and

turbulent ones. In particular, mass-averaged specific turbulent kinetic energy, TKE , specific mean flow kinetic energy, MKE , and specific dissipation rate, ε , are calculated as:

$$TKE = \frac{\sum_{CYL-CELL} \rho_i V_i TKE_i}{\sum_{CYL-CELL} \rho_i V_i} \quad (1)$$

$$MKE = \frac{\sum_{CYL-CELL} \rho_i V_i U_i^2}{\sum_{CYL-CELL} \rho_i V_i} \quad (2)$$

$$\varepsilon = \frac{\sum_{CYL-CELL} \rho_i V_i \varepsilon_i}{\sum_{CYL-CELL} \rho_i V_i} \quad (3)$$

where ρ_i , V_i , U_i , TKE_i and ε_i are density, volume, mean flow velocity, specific turbulent kinetic energy and dissipation rate within the i -th cell, respectively. The mean flow velocity accounts for its three local components, u_i , v_i and w_i , as follows:

$$U_i = \sqrt{u_i^2 + v_i^2 + w_i^2} \quad (4)$$

Representative 3D-derived mean and turbulent flow velocities are computed based on the definition of mean flow and turbulent kinetic energies:

$$U_{flow} = \sqrt{2MKE} \quad (5)$$

$$u' = \sqrt{\frac{2}{3} TKE} \quad (6)$$

while the average integral length scale is calculated as:

$$L_t = 0.09^{\frac{3}{4}} \frac{TKE^{\frac{3}{2}}}{\varepsilon} \quad (7)$$

3D MODEL RESULTS

The whole set of investigated operating condition is summarized in Table 2. They refer to different engine speeds and valve strategies typical of full load (FL) and part load (PL) operation. Some of the selected valve lift profiles are reported in Figure 2. In particular, in the operating points labelled as 1500PL, 3000PL and 5500PL, intake valve strategies with an advanced closure angle and a reduced maximum lift are applied, while in 1500FL, 2100FL and 5500FL, more conventional valve profiles are considered, all characterized by a maximum lift of about 8 mm (full lift).



Table 2. Tested engine operations. (*) Cases used for 0D model tuning

Case Number	Engine speed, rpm	IVC angle, CAD after CTDC	Label	Conditions
1	2100	541	2100FL	Motored(*)/Fired
2	5500	572	5500FL	Motored(*)/Fired
3	1500	531	1500FL	Motored
4	1500	455	1500PL	Motored
5	3000	465	3000PL	Motored
6	5500	495	5500PL	Motored

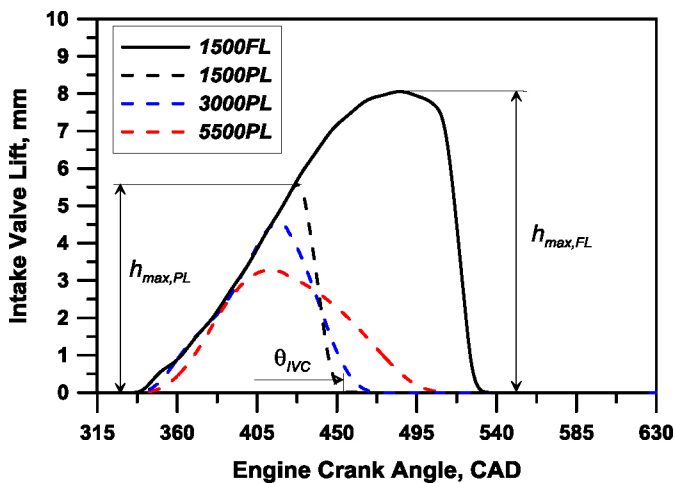


Figure 2. Intake valve lift profiles used for 3D analyses

Among the six cases reported in Table 2, two motored operating conditions are preliminary selected, namely 2100FL and 5500FL, to setup and tune the 0D model. Figure 3 to Figure 5 compare the predicted in-cylinder mean and turbulent flow fields for the above cases. In order to properly relate results at different engine speeds, both mean and turbulent quantities are normalized by the mean piston speed. In particular, Figure 3 and Figure 4 report the non-dimensional mean and turbulent flow velocity distribution at the maximum valve lift crank angle on a section plane cutting an intake valve and parallel to the cylinder axis. As visible, although a tumble-promoting flow pattern can be recognized for both engine speeds, both mean and turbulent flow patterns show relevant differences. This is mainly due to the much higher penetration of the intake jet flow, visible for the highest speed, which promotes higher turbulence levels close to the cylinder liner and piston top.

Conversely, Figure 5 shows the turbulent intensity field on a section plane orthogonal to the cylinder axis cutting the combustion chamber at a distance of 1.5 mm above the flame deck. As visible, the previously outlined relevant differences in the in-cylinder flow structures are now largely smeared off, thus confirming the progressive realignment of the calculated fields [22].

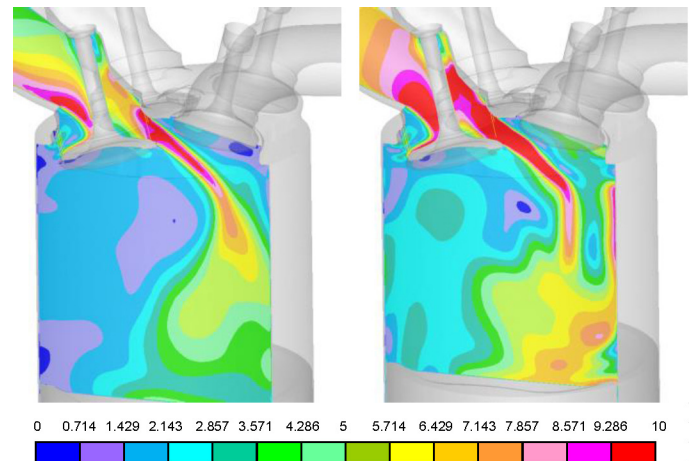


Figure 3. Non-dimensional mean velocity at 2100 (left) and 5500 rpm (right)

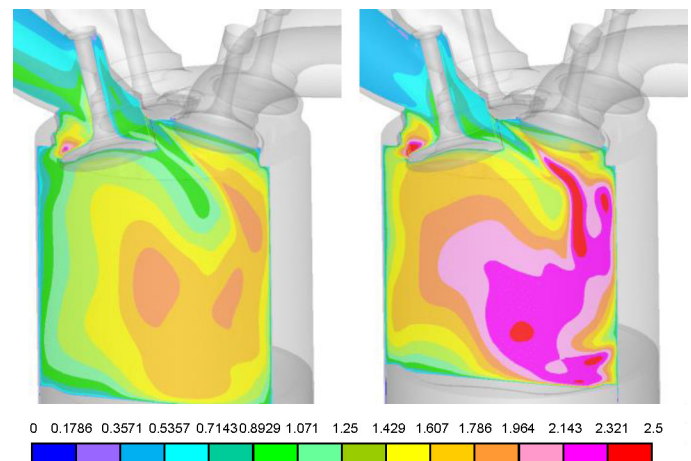


Figure 4. Turbulent intensity at 2100 (left) and 5500 rpm (right)

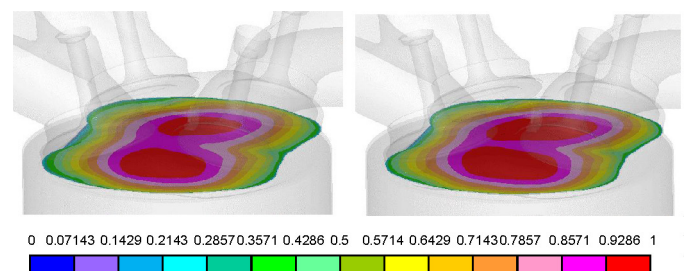


Figure 5. Turbulent intensity at 2100 (left) and 5500 rpm (right) 720°C

Detailed 3D results are hence synthesized in terms of U_{flow} , u' and L_p by the Equations (5), (6) and (7), respectively, during the whole engine cycle. The angular evolutions of U_{flow} and u' are shown in Figure 6 for the operating point at 2100FL, while the same quantities at 5500FL are plotted in Figure 7.

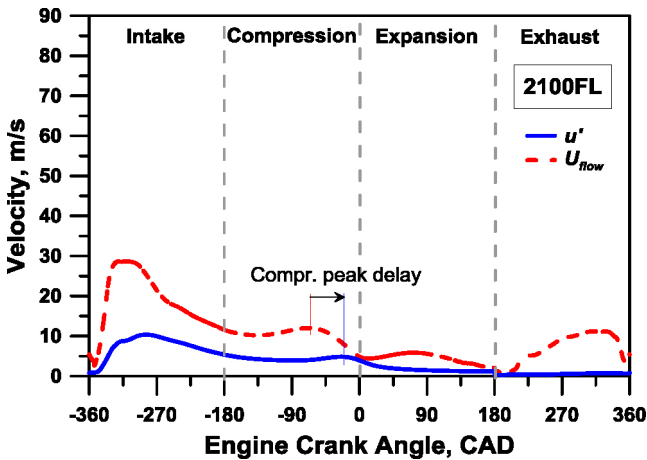


Figure 6. 3D-derived U_{flow} and u' at 2100FL

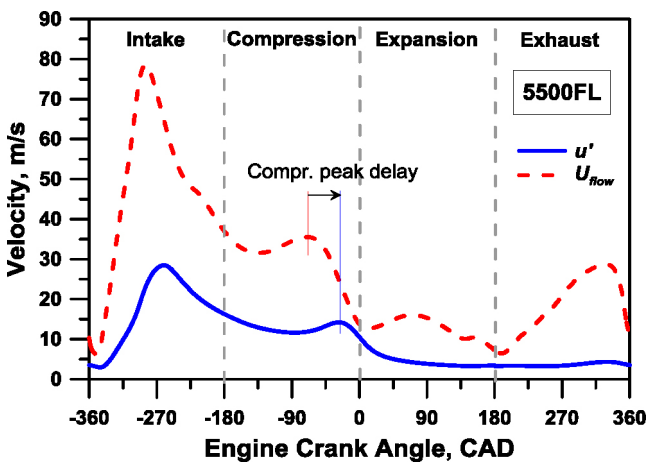


Figure 7. 3D-derived U_{flow} and u' at 5500FL

The above Figures point out that the mean flow velocity (dashed line) presents the peak level in the middle of the intake stroke, where a well-structured tumble vortex establishes, as depicted in Figure 3. With a certain delay, the turbulent intensity (continuous line) increases, as well, because of the energy transfer from mean to turbulent flow field (energy cascade mechanism).

Due to viscous forces, both macro- and micro-scale vortices then start to decay. Along the final phase of the intake stroke and the initial phase of the compression stroke a superimposition of energy transfer and destruction, at both macro- and micro-scale level, determines a gradual decrease in mean flow velocity and turbulent intensity, as well. In the second half of the compression stroke, indeed, a supplementary production of mean flow kinetic energy appears, due to the compression of the tumble vortex. The increased velocity gradients in the cylinder induce instabilities in the tumble vortex, which suddenly collapses, and produces, with a certain angular delay, a speed-up in the turbulent intensity profile. The latter hence presents

a second peak before the compression TDC, as clearly underlined in Figure 6 and Figure 7. During the expansion stroke, mean flow velocity reaches lower levels, and the observed variations are mainly related to the piston motion. During the exhaust stroke, moreover, the exiting flow through the valve causes a certain production of mean flow kinetic energy. On the other hand, the turbulent intensity continuously relaxes along late expansion and exhaust phases.

Depicted mean flow and turbulence trends are qualitatively similar in the two considered operating points, while, as expected, their maximum levels scale according to the engine speed. As shown in Figure 8, a quite different behavior indeed occurs for the integral length scale, resulting in very similar trends and levels for both the considered points, at least up to about 60 CAD after the CTDC. Then more pronounced differences arise, especially once the exhaust valve opens. Nevertheless, the detailed description of the phenomena along the final phase of the expansion stroke and along the exhaust stroke is not critical, since they do not considerably affect the next intake, compression and expansion phases. The above observations represent the phenomenological background for the development of the 0D mean flow and turbulence model discussed in the next section.

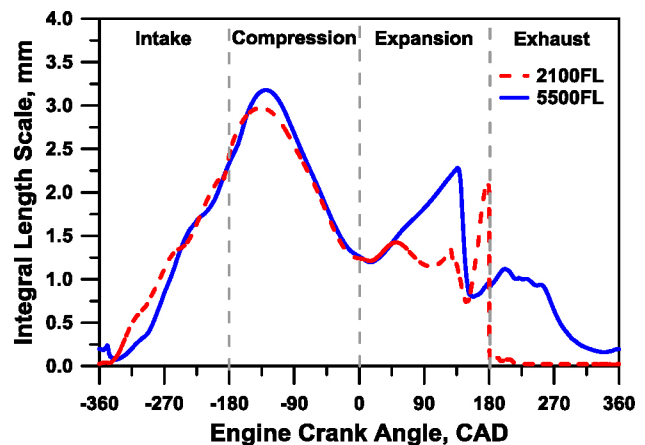


Figure 8. 3D-derived integral length scale at 2100FL and 5500FL

0D MEAN FLOW AND TURBULENCE MODEL DESCRIPTION

As known, 3D models solve mass, energy and momentum equations (along three reference axes), coupled to two additional balance equations for turbulent kinetic energy and its dissipation rate (k - ϵ model). In a 0D modeling approach, indeed, only single-value mass and energy equations are available. Hence any proposed 0D mean flow and turbulence model can only try to mimic mass-averaged profiles, such as the ones reported in Figure 6-Figure 8. Effects exerted by intake-exhaust port layout or combustion chamber design can be only roughly included in a 0D model, which hence requires a certain number of tuning constants. Nevertheless, if well set-up and tuned, it has the potential to properly foresee average mean flow and turbulence variations occurring under widely different engine operating conditions. In this way, it can provide reliable information to the combustion model, to rightly predict the burning rate.