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A 1D/3D Methodology for the Prediction and Calibration of a High Performance Motorcycle SI engine

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Abstract

Nowadays internal combustion engines development is widely supported by 1D and 3D codes. On the other hand, an extensive experimental activity at test bench involves increased development cost and time-to-market of a new engine. For this reason, a numerical methodology capable to provide a reliable estimation of engine performance starting from a reduced set of measured data represents a very promising approach.
In this paper, a hierarchical 1D/3D numerical procedure is proposed with reference to a motorcycle naturally aspirated spark ignition engine to predict its performance even in absence of experimental data.
To this aim, the real engine is geometrically characterized through a reverse engineering process. Different measurements including 3D cylinder geometry, main lengths and diameters of intake /exhaust systems and valve lift profiles, are carried out. Then, a 1D model of the whole engine is realized within the GT-Power™ code, while a 3D model of the sole cylinder is developed within ANSYS Fluent™ environment.
The exchange of data between 1D and 3D models starts with preliminary 3D CFD analyses, performed to evaluate the discharge coefficients of intake and exhaust valves. The latter are passed to 1D model to compute the time-varying boundary conditions at the intake and exhaust head ducts, under motored operation. Multi-cycle 3D CFD analyses are hence carried out to describe the in-cylinder mean and turbulent flow fields in motored conditions. The mass-averaged 3D results are then used to tune the turbulence sub-model included in the 1D engine model.
The last step of the procedure is the computation of the engine performance under fired conditions at full load by means of the 1D simulation. The numerical/experimental comparison of performance parameters demonstrates that the proposed methodology is capable to satisfactory describe the overall engine behavior even in absence of detailed experimental data.

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Keywords: internal combustion engine, 0D/3D modeling, turbulence; combustion.

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Nomenclature	
A_{ref}	Reference area for Discharge Coefficient
C_d	Discharge Coefficient
k	Turbulent kinetic energy
K	Mean flow kinetic energy
L_I	Integral length scale
$\dot{m}$	Mass flow rate
p	Pressure
R	Gas constant
T	Temperature
Greeks	
γ	Specific heat ratio
ε	Specific turbulent kinetic energy dissipation rate
Subscripts	
0	Related to total condition
1	Related to upstream condition
2	Related to downstream condition
is	Referred to isentropic flow
real	Referred to real flow
Acronyms	
0D/1D/3D	Zero-one-three dimensional
BDC	Bottom dead center
CFD	Computational fluid dynamic
MFB_{50}	Crank angle of 50% Mass Fraction Burned
SI	Spark ignition
RANS	Reynolds Averaged Navier Stokes
TDC	Top dead center

1. Introduction

Nowadays, 0D/1D codes are used during the engine development stages to guide and limit the expensive experimental analyses [1], [2], [3].

These codes are generally based on empirical sub-models [4], [5] and they do not often show predictive capability, especially for the simulation of the in-cylinder processes. To overcome this issue, 0D phenomenological models can be successfully used to describe the in-cylinder turbulence evolution and combustion process on a physical basis. In particular, 0D turbulence model allows to synthesize complex 3D phenomena, based on homogeneous (i.e. equally distributed in the space) and isotropic (i.e. without a preferential direction in space) turbulence assumption. Different 0D turbulence models are reported in 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 [6]. Referring to the combustion process, 3D CFD models allow for a better description of combustion phenomenon and noxious emissions but they still require high computational time and storage. On the other hand, phenomenological 0D combustion models show high computational efficiency and allow for an accurate prediction of engine performance. Since many years, two-zone quasi-dimensional models have been proposed in the current literature [7], [8]. Among the different procedures, the "fractal" combustion model appears to be a suitable approach for the description of the turbulent flame propagation phenomenon inside an internal combustion engine [7], [8], [9].

In this work, a hierarchical 1D/3D methodology is employed to predict the performance of a motorcycle naturally aspirated spark ignition (SI) engine. 1D model of the engine is realized within GT-Power^{TM} code. The main characteristics of the tested engine are reported in Table 1.

First, 3D CFD analyses, under steady-state condition, are realized to evaluate the discharge coefficients of intake and exhaust valves. The above data are passed to the 1D engine model, that is used to compute time-varying boundary conditions for the subsequent 3D CFD analyses. 3D results are then utilized to identify the set of tuning constants of the 0D turbulence sub-model. The latter, coupled to a combustion sub-model, is included as ‘user routines’ within GT-Power^{TM} code.

Table 1. Main engine data

Model	4 cylinders, 16 valves
Displacement	999.6 cm^3
Stroke/Bore	52.26 mm / 78 mm
Connecting rod length	111.95 mm
Compression ratio	13:1
Max Brake Power	130.4 kW @ 12500 rpm
Max Brake Torque	115 Nm @ 10000 rpm

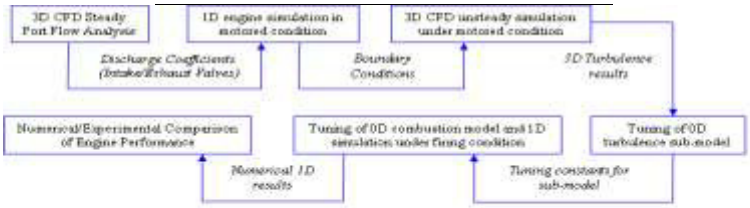

Figure 1. Flowchart diagram of numerical procedure

Finally, 1D calculations, under fired condition, are performed to evaluate the engine performance (brake power/torque, brake specific fuel consumption, air flow rate) at full load. The numerical results are then compared with the experimental data to validate the proposed approach. The whole procedure is synthesized by the flowchart diagram of Figure 1.

2. 1D/3D models description

1D engine model is developed within GT-PowerTM code, based on a one-dimensional description of the flow inside the intake and exhaust pipes and on a zero-dimensional description of the in-cylinder processes. Combustion and turbulence phenomena are reproduced with proper sub-models introduced in code through “user routines”. In particular, the combustion process is simulated by adopting the “fractal” combustion model [10]. An “in-house” 0D turbulence model [11], belonging to the K-k model family [12], [13], is adopted to describe the turbulence phenomenon. Referring to the heat transfer model, a modified Hohenberg correlation is adopted [14], while friction losses are described by the model of Chen-Flynn [15]. Concerning 3D model, a proper mesh is realized within ANSYS FluentTM. The computational domain (Figure 2a) includes the combustion chamber together with the port portions included in the engine head. The mesh (Figure 2b) is mainly composed of tetrahedral and hexahedral cells, with a refined discretization in the near-valve zone. 3D simulations are performed following RANS approach, while the standard k-ε turbulence model and the standard near wall treatment are employed. In order to take into account the near-wall requirements of turbulence model, three layers of cells having 0.2 mm in height are used. Proper constant temperatures on the different wall regions of the combustion chamber are specified. A 2nd order differencing scheme is used for the mass, momentum, energy, turbulence and scalar transport equations.

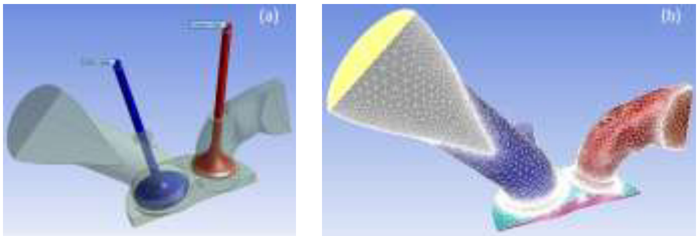

Figure 2. 3D computational domain (a) and mesh (b)

3. Port Flow 3D Analysis set-up and results

3D simulations are performed to evaluate the discharge coefficient for intake and exhaust valves, both in forward and reverse flow conditions. Different 3D port flow analyses, in steady state condition, are realized by varying the valve lift with a step of 1 mm. Numerical results are obtained by imposing a constant pressure drop (equal to 0.1 bar) through the examined valve [16], [17], [18]. The discharge coefficients of intake and exhaust valves are reported in Figure 3a and Figure 3b, respectively. They are defined as:

$$C_d = \frac{\dot{m}_{real}}{\dot{m}_{is}} = \frac{\dot{m}_{real}}{A_{ref}\dfrac{p_{01}}{\sqrt{RT_{01}}}\sqrt{\dfrac{2\gamma}{\gamma-1}\left[\left(\dfrac{p_2}{p_{01}}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_2}{p_{01}}\right)^{\frac{\gamma+1}{\gamma}}\right]}} \quad (1)$$

where $\dot{m}_{real}$, $\dot{m}_{is}$, A_{ref}, represent the actual (computed) and isentropic mass flow rates, and the reference area, respectively. The latter is assumed as the area of the valve head. In Eq. (1), p_{01}, T_{01} are the upstream total pressure and temperature, while p_2 is the downstream static pressure.

The figures depict trends and maximum values qualitatively congruent with the experimental data available in literature [19].

4. Unsteady 3D Analysis: set-up and results

In the case of unsteady 3D analysis, a moving mesh is set-up, where cells are progressively distorted, activated and deactivated to take into account the piston and valve motion. The near-valve cell size is properly chosen to guarantee mesh uniformity during the intake and exhaust strokes. The resulting grid is composed of about 590.000 and 305.000 cells at BDC and TDC, respectively. As said, time-dependent pressure and temperature boundary conditions are derived from the 1D simulations under motored operation. Spatially uniform initial conditions for the first engine cycle are imposed. All 3D analyses start from the expansion phase at TDC and cover a sequence of three full engine cycles. Variable time steps, according to the cycle events, are set to properly take into account dynamic events related to valve opening and closure.

3D results post-processing is performed to derive the time evolution of both mean quantities (mass, pressure, temperature, velocity magnitude) and turbulent ones (turbulence intensity, length scale). As an example, Figure 4 shows the velocity magnitude (Figure 4a) and turbulent kinetic energy (Figure 4b) flow fields at a specified crank angle before the intake valve closure for the considered engine speeds. These pictures refer to a plan passing through the symmetry axis of the cylinder and of the intake valve. The examined engine speeds lead to substantial different in-cylinder flow patterns in terms of both distribution and intensity.

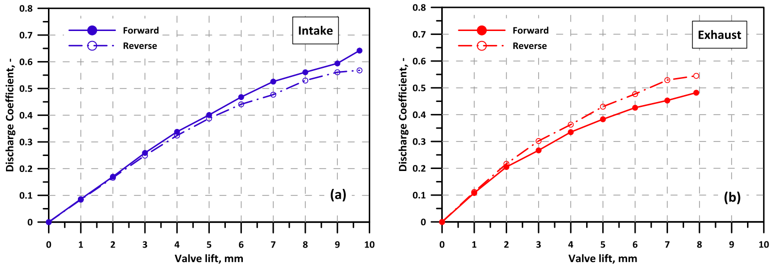

Figure 3. Discharge Coefficient for Intake (a) and Exhaust (b)valves in Forward and Reverse flow condition

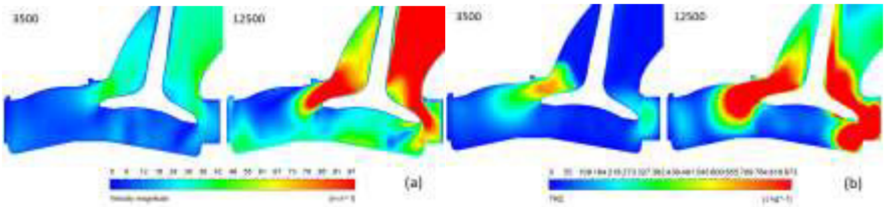

Figure 4. Mean flow velocity (a) and turbulent kinetic energy (b) @420 deg

5. Turbulence 0D sub-model: set-up and results

The 0D K-k turbulence model is tuned against 3D-CFD motored findings at a speed of 3500 rpm. 0D and 3D results are also compared for the highest speed of 12500 rpm. The first step in the tuning procedure, is a proper description of the integral length scale, LI. In the proposed approach, LI is schematized as a sequence of Wiebe-like functions passing through specified control points.

Other tuning constants of K-k model are adjusted in order to fit the 3D-derived mass-averaged turbulence and mean flow results. Through a simple trial and error procedure, a set of optimal constants is selected at 3500 rpm. For the considered cases, the tuning constants are kept unchanged, in order to assess the independency of the proposed turbulence model on the operating conditions.

Figure 5 shows the 0D/3D comparison of integral length scale for the examined operating points. A good accuracy is obtained near to the compression TDC, especially for the tuned case at 3500 rpm. In this way, the reliability of the 0D combustion model is maximized. 3D findings show that LI only slightly depends on the engine operating conditions, especially in the neighborhood of the compression TDC. This confirms the choice to use a unique LI reconstruction for the considered operating points.

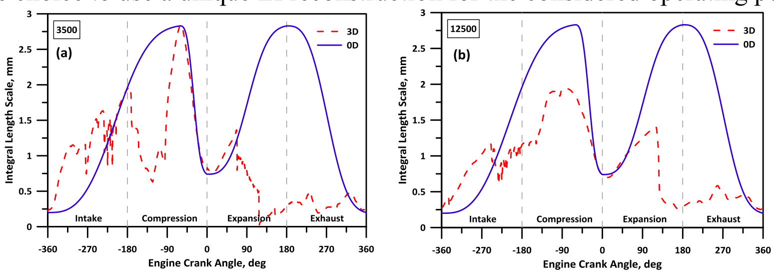

Figure 5. Integral length scale 0D/3D comparison at 3500 rpm (a) and 12500 rpm (b)

As shown in Figure 6a and Figure 7a, the proposed model, once tuned, is able to well forecast the in-cylinder velocity magnitude and turbulence levels, especially during the compression/expansion phases.A certain overestimation of both velocity magnitude and turbulence intensity at 12500 rpm (Figure 6b and Figure 7b) is realized. This overestimation has been considered acceptable around the TDC.

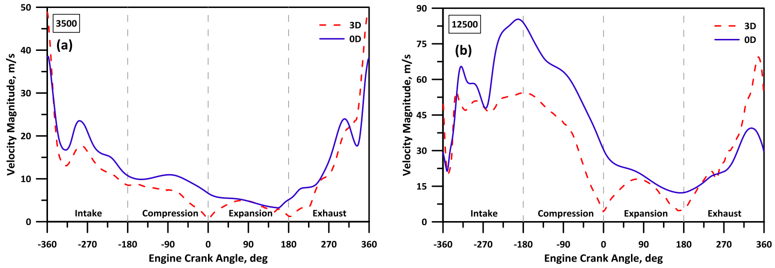

Figure 6. Velocity magnitude 0D/3D comparison at 3500 rpm (a) and 12500 rpm (b)

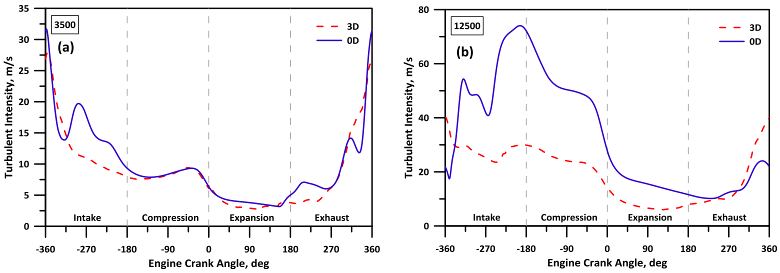

Figure 7. Turbulent intensity 0D/3D comparison at 3500 rpm (a) and 12500 rpm (b)

Referring to Figure 6 and Figure 7, the 0D turbulence model demonstrates to satisfactory describe the energy cascade mechanism from the mean to the turbulent flow, required to provide a reliable estimation of engine performance under fired condition, as discussed in the next section.

6. Combustion 0D sub-model: set-up and results

Fired operation requires the assignment of additional engine control parameters in the 1D model, such as the air-to-fuel ratio and the combustion phasing, MFB50. The air-to-fuel ratio is enriched according to the engine speed to limit the exhaust gas temperature, while the MFB50 phasing is preliminary set at the Maximum Brake Torque (MBT) value. The possibility to work at MBT setting was verified with additional analyses which include knock prediction, following the approach described in [20], [21]. Knock-free operation can be achieved at MBT for this engine at full load. The considered engine is equipped with a short variable length pipe included in the air-box. In the actual operation, its length is adjusted according to the engine speed to maximize the cylinder filling. The setting of the above geometrical datum is not available, and it is assumed fixed in the numerical analyses. Combustion, friction and heat transfer tuning constants are kept unchanged for each analyzed operating condition. The above tuning constants, utilizing the well-known literature approaches, are imposed on the basis of previous analyses performed on engines similar to the examined one. The comparisons of computed brake torque and power with manufacturer data are reported in Figure 8a and Figure 8b, respectively. Manufacturer data never entered the 1D/3D simulation procedure, and they are only here utilized to verify the numerical results. The overall agreement is hence quite satisfactory, especially in the medium/high speed range. Inaccuracies in numerical outcomes mainly regard low engine speeds (below 5000 rpm), where probably the neglected effects of the intake pipe length variations are more important.

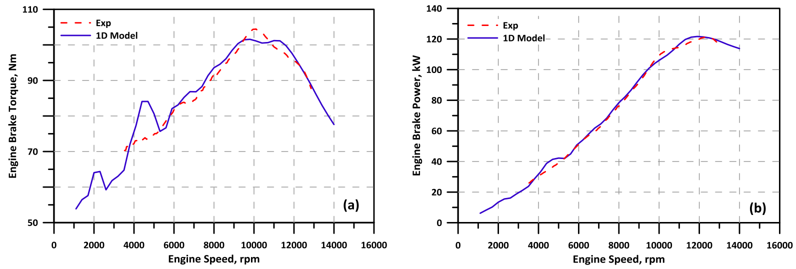

Figure 8. Numerical/Experimental comparison of brake torque (a) and brake power (b) at full load

7. Conclusion

An integrated 1D/3D numerical methodology is proposed to describe the overall performance, at full load operation, of a motorcycle naturally aspirated SI engine. The engine model, developed in GT-PowerTM, is integrated with “user routines” for turbulence/combustion description. Turbulence model is tuned with reference to 3D CFD simulations of the flow field inside the cylinder under motored operation. 3D analyses are also performed to evaluate the discharge coefficients of intake and exhaust valves, to be assigned in the 1D simulations. The fractal combustion model is employed to estimate the engine performance under fired operation. The engine behavior is well predicted, without requiring a case-depending tuning, in terms of brake torque and power, especially in the medium-high speed range. Summarizing, the proposed hierarchical 1D/3D approach proved to be a useful tool to predict the engine performance during its early development phase, even in absence of experimental data.

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Biography

L. Teodosio is a PhD Student at University of Naples “Federico II”, Italy. He is dealing with 1D-3D CFD analyses in automotive engines.