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Development of a lumped model for the characterisation of the intake phase in spark-ignition internal combustion engines

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Abstract

The present work aims to develop a control-oriented lumped model to investigate the fluid dynamic behaviour of multi-valve spark-ignition engines (ICEs). Specifically, the attention has been focused on the intake phase and in-cylinder air charge estimation. To this purpose, a spark-ignition engine has been characterised at a flow rig in terms of flow coefficients. The experimental data have been used to define the fluid dynamic behaviour of the different intake system components and to calibrate and validate the proposed model that has been developed in Matlab/Simulink environment. Furthermore, in order to evaluate the capability of the zero-dimensional code and to estimate the instantaneous in-cylinder mass flow in different operating conditions, the numerical data have been compared to the results of a one-dimensional commercial software.
The comparison between numerical and experimental data shows a good agreement. The investigation highlights that the proposed control-oriented lumped model represents a useful and simple tool to evaluate the engine breathability and to define the proper valve timing.

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Keywords: Internal Combustion engine; intake system; lumped model

1. Introduction

The design and the optimisation of highly efficient internal combustion engines (ICEs) require a thorough

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understanding of the different processes that take place within the modern automotive systems [1-2]. Specifically, an accurate knowledge and control of the intake phase and the air fuel mixing is fundamental to respect the more and more severe regulations on the exhaust gas, to improve engine efficiencies, and to reduce fuel consumptions.

To this purpose, several methodologies can be adopted, based both on experimental and numerical approaches [3-5]. Specifically, mathematical models are largely used owing to the possibility to investigate different geometries and operating conditions and to reduce significantly the time and cost of the research [6-8]. The models can be divided in three main groups: zero-dimensional or lumped, one-dimensional, and three-dimensional models [9].

Zero-dimensional codes are based on thermodynamic and phenomenological laws and are characterised by the highest simplicity and lowest computational effort. At the same time, 0D codes guarantee high accuracy due to the possibility to define coupled sub-models able to properly simulate complex systems like the intake apparatus of modern internal combustion engines (ICEs) [6]. For this reason, the development of simple control-oriented numerical tools appears of great interest in order to estimate the instantaneous mass flow rate entering the combustion chamber and to define the proper intake and exhaust valves timing [10-13].

This paper aims at developing a control-oriented lumped model to characterise the dynamic behaviour of the intake process in multi-valve internal combustion engines. A phenomenological code has been proposed considering the continuity and momentum equations.

The main components of the intake system have been modelled adopting the geometric and fluid dynamic characteristics. In particular, an experimental analysis has been performed at the flow rig in order to define the flow coefficient of the intake system. Particular attention has been focused on intake duct, plenum, throttle valve, engine head, and intake and exhaust valves. Furthermore, a comparison with a one-dimensional commercial code has been presented.

2. Numerical model

An unsteady lumped model has been developed in Matlab/Simulink environment in order to characterise the intake system of an innovative variable valve timing (VVT) spark-ignition engine, whose main characteristics are listed in Table 1. The different components of the intake system have been modelled: the manifold, the intake and exhaust ducts, the cylinders, the intake and exhaust valves, the plenum and the throttle valve [13-14]. The zero-dimensional model acquires the valve lifts laws, that are a function of the engine speed and load. The simplified scheme of the engine is shown in Figure 1.

The modelling of the different components has been performed taking into account the inertia phenomena and the mass accumulation capability of the intake system elements [8,12] in order to correctly estimate the fluid dynamic efficiency and, therefore, the overall performance of the engine. In particular, inertial effects are due to the inversion of the driving force (the pressure difference within the component), that may happen during the final part of the intake phase. In addition, each component is able to accumulate mass due to the periodic behaviour of the engine.

To this purpose, two main elements have been defined: the pure "inertial" component (i.e. pipe), characterised by the inertial effects, and the pure "capacity" component (i.e. plenum), capable to accumulate and release mass.

The inertial component is a connecting element between two reservoirs characterised by p_{in} and p_{out} pressure levels.

Table 1: Main engine characteristics

Engine	Four-stroke
Ignition method	Spark-ignition
Compression ratio, r_c	10
Number of cylinders, N_c	2
Number of intake valve per cylinder, $N_{v,i}$	2
Number of exhaust valve per cylinder, $N_{v,e}$	2
Stroke/Bore, L/B	1.068
Intake valve diameter/Bore, D_i/B	0.373
Throttle diameter/Bore, D/B	0.745

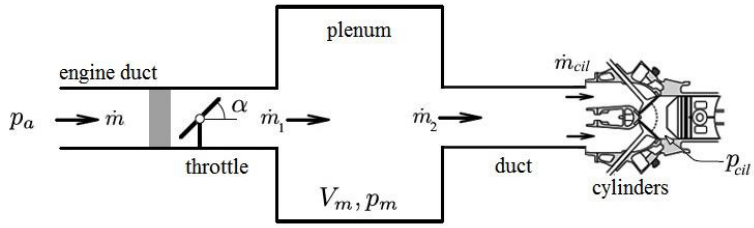


Figure 1.Scheme of the intake system.

In order to take into account the fluid inertia, the mass flow rate has been evaluated from the momentum equation:

$$\dot{m} = A\rho\left(c_{in} + \int_0^t \frac{p_{in} - p_{out}}{\rho l}\, dt\right) \tag{1}$$

Where c_{in} is the velocity at the pipe inlet, A is the cross section, l is the pipe length.
On the other hand, the capacity component is an ideal reservoir, whose pressure varies according to the following equation:

$$\frac{\partial p_m}{\partial t} = \frac{R^* T_{\mathrm{int}}}{V}\left(\dot{m}_{in} - \dot{m}_{out}\right) \tag{2}$$

Where T_{int} is the temperature of the intake system, that is assumed to be constant; $\dot{m}_{in}$ and $\dot{m}_{out}$ are the inlet and outlet mass flow rate respectively, V is the element volume.
It is worthy to notice that each component presents both inertial and capacity characteristics that are considered in the highly dynamic operation of the engine.
As an example, the description of the sub-model corresponding to the duct that connects the plenum to the cylinder is illustrated. A similar approach is used for the other components.
For the aforementioned sub-model it is necessary to take into account the inertial effects due to variations in the cylinder pressure and the possible inversion of the flow due to the increase in the pressure within the combustion chamber during the compression phase.
The pressure downstream of the duct has been evaluated as the pressure of the cylinder plus the pressure drop through the intake valve, which is estimated adopting the experimental flow coefficient:

$$p_{out} = p_{cyl} \pm \Delta p_{valve} \tag{3}$$

where the sign $\pm$ depends on whether the flow is incoming or outgoing from the cylinder (backflow).
The pressure drop through the valve is based on the following equation

$$\Delta p_{valve} = \frac{1}{2}\rho_{curtain}\left[\left(\frac{c_{curtain}}{C_f}\right)^2 - c_0^2\right] \tag{4}$$

that is valid for direct flow to the cylinder and depends on the valve lift.
On the other hand, when backflow occurs:

$$\Delta p_{valve} = \frac{1}{2}\rho_{curtain}\left[c_0^2 - \left(\frac{c_{curtain}}{C_f}\right)^2\right] \tag{5}$$

$c_{curtain}$ is the gas velocity immediately downstream of the valve and can be calculated as follows

$$c_{curtain} = \frac{(\rho A c)_{duct}}{(\rho A)_{curtain}} \tag{6}$$

For the model calibration and validation, pressure losses and mass flow rates have been estimated and compared to the corresponding values registered in a parallel experimental campaign. Furthermore, a 1D commercial code has been adopted to evaluate the capability of the proposed lumped model to characterise the instantaneous intake system behaviour as a function of the crank angle. More details are provided in the next sections.

2.1. 1D Model

The 1D numerical analysis has been carried out adopting a commercial code. The software considers the flow in the pipes as one-dimensional. As a consequence, the pressures, the velocities, and the other parameters represent mean values over the cross-section of the pipes. The model is based on mass and energy balances. Specifically, the internal energy change within the cylinder is obtained as a function of the piston work, fuel energy input, wall heat losses and the enthalpy flow due to blowby, according to the following equation:

$$\frac{d(m_c u)}{d\alpha} = -p_{cyl}\frac{dV}{d\alpha} + \frac{dE_f}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - \dot{m}_b h_b \tag{7}$$

where:
- m_c is the mass in the cylinder;
- u is the specific internal energy;
- α is the crank angle;
- V_c is the cylinder volume;
- E_f is the energy fuel;
- Q_w is the wall heat loss;
- h_b is the specific blowby enthalpy;
- $\dot{m}_b$ is the blowby mass flow.

The one-dimensional code has been adopted to investigate the behaviour of the complete high performance spark-ignition engine and to validate the proposed lumped model during real operating conditions. Specifically, the two cylinders with the whole intake and exhaust systems have been studied, and the real timings of the intake and exhaust valves have been imposed. Furthermore, the fuel injection and the combustion process have been considered and the Vibe model has been adopted to evaluate the heat release characteristics.

The engine performances have been expressed in terms of percentages of the maximum torque C^*, cylinder pressure $p_{cyl,N}{}^*$, and mass flow rate $\dot{m}_a{}^*$:

$$C^* = \frac{C}{C_{\max}} 100 \tag{8}$$

$$p_{cyl,N}{}^* = \left(\frac{p_{cyl}}{p_{cyl,\max}}\right)_N 100 \tag{9}$$

$$\dot{m}_a{}^* = \frac{\dot{m}_a}{\dot{m}_{a,\max}} 100 \tag{10}$$

where:
- C is the engine torque;
- C_{max} is the maximum engine torque;
- p_{cyl} is the cylinder pressure at a given engine speed;
- $p_{cyl,max}$ is the maximum cylinder pressure for the given engine speed;
- $\dot{m}_a$ is the mass flow rate;
- $\dot{m}_{a,\max}$ is the maximum mass flow rate.

3. Experimental investigation

An experimental investigation has been performed at a steady flow rig in order to define the fluid dynamic characteristics of the engine and to calibrate the numerical models. The flow rig enables air to be forced through the intake system by means of a blower while the valve lifts are fixed to selected values. A by-pass valve permits to impose the pressure drop between the ambient and the combustion chamber. Temperature and pressure transducers are used to characterise the conditions of the ambient and inside the cylinder, while a laminar flow meter system is adopted to measure the global mass flow rate.
The flow coefficients have been used to define the global engine permeability. Specifically, the dimensionless coefficients are defined as the ratio of the actual mass flow rate $\dot{m}_a$ to reference mass flow rate $\dot{m}_r$ [15-17]:

$$C_f = \frac{\dot{m}_a}{\dot{m}_r} \tag{11}$$

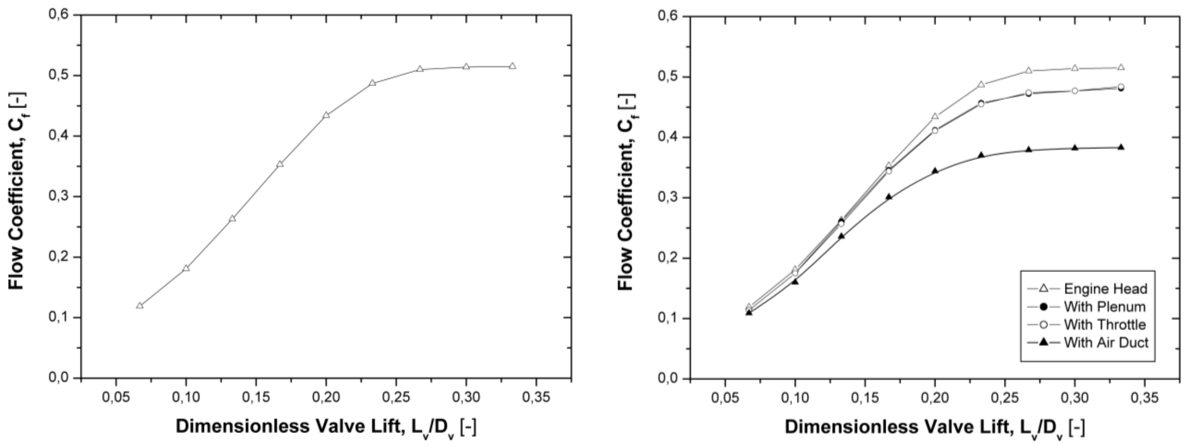
Measurements have been taken for a fixed ambient-cylinder pressure drop (8.3 kPa), while the dimensionless valve lifts (L_v/D_v) are set in the 0.067 ÷ 0.333 interval (Table 2). More details on the experimental apparatus and the dimensionless coefficients are given in literature [18-19].

Table 2: Measuring conditions.

Pressure drop, Δp	8.3 kPa
Dimensionless valve lift, L_v/D_v	0.067 ÷ 0.333
Dimensionless valve step, $\Delta L_v/D_v$	0.033
Throttle angle, F	90°

4. Results

Figure 2a shows the engine head breathability in terms of flow coefficients as a function of the dimensionless valve lift (L_v/D_v). The analysis highlights a progressive increase in the dimensionless parameter with the valve lift, due to the raise in the mass flow rate entering the cylinder.
Different flow regimes are observed, according to the literature [15,20]. Particularly, the flow remains attached to the valve seat and head due to the high viscous phenomena at low valve lift. Then a flow separation occurs when the lift rises, first, at the valve head and, successively, at the valve seat, and a plateau (C_f ~ 0.51) is reached when L_v/D_v is larger than 0.266. In this case, the dimensions of the intake ports and valve stems define the minimum flow area and further increases in the valve lift do not influence the mass flow rate entering the combustion chamber.



a) b)

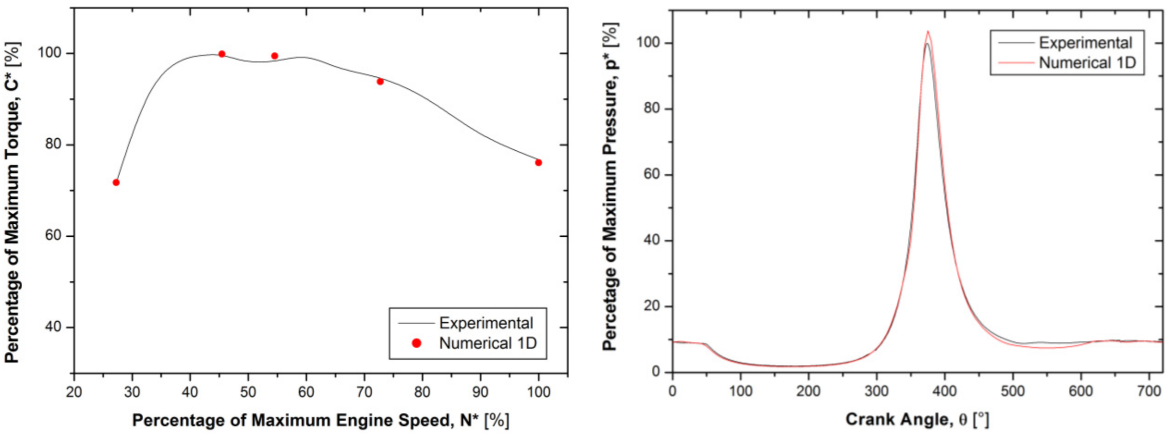
Figure 2. Influence of the dimensionless valve lift on the engine head flow coefficients (a).Effect of the different components on the intake system breathability (b).

The analysis has been repeated to investigate the effect of the different components of the intake system on the global engine permeability and to calibrate the zero- and one-dimensional models. Figure 2b illustrates a progressive decrease in the flow coefficients when the different components are added to the engine head. Specifically, at higher valve lifts, the dimensionless parameter moves from 0.51 to 0.48 when the intake plenum is considered, and it reduces to 0.37 if the intake duct is also present. On the other hand, similar values ($C_f \sim 0.12$) are registered at low lifts ($L_v/D_v = 0.067$). It is noteworthy to notice that the throttle valve has always a negligible effect on the engine fluid dynamic efficiency in the wide-open throttle (WOT) configuration.

The previous experimental results have been adopted to validate the zero- and one-dimensional models in steady state conditions. It is noted a good agreement between the experimental and numerical data. For example, the percentage differences for mass flow rates are lower than 2.7% while the difference in pressure drops are lower than 0.1 kPa. Specifically, when the lumped model is adopted, the maximum pressure difference for the intake plenum and engine duct is equal to 100 Pa whereas the value reduces to 70 Pa for the engine head.

Furthermore, information provided by the engine manufacturer have been used for the validation of the 1D numerical results in real operating conditions. To this purpose, Fig. 3a compares calculated and measured data in terms of percentage of the maximum torque. The figure illustrates the capability of the one-dimensional model to characterise the performances of the two-cylinder spark-ignition engine with good accuracy. The torque is well defined for all the engine speeds, and the maximum values with the corresponding engine regimes are accurately defined. In particular, the differences between reference values and 1D simulations are always lower than 2.5% when the full load condition is analysed. The figure demonstrates that the maximum torque is registered when $N^* =$ 60% and high values are recorded for all the investigated engine regimes (torque maintains always larger than 71% of the maximum torque).

The one-dimensional code permits also to characterise in detail the engine behaviour as a function of the crank angle. As an example, Figure 3b compares the experimental and numerical pressure within the combustion chamber at $N^* = 27\%$. The plot demonstrates a good agreement between measured and calculated pressures for all the crank angles. The pressure behaviour is well reproduced and the maximum values are properly characterised. A mean percentage difference lower than 3.6% has been found.



a) b)

Figure 3. Comparison between experimental and 1D results in terms of percentage of maximum torque (a) and in-cylinder pressure (b).

The assessment of the commercial 1D model has permitted to extend the validation of the proposed lumped model, due to the lack of detailed experimental data concerning the mass flow rate entering the combustion chamber as a function of the crank angle. The possibility to estimate the instantaneous in-cylinder flow accurately, in fact, is fundamental to define the optimal intake strategy and valves timing and, as a consequence, to maximise the engine breathability.

Figure 4 illustrates the comparison between the dimensionless mass flow rate entering the cylinder evaluated for different engine speeds adopting the two models. The plot reveals a general good agreement. The proposed 0D code is able to correctly characterise the intake process and the mass flow rate entering the combustion chamber. In particular, the maximum values of the mass flow rate are well characterised and the inertial effects are properly evaluated. Some differences are registered at the higher engine speeds when the valve lift is reduced from the maximum value. In any case the air mass trapped within the combustion chamber is properly defined. The percentage differences between 0D and 1D data are equal to 7.1% (N* = 27.3%), 7.3% (N* = 54.5%), 4.0% (N* = 72.7%) and 2.1% (N* = 100%).

The analysis reveals that the proposed lumped model is able to characterise properly the fluid dynamic behaviour of the intake system of modern internal combustion engines in different operating conditions. Specifically, the proposed code appears very attractive for an integration in engine control systems owing to its relative simplicity and low computational cost.

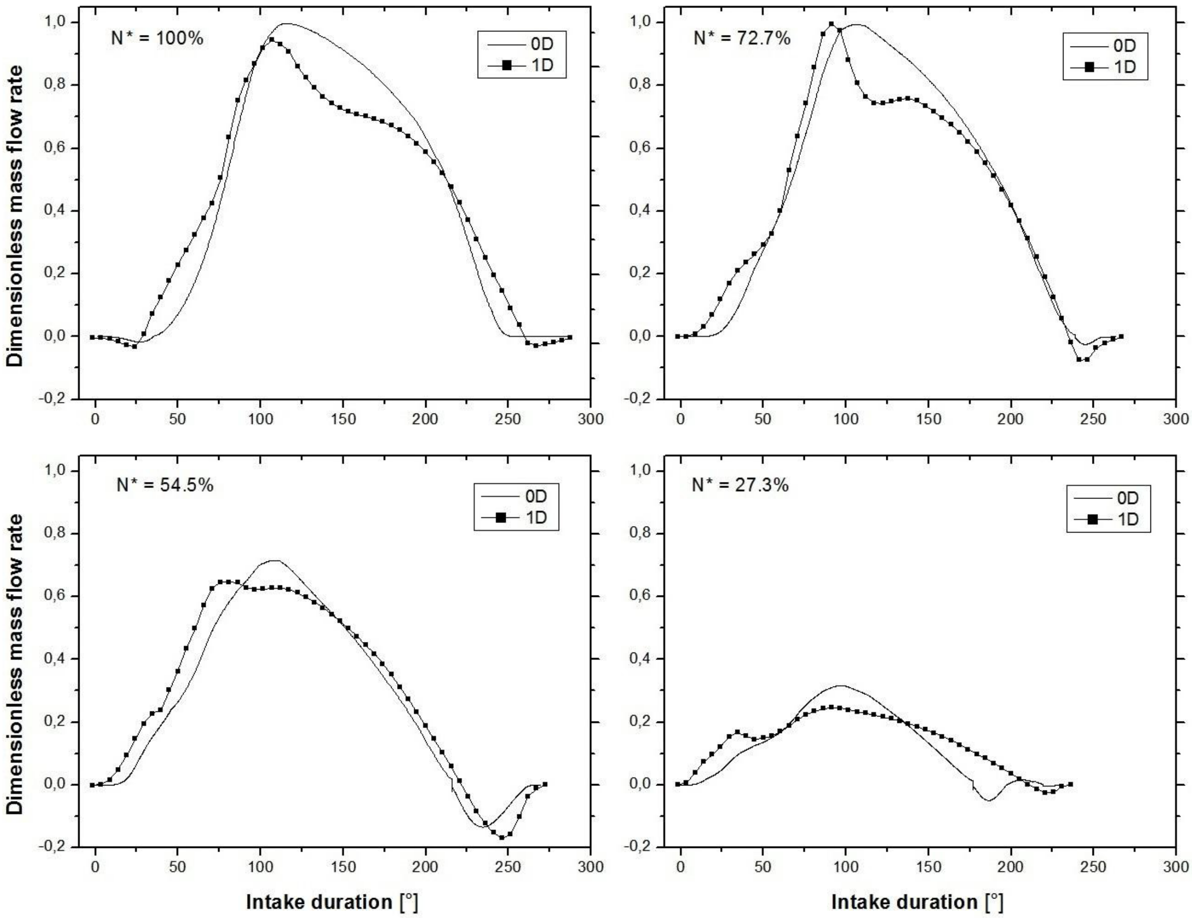


Figure 4. Comparison between 0D and 1D results in terms of dimensionless mass flow rate.

5. Conclusions

The work has focused on the development of a lumped model for the characterisation of the intake performances of spark-ignition internal combustion engines. To this purpose, a production multivalve engine with variable valve timing (VVT) has been analysed.

An experimental campaign has been carried out at a steady flow rig in order to evaluate the flow coefficients of the different components of the intake system and to calibrate the proposed numerical code. Furthermore, a comparison with a commercial one-dimensional model has been performed to investigate the capability of the 0D model to characterise the intake phase properly. Specifically, the mass flow rate as function of the crank angle and the trapped air mass have been studied for different engine speeds.

The analysis showed a good agreement: the maximum values of the mass flow rate are well defined for all the investigated engine speeds and the inertial effects are properly taken into account.

The analysis confirms the capability of the proposed lumped model that represents a useful and simple tool to define the proper valve timing and to estimate the mass flow rate entering the combustion chamber.

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