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Experimental and Numerical Study of Admission and Exhaust Flows in a Spark Ignition Engine

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Abstract

The yield of the process of gas exchange in four stroke internal combustion engines is given by volumetric efficiency, which depends on the dynamics of gases in the exhaust and intake manifolds. These gases are considered as compressible and ideal and the laws of conservation of mass, momentum and energy, 1D-transient, are applied. The numerical results are compared with time resolved measurements performed on a spark ignition single-cylinder engine. For the comparison, pressure in the combustion chamber, pressure of the gas at different locations along the pipelines and wall temperature of intake and exhaust ducts are considered. In order to validate the simulation in a large range of conditions, different engine speeds and lengths of intake and exhaust ducts have been investigated.

Introduction

The increase of price of fuels, derived from petroleum and the introduction of increasingly restrictive environmental norms, have motivated researchers to develop engines, which are more efficient and less harmful to the environment. The efficiency of internal combustion engines depends greatly on the use of inertial and pulsating phenomena, which occur in the intake and exhaust ducts. The optimization of the working conditions of internal combustion engines necessarily requires a deep analysis of the different variables involved in this process. Thus, for example, the intake and exhaust systems can be sized and have their geometrical configurations defined, according to the characteristics that one desires to privilege: fuel consumption, maximum power, torque and volumetric efficiency at low speed. The acoustic design of the intake system is a predominant topic in the study of internal naturally aspirated spark ignition engine [1]. The design of the inlet line requires many tests. However, the industry needs to get the project done quickly and at the lowest feasible cost of production. While testing of engines is widely used for this purpose, these tests are expensive and require the engine prototype in advanced phase of development [2]. The preliminary design of the inlet line is to minimize the pressure loss. However, one can increase the volumetric efficiency taking advantage of the pressure waves, which are generated by the process of gas exchange in the combustion chamber. In recent research tendency is to improve the utilization of the energy contained in fuels.

The study of internal combustion engines requires that the computational models adequately represent the processes of the engine. Therefore, the computational model must be verified using experimental data that can be obtained from a simplified physical model or from an engine functioning in its full complexity. When using a simplified physical model, it is necessary that the phenomena under analysis are adequately reproduced in it.

Specific Objectives

The idea is to create a computational code allowing prediction of the amount of mass that enters through the admission valves. So we can use this code for mathematical optimization of intake process.

The mass flow passing through the valve is governed by inertial and pulsating phenomena, found in the ducts of internal combustion engines. These phenomena are a function of the geometry and operating conditions of the engine. The aim of this paper is to compare experimental pressure data, obtained in a single cylinder engine with the data obtained through an unsteady 1D code. The comparison is done for different engine speeds (800 to 3000 rpm) and intake and exhaust ducts lengths (150 to 1000 mm).

The first part describes the mathematical model of the processes in the combustion chamber and the equations governing the flow in the intake and exhaust ducts. In the second part are illustrated the assembly and equipment used to obtain the experimental data. The results are given in form of graphs comparing the two responses. Finally, the conclusion section is commented and some interesting points of study and some perspectives for the following of this work are given.

Mathematical Models

A good understanding of admission processes is crucial to improve the characteristics of an engine. Actually, computer programs based in 3D-CFD (Computational Fluid Dynamics) techniques can reproduce detailed information of fluid flow. But

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simulation of an entire cycle of fluid flow in cylinders and manifolds, and the geometric complexity of the movement of the piston and valves make this type of simulation having a high computational cost. Normally optimization techniques of volumetric efficiency require many calculation of the objective function. Therefore simulations using 3D engine for this problem would make the analyze time too long [3].

Models of one-dimensional gas flow are important tools in the automotive world, both in design and in optimization. These models have been widely used to reproduce the behavior of the engine in anticipation of noise emission, because they show a balance between accuracy and computation time.

The flow of gas is considered compressible, treated as one-dimensional, non-stationary, non-isentropic and the cross sectional area of the ducts may vary depending on the coordinate space [4]. It is necessary to consider the friction and heat transfer between the flow and the walls of the tube [5]. The variation of specific heat ratio of the fluid, *k*, can be neglected. Applying the equations of conservation of mass, momentum and energy, leads to the following systems of equations:

$$\frac{\partial U}{\partial t} + \frac{\partial F}{\partial z} = f$$

Where:

$$U = \begin{Bmatrix} \rho \\ \rho w \\ \dfrac{P}{k-1} + \dfrac{\rho w^2}{2} \end{Bmatrix}$$

$$F = \begin{Bmatrix} \rho w \\ \rho w^2 + P \\ w\left(\dfrac{kP}{k-1} + \dfrac{\rho w^2}{2}\right) \end{Bmatrix}$$

$$f = \begin{Bmatrix} -\dfrac{\rho w}{A}\dfrac{dA}{dz} \\ -\dfrac{\rho w^2}{A}\dfrac{dA}{dz} - \rho F_f \\ -w\left(\dfrac{kP}{k-1} + \dfrac{\rho w^2}{2}\right)\dfrac{1}{A}\dfrac{dA}{dz} + \rho \dot{q} \end{Bmatrix}$$

Where ρ is density, w is velocity, A is cross-section area, P is pressure, k is the ratio of specific heat, F_f is friction force and $\dot{q}$ is heat transfer per mass. One can prove that these equations form a system of hyperbolic equations and thus have three real eigenvalues, w, $w+c$ and w-c, were c is the speed of sound. When we solve convective dominant problems, as hyperbolic conservation laws, the kind of scheme applied to the convective term becomes quite sensitive to stability and accuracy [6]. A second-order method widely used in solving the governing equations is the Lax-Wendroff two-step scheme or Richtmyer scheme. The intermediate step is done by Lax-Friedrich method.

$$U_{i+1/2}^{n+1/2} = \frac{U_{i+1}^n + U_{i-1}^n}{2} - \frac{\Delta t}{2\Delta z}(F_{i+1}^n - F_i^n) + \frac{\Delta t}{4}(f_{i+1}^n + f_i^n)$$

This method is first order of accuracy and explicit. The second-order is retrieved using the leap-frog method, remaining equal to:

$$U_i^{n+1} = U_i^n - \frac{\Delta t}{\Delta z}\left(F_{i+1/2}^{n+1/2} - F_{i-1/2}^{n+1/2}\right) + \frac{\Delta t}{2}\left(f_{i+1}^{n+1/2} + f_{i-1}^{n+1/2}\right)$$

Another second order method also applied to solve the equations of the gas flow in the ducts is the MacCormarck scheme:

$$U_i^{n+1} = \frac{1}{2}\left[\left(\bar{U}_i + U_i^n\right) - \frac{\Delta t}{\Delta z}(\bar{F}_i - \bar{F}_{i-1}) + \Delta t \bar{f}_i\right]$$

Where the predictor step is:

$$\bar{U}_i = U_i^n - \frac{\Delta t}{\Delta z}(F_{i+1}^n - F_i^n) + \Delta t f_i^n$$

The time step should be chosen to ensure the stability of explicit method, therefore it must satisfy the Courant-Friedrich-Lewis condition:

$$\Delta t \leq \frac{\Delta z}{(|w| + c)}$$

Numerical schemes of high order accuracy for hyperbolic conservation laws must overcome three difficulties: to ensure conservation of the properties, to preserve the high order accuracy in time and space and to control the generation of spurious oscillations in discontinuities [7]. The literature shows the central differential schemes of second order or higher, with constant coefficients exhibiting spurious oscillations near discontinuities such as shock waves and contact surfaces [8]. Shock waves can be found in exhaust ducts and contact surfaces can be generated by the contact of hot and cold gases [4]. One of several techniques used to capture the shock is Davis TVD scheme. This method consists in adding a diffusive term in the methods of second order, in order to damp the oscillations that appear in the presence of discontinuities.

$$[\bar{G}^+(r_i^+) + \bar{G}^-(r_{i+1}^-)]\Delta U_{i+1/2}^n - [\bar{G}^+(r_{i-1}^+) + \bar{G}^-(r_{i+1}^-)]\Delta U_{i-1/2}^n$$

Where G is called artificial viscosity and is calculated as

$$\bar{G}^{\pm}(r_i^{\pm}) = 0{,}5C(\nu)\left[1 - \phi(r_i^{\pm})\right]$$

$$C(\nu) = \begin{cases} \nu(1-\nu) & \nu \leq 0{,}5 \\ 0{,}25 & \nu > 0{,}5 \end{cases}$$

and

$$r_i^+ = \frac{\left[\Delta U_{i-1/2}^n, \Delta U_{i+1/2}^n\right]}{\left[U_{i+1/2}^n, \Delta U_{i+1/2}^n\right]}$$

In the last equation the bracketed terms [. , .] denote the inner product. The limiter ϕ may be calculated by the following equation:

$$\phi(r) = \begin{cases} min(2r, 1), & r > 0 \\ 0, & r \leq 0 \end{cases}$$

For this scheme to be stable, the time step should be reduced by 70% of Courant number.

The boundary conditions, including valves, open closed end, were found using the methods of the characteristics by the procedure presented by El-Rahman et al. [9].

The thermal process of the gas inside the combustion chamber is modeled by zero-dimensional model. This work shows the study in a single cylinder engine without combustion and without fuel injection. Applying the mass and energy balances equations and the ideal gas equation, we arrive at the following system of differential equations:

$$\frac{dm}{d\theta} = \frac{dm_{ad}}{d\theta} + \frac{dm_{ex}}{d\theta}$$

$$\frac{dT}{d\theta} = \frac{1}{mc_v}\left[\frac{\delta Q}{d\theta} - P\frac{dV}{d\theta} + (h_{ad} - u)\frac{dm_{ad}}{d\theta} + (h_{ex} - u)\frac{dm_{ex}}{d\theta}\right]$$

$$\frac{dP}{d\theta} = P\left[\frac{1}{T}\frac{dT}{d\theta} + \frac{1}{m}\frac{dm}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}\right]$$

where P, V, T, m, u, h, θ and Q are the pressure, volume, temperature, mass, internal energy, enthalpy, crankshaft angle and heat transfer, respectively. The indices ad and ex represent the enthalpy or mass flow crossing the boundary of the system through the admission and exhaust valves, respectively. The rate of heat transfer is obtained by Woschni correlation [10], this correlation is the most used today [11]. Through geometric relationships of the engine, we can calculate the volume and its derivative as a function of crankshaft angle.

The equation system above is solved by the fourth-order Runge-Kutta method. The mass flow through the valve is described by the one-dimensional, stationary, compressible, isentropic flow equation and the effects of turbulence are included in real flow through the discharge coefficient Cd, obtained experimentally [12]. The step of crankshaft angle is obtained from the time step of gas flow solution in the intake and exhaust ducts.

Experimental Apparatus

The objective of experimental apparatus is to test the reliability of the model presented above. It is proposed to perform the measurement of gas pressure at certain points of an I.C. engine. We used the engine bench of laboratory CORIA. The engine used for this experiment was an AVL single cylinder engine, spark-ignition and direct injection. Some characteristics of the combustion engine can be seen in table 1.

An input important to the computer code is the valve opening curve. This diagram is obtained through experimental measurements of valve travel using a dial indicator.

AVL 5482 Engine	
Stroke	86 mm
Bore	82 mm
Rod	144 mm
Compression Ratio	8,5

Table 1 - Data combustion engine used in the experiments.

The crank angle was obtained directly through a marking of degrees at the flywheel. The measurement of displacement valves was made at constant intervals of 5 degrees of crankshaft except when closing the valves. This average was inserted into the computer code and intermediate values of measuring points were determined by linear interpolation. Table 2 shows the values of valves open and close obtained experimentally.

Admission valve open (avo)	1º After TDC
Admission valve close (avc)	68º After BDC
Exhaust valve open (evo)	79º Before BDC
Exhaust valve close (evc)	16º After TDC

Table 2 – Results of measures of valves timing.

Attached to the same shaft of the I.C. engine is an electric machine which can operate both as electric motor and brake. In this experience there was no fuel injection, so the electrical machine run the engine at a specified rotation speed. This study without combustion is a preliminary step that allows observing the pressure waves in the exhaust and intake manifolds. It also allows checking the accuracy of the computational code without some complexities that appear with combustion. As this engine does not have a long overlap region valve, it is expected to obtain results in the intake manifold not sensitive to the fact that engine is operating with or without combustion.

The engine is instrumented with an angular position sensor (encoder) AVL with precision of 0.1 degree. The measurement of the rotation speed is also done through this encoder.

The measurement of the pressure in the combustion chamber is performed using a piezoelectric pressure transducer AVL GU22C with linearity error of 56 mbar. The piezoelectric transducer captures only the rate of change of pressure over time, so its signal is integrated. In this integration process will lack a reference, which is estimated in the present work by using the thermodynamics relationships of adiabatic compression. This relationship is applied to the compression phase

between two points specified and the ratio value of specific heats is also provided by AVL.

For measuring pressure inside the duct, the sensor has to be able to capture the dynamic pressure wave. This transducer should have a relatively high sensitivity in order to capture the small pressure fluctuations. So for this study was chosen a piezoresistive transducer Kistler model 4007B with the amplifier Kistler model 4618 with an adapter that allows the cooling of the transducer through a water circuit.

Test	Length of intake duct (mm)	Length of exhaust duct (mm)
1	150	400
2	550	
3	1000	

Table 3 – Ducts lengths configurations.

This transducer is connected also to 14-bit analog-digital convertor and its incertitude is equal to approximately 5 mbar.

In table 3 is shown the configurations of the tests. In the models, it is necessary to know the temperature profile of the wall along the pipeline. To resolve this, it was installed thermocouples in ten positions along intake and exhaust ducts. The intermediate values of temperature were obtained by linear interpolation.

Results and Discussion

In this section, a comparison of the measurements with the simulations will be presented for different ducts lengths and rotation speeds. The acronyms in all the graphs below are: LW2S for Lax-Wendroff two steps, LW2S-FL for Lax-Wendroff two steps with flux limiter and MacCor for MacCormarck method. In addition, in an engine cycle, the range of pressure variations between the process gas exchanges and closed-cycle process being very wide, to facilitate visualization and comparison of the results, on each graph, zooms in intake, exhaust and peak pressure regions has been plotted. In order to make simulation comparison possible, the same mesh size in the intake and exhaust ducts and the CFL number was considered as 0.7. For the experimental results, only the averaging of one hundred consecutive cycles are reported.

In figures 1 to 3 are faced by experimental and numerical data gas pressure inside the cylinder. In these figures, the lengths of ducts are kept the same and the rotation speed is varied in 1000, 2000 and 3000 rpm. We can visualize the maximum pressure increase with engine speed where it can be explained by the increase of mass admitted. We can also see a good agreement between the results numerical/experimental, mainly in the capture phase and frequency. We observed a small deviation in the end of compression for the LW2S-FL. This occurs in cases where this method did follow the experimental curve as seen in figure 3.

In figure 3, 4 and 5 are shown curves of cylinder pressure at constant speed (3000 rpm) and changing the length of the intake duct (150, 550 and 1000 mm). We can visualize increased the high pressure at the end of compression with increasing length of the duct. The same trend can be seen through the simulation model. This is because more mass is entering the cylinder during the intake process, as can be seen by the pressure level at the end of the admission process. We shape of the curves on admission and exhaust processes. We can also visualize the well model follows the change in

Figures 6-8 show the results of intake manifold pressure at 90 mm of inlet valves. The results were

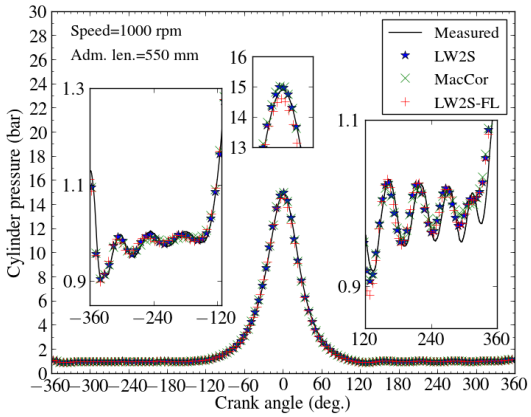

Figure 1 – Pressure cylinder results for motored engine at 1000 rpm and 550 mm admission length duct.

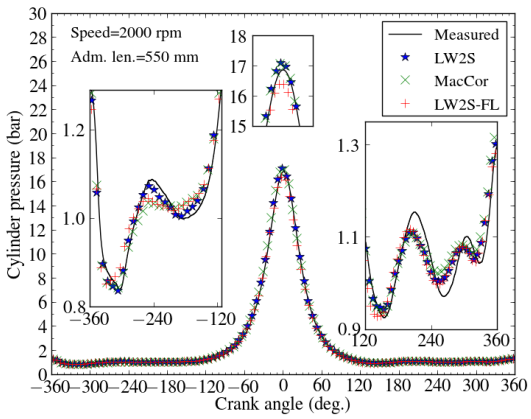

Figure 2 – Pressure cylinder results for motored engine at 2000 rpm and 550 mm admission length duct.

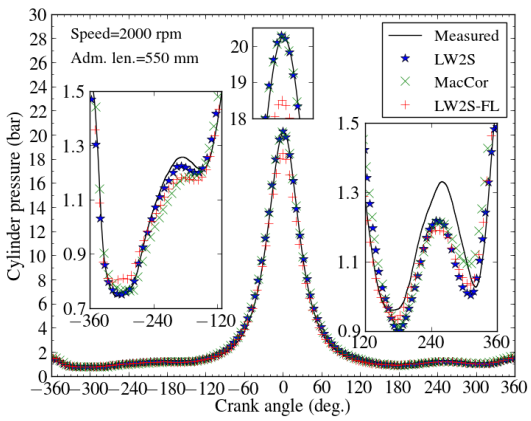

Figure 3 – Pressure cylinder results for motored engine at 3000 rpm and 550 mm admission length duct.

obtained with a fixed length of 550 mm and rotation speed at 1000, 2000 and 3000 rpm. We can remark the variation of pressure wave amplitude which changes with the speed. We have a good agreement when the intake valve is opened. In the case of the inlet valve is close, the correlation occurs only in lower pressure amplitude (1000 to 2000 rpm). Already at 3000 rpm the model presents an underestimation of the pressure level. On the other hand, the frequency of the pressure waves has a good correspondence with the experimental data.

Figures 8-10 show the results of pressure in intake manifold, but now keeping a constant speed and varying

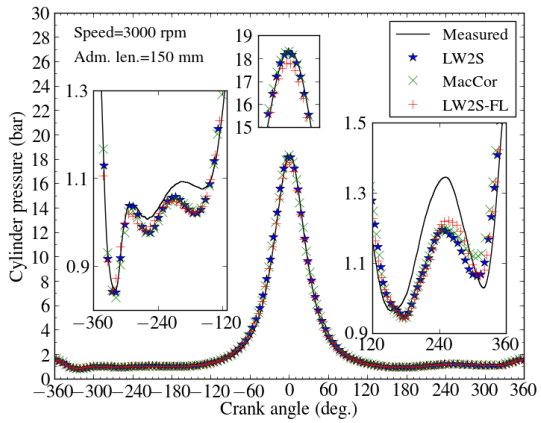

Figure 4 – Pressure cylinder results for motored engine at 3000 rpm and 150 mm admission length duct.

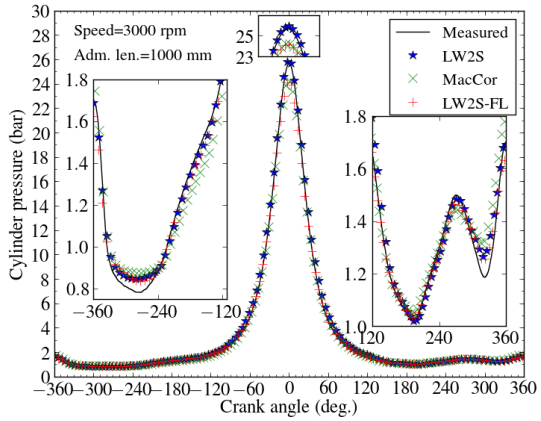

Figure 5 – Pressure cylinder results for motored engine at 3000 rpm and 1000 mm admission length duct.

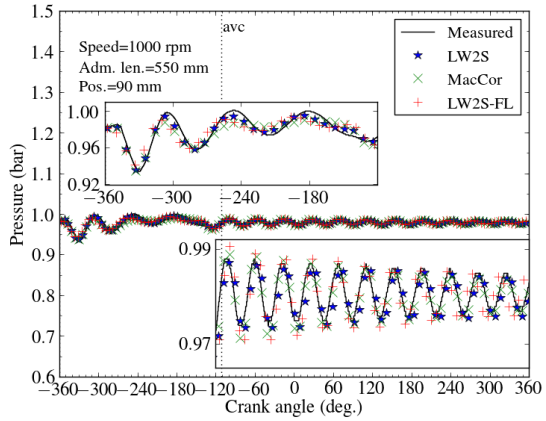

Figure 6 – Pressure duct (position 90 mm from valves) result for motored engine at 1000 rpm and 550 mm admission length duct.

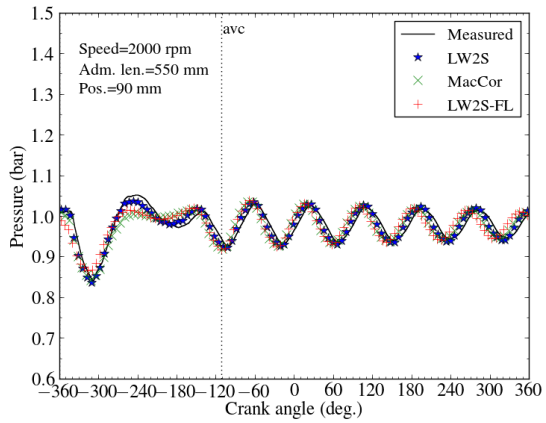

Figure 7 – Pressure duct (position 90 mm from valves) result for motored engine at 2000 rpm and 550 mm admission length duct.

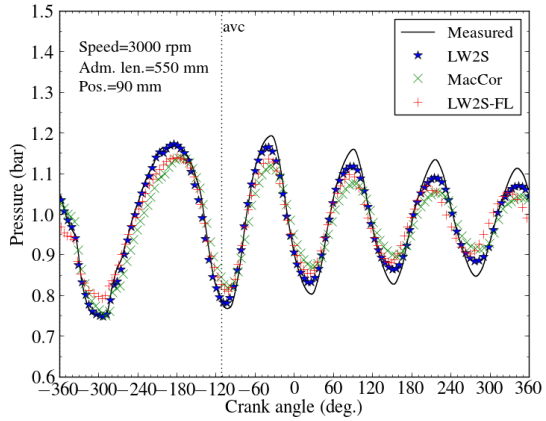

Figure 8 – Pressure duct (position 90 mm from valves) result for motored engine at 3000 rpm and 550 mm admission length duct.

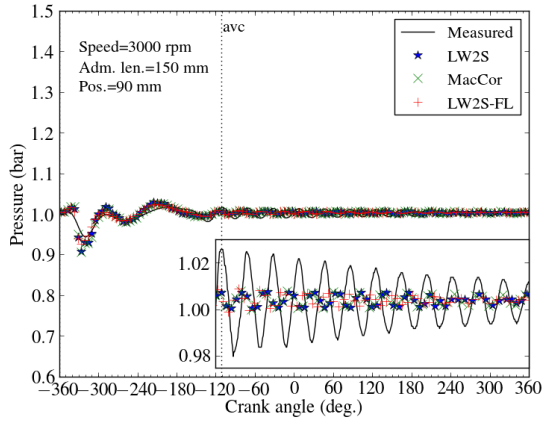

Figure 9 – Pressure duct (position 90 mm from valves) result for motored engine at 3000 rpm and 150 mm admission length duct.

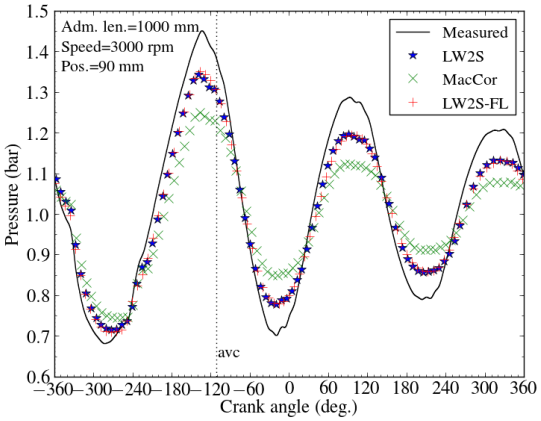

Figure 10 – Pressure duct (position 90 mm from valves) result for motored engine at 3000 rpm and 1000 mm admission length duct.

the length of the intake runner. In this case we can see that the model follow the experimental results. We have an optimal concordance during admission phase. Unfortunately there is an underestimation of magnitude pressure for the region of intake valve closed. This may have happened due to some divergence of parameters of heat transfer and/or viscosity friction in the duct.

Conclusions

The main objective of this study was to compare results obtained by a motored engine with a 1-D gas flow model. The 1-D model was solved by numerical methods for second-order well known in the literature of I.C. engines. The results of pressure in the combustion chamber and the ducts correspond well to physics found in conventional engines. In general the results of the numerical 1-D model were able to reproduce the pressure waves in ducts. In some regions there was an underestimation of the pressure, but the results of frequency and phase is positive. Note also that the 1-D model was able to capture well the pressure fluctuation even by changing the rotational speed and length of the intake duct.

For this article, some parameters were estimated using data from the literature for other I.C engines. For the sequence of work, these parameters will be studied in more detail and it is estimated that will be an improvement of the results presented here. This is a preliminary study without combustion, since the goal was to reduce the number of parameters required to know before. But we intend to extend this study to cases occurring combustion and thus closer to the real case.

With a numerical model experimentally validated, it is possible to analyze the influence of parameters related to the geometric configuration of the admission and exhaust systems (transversal section area, the variation of this area along the duct, length of the ducts, etc.) on parameters such as the volumetric efficiency and the fraction of residual gases. This model will also allow an optimization of the valves synchronism and the geometry of the admission and exhaust systems.

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