Evaluation on the influence of piping geometry and valve opening time on an internal combustion engine.

Tiago A. C. Resende [1], Júlio C. C. Campos [2], José A. Silva [3], Gustavo R. Souza [4], Marco A. da C. Alves [5], Alexsander P. Rezende [6]

[1] tiagoalceuc@yahoo.com.br. Federal Center for Technological Education of Minas Gerais CEFET/ Department of Computing and Mechanics. [2] juliomcampos20@gmail.com. Federal University of Viçosa UFV/Department of Production and Mechanical Engineering. [3] jant@ufsj.edu.br. Federal University of São João del-Rei UFSJ/Department of Fluid and Thermal Sciences. [4] souzagr@ufsj.edu.br. Federal University of São João del-Rei UFSJ/Department of Fluid and Thermal Sciences. [5] marco.alves@engenharia.ufjf.br. Federal University of Juiz de Fora UFSJ/Department of Production and Mechanical Engineering. [6] alexsanderprezende@hotmail.com. Federal University of São João del-Rei UFSJ/Department of Fluid and Thermal Sciences

ABSTRACT

This research aims to find the best camshaft profile suitable for a single cylinder internal combustion engine aiming at its maximum volumetric efficiency. The methodology employed was an iterative search of the best length and diameter of inlet and exhaust pipes using Lotus Engine Simulation Software. The behavior of the high-pressure pulses due to the acoustic and aerodynamic phenomena of the intake manifold, together with the time of opening and closing of the valves, was considered. A numerical model was validated when compared to the experimental model using a Robin Subaru EH 17-2 engine. The results obtained when comparing the third with the last simulation showed that there was an increase of 11.5% in the maximum power and 6.3% in the torque. There were gains in the specific fuel consumption in almost every rotation regime. In relation to the volumetric efficiency, an improvement was made in its value for the new configuration of opening and closing of the valves and geometry of the pipes. It is concluded that the obtained arrangement of the pipes ensured that the frequency and amplitude of the pressure waves were synchronized to the dynamics of opening and closing the valves to stimulate the filling of the cylinders.

Keywords: Intake system, Exhaust, Volumetric efficiency, Lotus Engine Simulation, Intake manifold.

RESUMO

O objetivo da pesquisa é encontrar qual o melhor perfil do comando de válvula adequado para um motor de combustão interna monocilíndrico visando sua máxima eficiência volumétrica. A metodologia empregada foi uma busca iterativa do melhor comprimento e diâmetro das tubulações de admissão e de escape utilizando o Software Lotus Engine Simulation. O comportamento dos pulsos de alta pressão devido aos fenômenos acústicos e aerodinâmicos do coletor de admissão, em conjunto com o tempo de abertura e fechamento das válvulas, foi considerado. O modelo numérico foi validado ao comparar com o modelo experimental utilizando um motor Robin Subaru EH 17-2. Os resultados alcançados ao comparar a terceira com a última simulação constataram-se que houve um aumento de 11,5% na potência máxima e 6,3% no torque. Ocorreram ganhos no consumo específico de combustível em quase todo regime de rotações. Em relação à eficiência volumétrica incidiu uma melhora em seu valor para a nova configuração de abertura e fechamento das válvulas e geometria das tubulações. Conclui-se que o arranjo obtido das tubulações garantiu que a frequência e a amplitude das ondas de pressão ficaram sincronizadas à dinâmica de abertura e fechamento das válvulas para estimular o enchimento dos cilindros.

Palavra-chave: *Sistema de admissão, Exaustão, Eficiência volumétrica, Lotus Engine Simulation, Coletor de admissão.*

1 Introduction

The importance of some parameters of the internal combustion engine, such as efficiency, power, consumption and emissions of pollutants, are directly connected to the volumetric efficiency (ANDREATTA, 2016), which in turn can vary as the behavior of components responsible for the admission changes of air-fuel. One of the objectives of the new internal combustion engine designs is to reduce the consumption and generation of gaseous pollutants (CHALET et al.,2011; DEB et al., 2014) by means of improvements in thermal efficiency and by searching for engines with the piston, valves and ducts with higher capacities of suction (OCH et al,.2016).

Volumetric efficiency is an important variable because, from this value, the calculations performed by the engine control unit estimate the correct amount of fuel to be injected, reducing fuel consumption and engine emissions (MONIR et al,.2015).

In the performance of the valves, the behavior of the valve defines the kinematics of the behavior of the fluid flowing into the cylinder, however, several specifications must be remembered. It is important to ensure that the timing for the opening and closing of the valves is ideal so that the filling of the engine by the intake gases is done in the necessary proportion (CHALLEN, 1999).

A significant improvement in engine performance can be achieved by computational means using simulation tools, so it is not possible nowadays to develop new engines without the constant use of these modeling tools. Unnecessary costs with experimentation are reduced and inefficient engine configurations can be eliminated (ALBRECHT, 2005; GALINDO et al., 2010).

This way, it is emphasized the need to study the components that affect the volumetric efficiency (TORREGROSA et al,.2011), for example, intake and exhaust system. Therefore, the proposal of this work is the computational simulation of a single-cylinder engine, aspirated and fueled with gasoline, using the free software Lotus Engine Simulation (LES), to analyze the behavior of the intake gases with the variation of the opening and closing angles of the valves.

1.1 Influence of intake and exhaust duct geometry

The function of the intake and exhaust ducts is to channel the gases entering and exiting the cylinder. Its ideal dimension has the function of optimizing the volumetric efficiency with the minimum reduction of pressure, that is, reducing to the maximum the losses by friction of the flow. In this intention it must guarantee the homogenization of the gases in the entrance of all the cylinders satisfying the characteristics of each engine, especially the pulsating effects on the pipes (PROVASE ,2014).

Several acoustic and aerodynamic phenomena occur in the intake manifold, so the engine performance in conjunction with the opening and closing time of the valves is strongly influenced by these phenomena. The size of the intake manifold can optimize the high-pressure pulses in the intake system by raising the volumetric efficiency (MEZHER., 2013). However, for certain ranges of engine speeds, higher pressure points will occur in the manifold near the intake valve at the time of opening, thus ensuring the greatest air supply to the cylinders (COSTA, 2014).

Atmospheric air undergoes an acceleration when the inlet valve opens, inversely when it closes, abrupt restriction to the flow occurs, generating a sudden stop of gases. The abrupt deceleration forms a high-pressure area, causing a pressure wave that runs the inverse path to the tubing, which is reflected back to the valves again. The reflection of the high-pressure wave occurs in the plenum, and a part can also reflect in the acceleration butterfly or any discontinuity in the intake pipe (DOMSCHKE, 1963).

This pressure wave can be called resonance, a phenomenon that consists of the fact that the column of air contained in a tube vibrates at a frequency inversely proportional to the length. This phenomenon is generated by the abrupt closing of the inlet valve, through which the gases were flowing with a negative pressure, due to the suction produced by the downward movement of the piston. The sudden increase in pressure, due to the closing of the intake valve, causes a resonance that runs through the tubing in the opposite direction, reflecting and returning with the same signal to the valve. This reflected wave, when finding the open admission valve increases the volumetric efficiency (DOMSCHKE, 1963).

However, one way to improve the volumetric efficiency is to adjust the length of the tubes so that the high-pressure waves find the intake valve open at its return (ENGELMAN, 1973).

This shows that the geometry of the ducts should change in large proportions as the rotation varies. As in most modern engines, they do not have a variable geometry manifold, the adjusted dimensions of the intake ducts have their main benefit in a very limited speed range.

2 Validation of the model used

An experimental test was performed using the Robin Subaru EH 17-2 single-cylinder four-stroke internal combustion engine with gasoline present in the engine laboratory of the Federal University of São João del-Rei. The experimental results were compared with the same engine present in the Laboratory of Thermal and Hydraulic Sciences of the Federal University of Juiz de Fora by (SILVA, 2017) in order to validate the quantities found.

Figure 1 show the complete model built in the LES software of the bench engine simulated for validation purposes.

Figure 1 – Model built on Lotus Engine Simulation software for the Robin Subaru EH 17-2B Engine
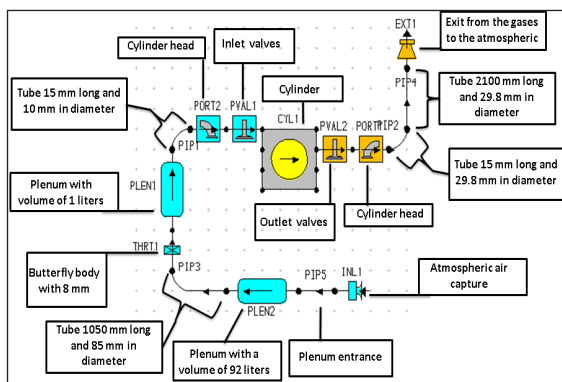
Source: Edited by the author, from LES

After the experimental and numerical simulations, respectively, of the bench engines and in the LES, it was possible to evaluate the results of the volumetric efficiency. Fig. 2 represents this evaluation, where the dashed lines represent the numerical model and the continuous lines the experimental model in the revolutions of 1,300 rpm, 1,715 rpm, 2,180 rpm, 2,855 rpm and 3,120 rpm.

Figure 2 – Representation of the volumetric efficiency of the Robin Subaru EH 17-2 bench engine and the LES after the simulations.
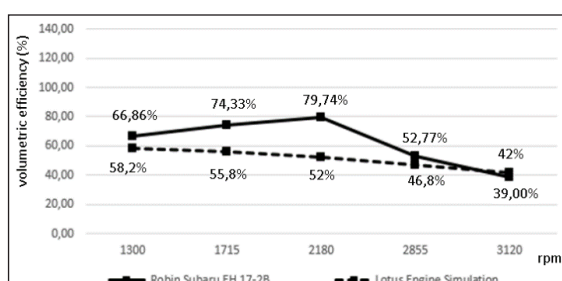
Source: Author

According to (WINTERBONE,1999; CEVIZ,2007) the most influential in improving the volumetric efficiency is the pressure at the inlet of the intake valve, just before its closure. However, the divergence of the volumetric efficiency in the numerical simulation and of the experiment can be explained by the differences of the pressure waves between the two models that act favorably to the volumetric efficiency. In this way, as the construction of the numerical model approximates the experimental model, there are factors of difficult accuracy in which the differences with the experimental model, such as the geometry of the collectors and ducts, are unavoidable.

These differences generate different pulses of pressure for the numerical and experimental model and, with the synchronism with the opening and closing of the intake valve, can generate a lag in the volumetric efficiency between the models (MONIR., 2015)

Other justifications for this behavior are due to the variables not included in the numerical simulation, since some values are stipulated and often do not represent reality. In this context, the load losses in the admission, residual gas fraction, load losses in the pipes and connections or faults in surfaces with protrusions in contact with the gases at high speed (RIBEIRO, 2010) stand out.

According to (ROSTEK, 2017), the factors inherent to simulated engine conditions or their operating situation may also influence the results. An example of this is the correct thickness of the oil film present at all points of friction, which may alter due to changes in the viscosity of the lubricating oil from the irregular frequencies of the oil change or contact with oxidized parts of the engine or of the quality preventive maintenance offered to the engine. These items are, therefore, of difficult measurement to be inserted in the software.

Also, in the numerical simulation, the values of the frequency and amplitude of the vibration of the laboratory engine were not inserted because, although it had ballast, it was not fixed to a base. According to (HARRISON, 2004), the rigidity of the engine structure and its fixation can influence its performance.

3 Methodology

In order to perform the engine analysis in the simulation with the LES software, a four-stroke spark ignition engine was used, with the technical characteristics as per tab.1. However, some indispensable operating parameters were defined by the LES, such as internal clearance, surface materials,

dimensions, combustion model, piston mass and connecting rod, roughness of the inner walls of the inlet and exhaust pipes were considered unchanged, assuming the program default values.

Table 1 – Parameters added in the LES

Parameters	Used value	unity
Atmospheric pressure	750,06	mmHg
Room temperature	20	°C
Ratio fuel/air	Stoichiometric	
Piston clearance	0,05	mm
Piston pin eccentricity	0	mm
Fuel	Gasoline	
Fuel Injection	Electronic injection (multipoint)	
Butterfly Opening	Full load (100%)	
Block Cooling	Water	
Diameter of the piston	87	mm
Piston stroke	84	mm
Compression ratio	9,5	
Volume of the plenum	2,5	liters
Number of valves	Four per cylinder	
Inlet valve diameter	28	mm
Exhaust valve diameter	22,5	mm
Opening the intake valve	8,5	mm
Exhaust valve opening	8	mm

As the behavior of the valves is intrinsic to the geometry of the inlet and exhaust pipes (CERDOUN, 2016), it was necessary to define, through the parametric simulation performed by the LES, the best length and diameter of the ducts for each opening and closing angle of the valves.

In the parametric simulation of the pipes it is necessary to insert in the software the opening and closing angles of the valves, however, at this stage, there have not yet been any optimizations of the valves. So, the solution was to use standard LES values, even though it was not the best for the simulated engine. It was necessary to define the length and diameter interval of the exhaust ducts in which the simulation will occur, according to tab. 2. From this choice, the LES makes a series of combinations of these values by identifying the volumetric efficiency for each exhaust pipe geometry for rotations of one thousand to seven thousand RPM. This occurred successively until it was observed that the results of all the pipes, after each simulation, converged to a single value. The choices of these intervals, tab. 2, were made based on numerous previous simulations.

Table 2 – Length and diameter intervals defined before parametric simulations.

|		Length	Diameter	Unity
Inlet pipe	Interval of iterations	250	34	mm
		450	44	mm
Outlet pipe	Interval of iterations	350	28	mm
		600	37	mm

Once the ducts were sizing, it was possible to begin the parametric simulations of the behavior of the intake and exhaust valves, from the intervals of the tab. 3.

Table 3 – Angle intervals defined before parametric simulations.

		Breaks	Referential
Inlet valve	Opening angle	60°- 0°	Before TDC
	Closing angle	0°- 60°	After the BDC
Outlet valve	Opening angle	60°- 0°	Before TDC
	Closing angle	0 - 60°	After the BDC

Figure 3 shows the entire constructive scheme of the engine before the simulation, in which the dashed rectangles were the values optimized by the software.

Figure 3 – Construction diagram of the whole simulated engine
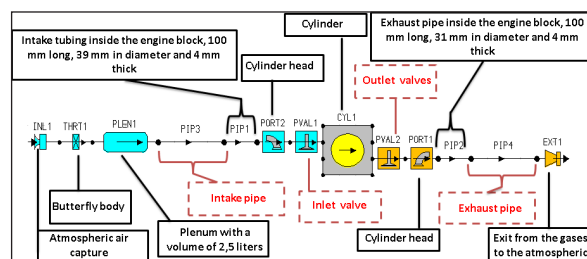
Source: Edited by the author, from LES

4 Results and discussions

4.1 Sizing of pipelines

After initiating the parametric simulations of the exhaust pipes, the LES makes a series of combinations between the length and diameter of the exhaust duct aiming the maximum volumetric efficiency for the revolutions of thousand to seven thousand rpm. FIG. 4 shows the plot of the volumetric efficiency contour with the length of the exhaust pipe (on the horizontal axis) compared to the diameter (vertical axis) for each simulated rotation. There is a significant variation in the volumetric efficiency of the engine with the variation of the exhaust duct geometry or with the modification of the rotation.

To do so, within this entire universe of results, it is necessary to define a single length and diameter for the exhaust pipe. After analysis of all values, it was observed that to maximize volumetric efficiency the length and diameter should be 500 mm and 31 mm, respectively.

Within the values found in the simulations, all the volumetric efficiency curves with the simulated rotations were superimposed, in this way the best result was evidenced, which was 350 mm for the length and 39 mm for the diameter of the intake pipe, according to tab. 4.

For the discussion and analysis of the results obtained concerning the ducts it is convenient to analyze the volumetric yield curves map of the figures 4 and 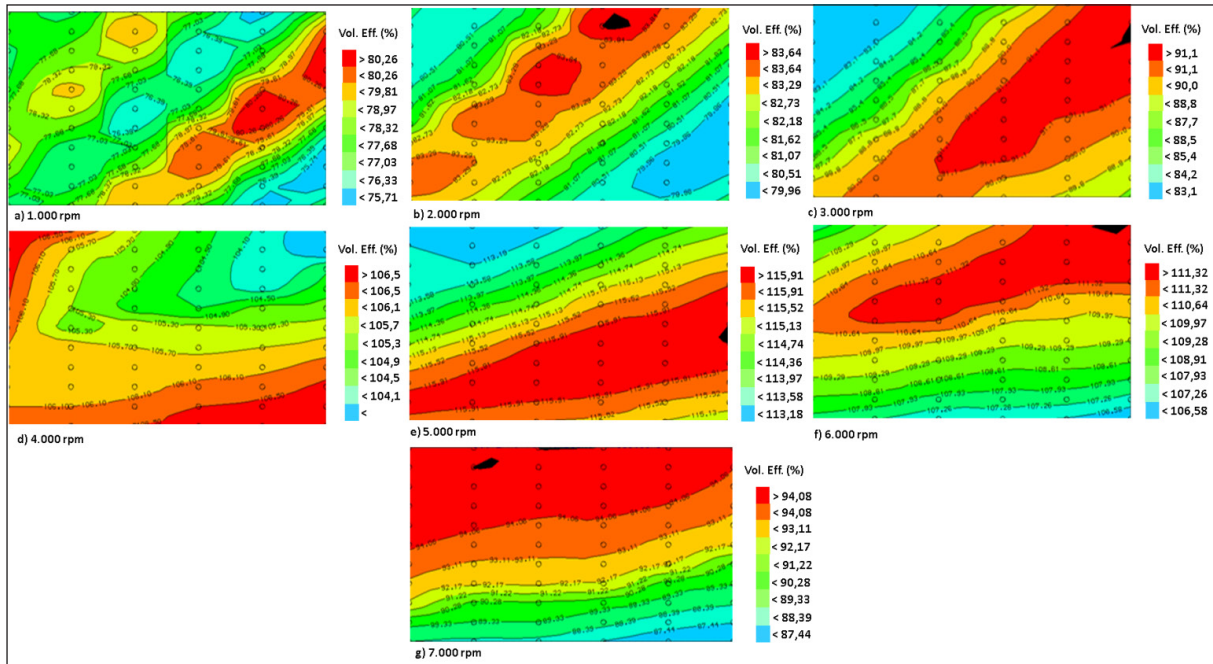5 that show the plot of the volumetric efficiency contour with the length of the intake pipe on the horizontal axis compared to the diameter, vertical axis, for each simulated rotation. As can be seen in these maps, the values of the volumetric efficiency are changing as the length of the tubes, represented on the abscissa axis and the diameter value shown on the axis of the ordinates.

Table 4 – Results of the inlet and outlet pipe geometries.

|	Outlet pipe	Inlet pipe
Length (mm)	500	350
Diameter (mm)	31	39

Figure 4 – Contour plot showing the variation of the volumetric efficiency with the variation of the length of the exhaust duct (horizontal axis) in a range of 350 mm to 600 mm and diameter (vertical axis) in a range of 28 mm to 37 mm.

Source: Lotus Engine Simulation (2016).

Figure 5 – Contour plot showing the variation of the volumetric efficiency with the variation of the length of the intake duct (horizontal axis) in a range of 250 mm to 450 mm and diameter (vertical axis) in a range of 34 mm to 44 mm - Lotus Engine Simulation (2016).
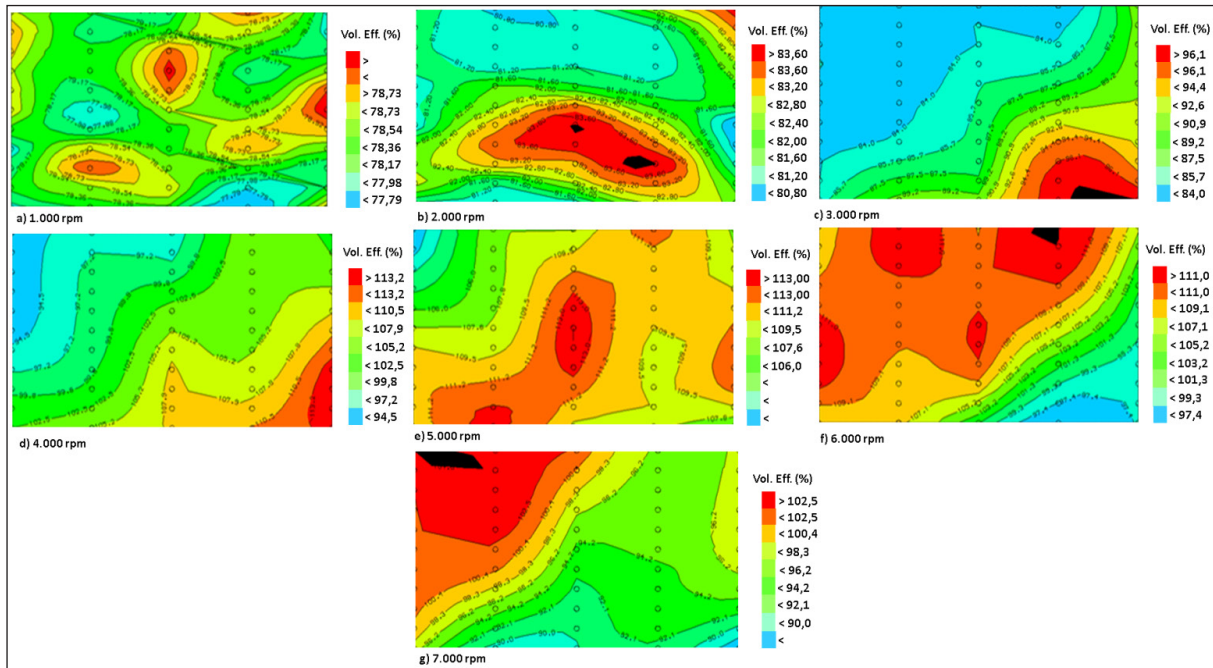
Source: Lotus Engine Simulation (2016).

4.2 Improving the opening and closing angles of the valves.

After completing the sizing of the ducts, it was possible to start the optimization simulations of the intake valve opening angle (IVO), closing the intake valve (IVC), opening the exhaust valve (EVP) and closing the exhaust valve (EVC). For the beginning of the simulation, the IVO and the EVC were considered in the TDC and the EVP and IVC in the BDC. It was adopted 0° (degrees) for all the closing and opening angles of the valves until the end of the first simulation, thus, from the results of the first simulations, these values were altered in order to find the optimum point aiming at a better efficiency volumetric.

It took twenty-one optimization simulations to find out which opening and closing angles of the inlet and exhaust valves were ideal for the simulated engine, aiming at the volumetric efficiency. After the last simulation, in which several opening and closing times of all the valves were represented and tested, a significant improvement in the performance of the simulated engine was observed. The results achieved are presented in the tab. 5.

Table 5 – Results of the opening and closing angles of the valve

	Outlet pipe	Inlet pipe
Opening (degrees)	23°	41°
Closing (degrees)	55 °	31°

Source: Lotus Engine Simulation (2016).

The gases present in the intake pipe can not change speed instantly because they have mass. The application of this principle is that the air enters the engine due to the downward movement of the piston, on reaching the BDC it starts to rise towards the TDC, even with the suction effect achieved by the piston ceasing to exist, the intake gases continue entering the cylinder due to fluid inertia. This phenomenon increases the air pressure in the cylinder upon admission and consequently its specific mass. In this way, it is interesting to keep the inlet valve open after the piston reaches the BDC. By opening the exhaust valve before the PDC, still in the expansion time, the remaining pressure itself initiates the expulsion of the gases into the atmosphere.

This behavior greatly favors the volumetric efficiency of the engine, especially at high rotations,

when the velocity of the incoming gases is high and has a great inertia, although the piston starts to rise during the compression time, this inertia continues to push the gases of into the cylinder. By delaying the closing of these valves at 55° after BDC, it aims to take advantage of the air-fuel inertia.

Anticipating the opening of the exhaust valve at 41° before BDC is intended to take advantage of the pressure from the burning of the fuel to expel the gases, where at this point, little torque is generated on the crankshaft by the remaining pressure, since the angle of the crankshaft with BDC approaching zero.

Opening the inlet valve 23° before the TDC and remaining with the outlet valve 31° open after the TDC will generate the overlap, which means keeping the two valves open at the same time, that is, taking advantage of the negative pressure generated by the high-speed exhaust flow to inhale the inlet gases after opening the inlet valve.

Early entry or increasing the amount of mass of low temperature gases entering the cylinders allows lowering of the overall temperature of the cylinder and the valves, thereby increasing the volumetric efficiency with increasing the specific mass of the intake gas.

4.3 Comparison of results

Figure 6 shows the comparison of the engine parameters for the third and twenty-first simulation.

In this figure, an improvement in the behavior of the engine tested was observed, since the maximum power (black line) ranged from 26 kW (35.35 hp) to 5,653 rpm (1) and went to 29 kW (39.43 hp) at 6,143 rpm (2), this was possible by modifying the moment of opening and closing of the valves. In this way, an improvement of 11.5% in power was obtained. At the same time, the maximum torque (blue line) increased from 47 Nm (4.79 kgfm) to 5,041 rpm to 50 Nm (5.10 kgfm) at the same rotation, resulting in an improvement of 6.3%.

However, all power and torque regimens were above the values found in the third simulation, except in the range of 2,225 rpm to 2,837 rpm (4).

Regarding fuel consumption, improvements can be seen, since at the point of maximum torque, there was a reduction of 7.5% of fuel consumption. It was evidenced in all engine work regimen; consumption gains mainly below 1,490 rpm (5).

FIG. 7 represents the comparison of the third and seventy-first simulation for the volumetric efficiency and residual gases.

As for the volumetric efficiency, it also had a gain in its value in almost the entire engine speed regime

for the new valve opening and closing configuration. However, in the interval of 2,225 rpm at 2,837 rpm (1) this finding was not evidenced. It is noted, when comparing Figs. 6 and 7, the higher the volumetric efficiency, the better the engine efficiency, which means rotations with high suction capacity of the atmospheric air generating better power, torque and fuel consumption.

Figure 6 – Comparison between the values found after the third simulation (solid line) and the results of the twenty-first (dashed line). (Power, Torque, Effective Average Pressure and Specific Consumption)
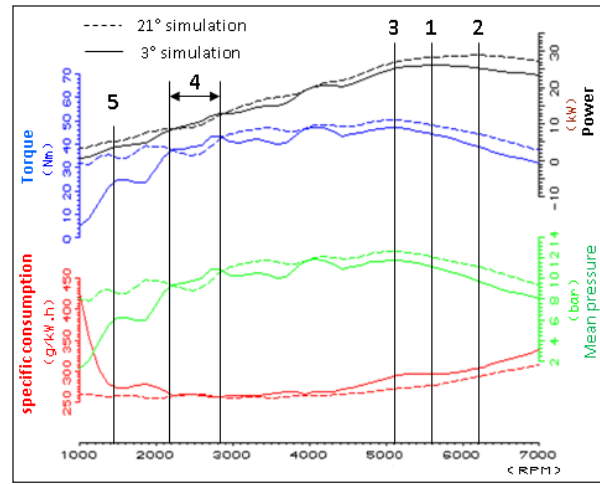
Source: Lotus Engine Simulation (2016).

Figure 7 – Comparison between the values found after the third simulation (continuous line) and the results of the twenty-first (dashed line). (Volumetric Efficiency and Residual Gases)
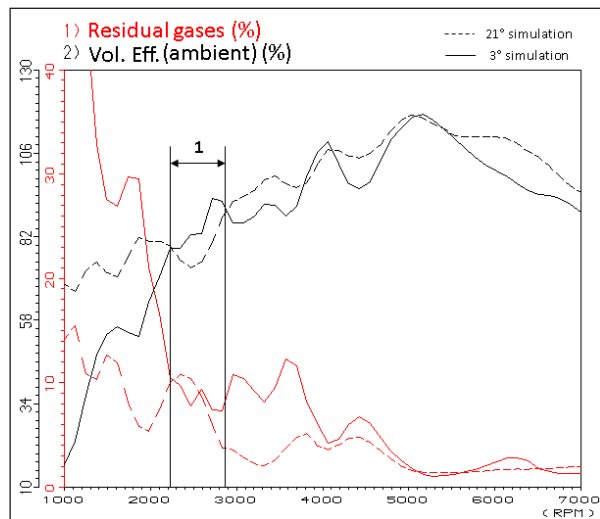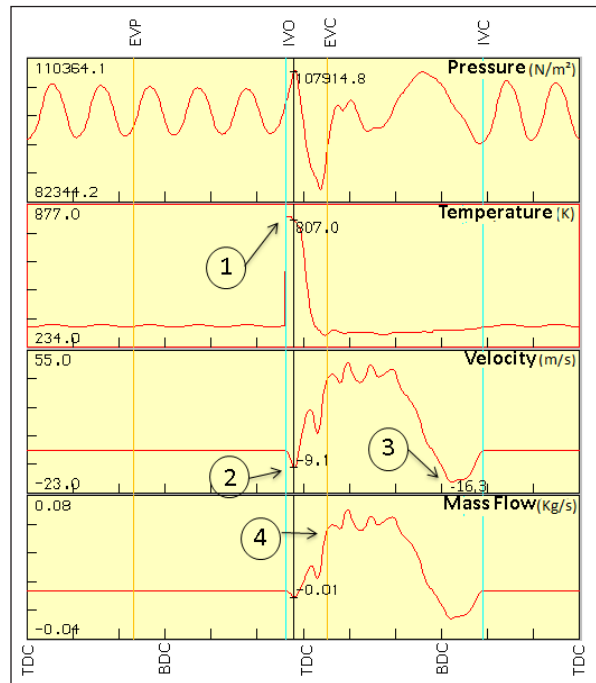

Source: Lotus Engine Simulation (2016).

Figure 8 shows the gas data in the inlet manifold near the cylinder inlet at 2000 rpm, during the four engine cycles, the vertical line (IVO) represents the intake valve opening and the line (IVC) the closing. The vertical line (EVO) shows the occurrence of the opening of the exhaust valve and the line (EVC) the closing of the exhaust valve. A characteristic is that in the temperature curve, it is observed that it makes a jump from 326 K to 807 K (1), just after the opening of the intake valve. This phenomenon is explained by the return of the exhaust gases through the intake pipe, where the fluid returned at a velocity of 9.1 m/s (2), showing that, for low rotations, the inlet valves could delay the opening.

As for the exhaust valves, for the 2000 rpm rotation, its closing could be advanced, as a return of the air-fuel with a velocity of 16.3 m / s (3) occurred. The return of the inlet gases has shown that, when the engine is at low speed, the depression formed by the exhaust gases is small and part of the product of the fuel burns escapes through the inlet pipe before returning back to the cylinder. While this exhaust gas movement occurs, the intake air is not entering the cylinders, thus reducing the volumetric efficiency.

After the exhaust valve (4) was closed, the mass flow rate of the inlet gases through the inlet valve at the cylinder was in full rise. This shows that the valves were open at the same time aided in the intake (KAKAEE, 2016). However, it should better evaluate whether the air-fuel is passing directly through the chamber and exiting the exhaust, thus losing fuel.

Figure 9 shows the behavior of the gases measured in the inlet duct at 6,000 rpm over the four engine cycles. In this figure the vertical line (IVO) represents the opening of the intake valve and the line (IVC) the closing. It is observed that, at high rotations, the return of gases by the intake no longer occurs, evidencing that the opening and closing of the intake valves are better adapted for high rotations. However, after the closing of the outlet valve, there was a decrease in the velocity of the inlet gases entering the cylinders, which reached a peak of 78.4 m/s (1), corroborating so that it could have a longer delay in its closure in this rotation. In this way, negative pressure inside the cylinders, generated by the exhaust flow during the overlap, optimized the suction of the intake gases. It is observed that the moment of opening of the intake valve occurred when the pressure in the intake manifold was high, 129,266.6 N / m² (2) improving the cylinder filling.

Figure 8 – Gas behavior measured at the intake duct at 2,000 rpm (Pressure, Temperature, Velocity and Mass Flow)

Source: Lotus Engine Simulation (2016).

Figure 9 – Gas behavior measured at the intake duct at 6,000 rpm (Pressure and Velocity)
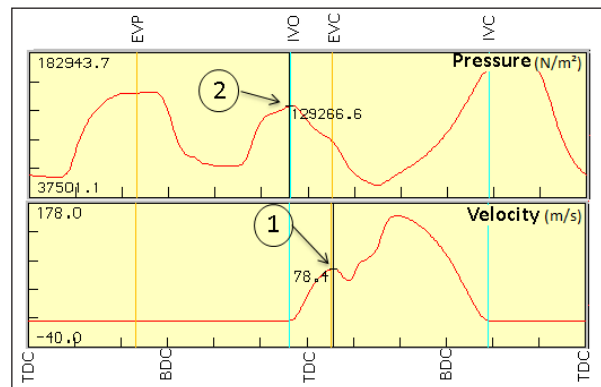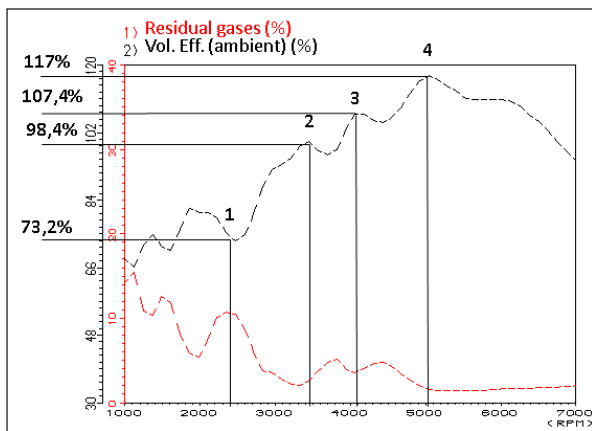

Source: Lotus Engine Simulation (2016).

Figure 10 represents the volumetric and residual gas efficiency remaining inside the cylinder after all parametric simulations. It is noticeable, however, that these values vary as the rotation changes. In some rotations, peaks or volumetric efficiency values were recorded with respective values: 73.2% at 2591.84 rpm (1), 98.4% at 3,449 rpm (2), 107.4% at 4,061 rpm (3) and 117% at 5,040.82 rpm (4). From the rotation of 3,449 rpm, the volumetric efficiency was higher than 100%, that is the engine became supercharged.

It is a challenge to estimate the volumetric efficiency in an experimental engine, due to the non-linearity of the gas flow phenomenon and the intrinsic complexities of this behavior (KAKAEE, 2016).

In order to analyze which behavior of the intake gases that more influence the gain of the volumetric efficiency, a further fifty simulations were performed, between 1,000 rpm and 7,000 rpm in 120 rpm intervals. The objective of this simulation increase is to identify, among the numerous phenomena of gas behavior, which has the greatest importance in the volumetric efficiency.

It was identified that from 3,449 rpm, the pressure in the pipeline near the inlet valve, at the instant it remains open, was higher than the atmospheric pressure, justifying supercharging (CAPETTI, 1927). The overpressure occurred due to the oscillations of the inlet gas pressure. However, pressure variation was responsible for the gain in volumetric efficiency at various engine rotations, mainly at 3,449 rpm, 4,061 rpm and 5,040.82 rpm where the intake valve opening occurred at exactly the peak of pressure (ENGELMAN, 1973).

**Figure 10 –** Result of volumetric efficiency and residual gases after parametric simulations

Source: Lotus Engine Simulation (2016).

Figure 11 shows the gas pressure upstream of the inlet valve for the four rotations of fig. 10, 2,591.84 rpm; 3,449 rpm; 4,061 rpm and 5,040.82 rpm. However, it was observed that in the rotation of 2,591.84 rpm (1), the pressure wave is small, which has little influence on the volumetric efficiency. However, for the rotation of 3,449 rpm (2) the inlet pressure was 110,030 N/m², as the atmospheric pressure of 100,000 N/m² was considered in the simulations, the pressure at that time was 110% of the atmospheric pressure. The same occurred for the rotation of 4,061 rpm, the collector pressure was 115,454 N/m² (3), 115% atmospheric pressure and

5,040,82 rpm with collector pressure reaching 121,374.9 N/m² (4), 121% of the atmospheric pressure.

At these three rotations the inlet valves opened exactly when the pressure upstream of the valve was maximum. In this situation there were substantial gains in volumetric efficiency, because the pressure of the intake manifold, by the time the intake valve opens, significantly influenced the volumetric efficiency. This finding was also evidenced by (ENGELMAN, 1973). This physical phenomenon of resonance that occurs in the engine intake pipes, as discussed in Fig. 11 occurs due to the opening and closing of the inlet valves which create a pulsating flow of gases in the inlet pipe and consequently the pressure gradients caused generate waves that propagate through the pipe in both directions (MEZHER, 2013).

Figure 11 – Gas pressure upstream of the inlet valve for revs of 2591.84 rpm; 3449 rpm; 6061 rpm and 5,040.82 rpm
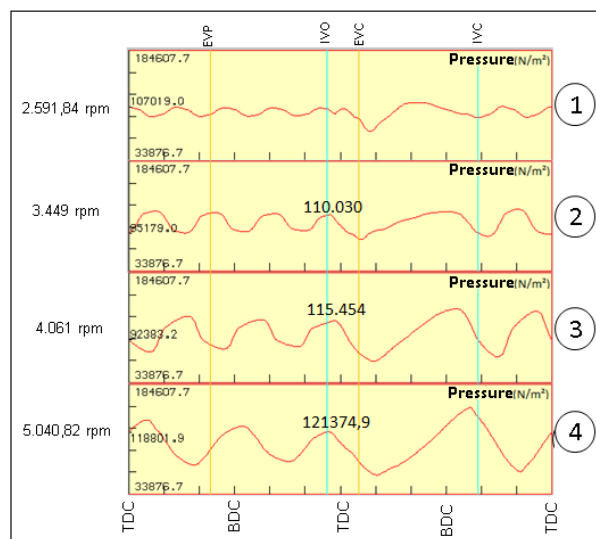
Source: Lotus Engine Simulation (2016).

This fact has to be considered in the design of the engine and the same can be divided into high pressure waves and low-pressure waves. As the inlet valves open, this rarefied oscillation is created that moves from the cylinder head to the plenum where it is reflected and returns as an over-pressure wave to the valves. However, the geometry of the plenum and pipes must be correctly designed so that the higher pressure of this ripple finds the intake valves open, thereby maximizing the air-fuel input into the engine, simulating forced induction.

The relationships of possible reasons for high volumetric efficiency are summarized in, for example:

a) Variables that were not included in the simulations, such as possible pipeline curves, losses of pipeline load etc.

b) No filters, catalyst and silencer in the simulation, which increase the losses of gas charges.

c) No geometric limitations for pipes.

d) Internal pipe roughness (0.00150 mm), reducing the flow pressure loss at the inlet.

e) Synchronism of the length, diameter and opening angle of the valves can generate a pressure oscillation that can increase the mass flow by 20%. This finding was also evidenced by (ENGELMAN, 1973).

As in the parametric simulation of this research were found the opening and closing angles of the intake valve, being 23° before the TDC and the closing of 55° after the BDC, it is possible to find the angle range that the valve is open. It's the sum of 23**°, 55° and 180°, resulting 258°.**

As the inlet valve opens once every two revolutions, so will be the difference of 720**° and 258°, resulting 462°.**

The time the valve closes, generating a pressure wave and the moment it opens, has a range of 462°. In this way it is possible to find the time tt for the rotation range from one thousand to seven thousand rpm, considering the sound velocity of 340.29 m/s, so it is possible to assemble a graph, fig. 12, showing that the lower the rotation, the larger the intake pipe should be, aiming only at the pressure wave and neglecting the losses of loads, according to eq.1. (DOMSCHKE, 1963).

Figure 12 – Ideal theoretical length of intake ducts for various engine speed.
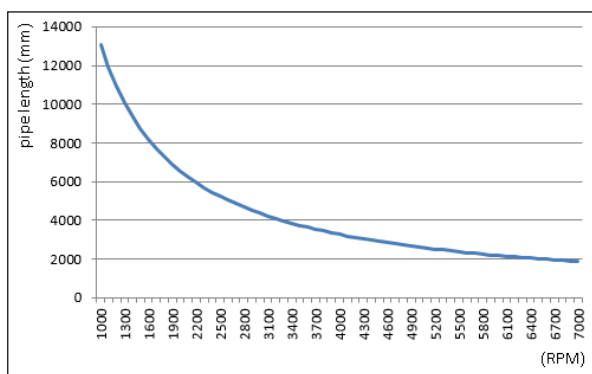
Source: Author

To calculate the length of the entire inlet pipe for a given rotation, use equation 1 as follows:

$$L = t \frac{360}{60} n \frac{2L}{1000\,c} \frac{1}{0{,}012n} \qquad (1)$$

Considering $L, L,$ the **length of the tube,** cc the speed of sound (m/s), nn the engine speed (RPM) and tt the time for the wave to go back and forth inside the pipe.

It is evident that nn is annulled because it is possible to simplify in eq.1, however, tt is directly connected to nn. So, nn still influences the formula.

Two considerations become necessary after performing these calculations. The first is that at 1,000 rpm the intake pipe should have an increase in length of 700% compared to 7,000 RPM. This shows that the geometry of the ducts should change in large proportions as the rotation varies. As in most modern engines, they do not have a variable geometry manifold, the adjusted dimensions of the intake ducts have their main benefit in a very limited speed range. The second consideration is that the lengths found are very extensive, ranging from 1,800 mm to 7,000 RPM going up to 13,100 mm at 1,000 rpm. It is almost impossible to adapt a pipeline of this length into the chassis of the vehicle. In this way the ducts must be shortened in a way that maintains the benefits of the pressure wave. This can be done by dividing the length by half, in this way the pressure wave will travel twice through the pipeline and still reach the valve at the right time. However, more divisions can be made in the pipelines to achieve the same goal.

5 Conclusion

The results of the numerical simulations were very satisfactory, since the proposed model reproduced in an expected way the behavior of the volumetric efficiency. However, it was evident that the experimental model has a higher efficiency compared to the theoretically obtained curve data. It is noteworthy that the simulated bench engine has all the geometry of the inlet and plenum ducts not optimized causing a deficiency in the filling of the cylinders justifying the low volumetric efficiency

In the design of the engine, both the combustion chamber geometry and the compression ratio must be designed not only for the resonant frequency, but also based on other relevant guidelines such as the purpose

of the engine, detonation, pre-ignition, burning among others. However, piping design must ensure that pressure waves occur at the correct time to encourage cylinder filling.

It should be noted that for different rotations, gas flows with different characteristics entered and exited the engine. These differences can reach high or low velocities, so that any variation of rotation will imply changes in flow velocities. However, the ideal would be for the engines to always work at the same rotations and loads, which is not possible in vehicle applications. Therefore, the ideal is to dimension the valves, inlet and exhaust ducts and the profile of the valve controls so that the cylinders have the best filling over a wide range of rotation, not in a single condition.

The perfect flow of the gases through the inlet and exhaust pipes and, consequently, the filling and emptying of the cylinders, is strongly linked to the dynamics of opening and closing the valves. In general, there is no general formula for setting the correct opening, closing and lifting times for the valves, since such a configuration depends on several factors and especially the engine application.

REFERENCES

ALBRECHT A, CORDE G AND KNOP V. et al. 1D Simulation of Turbocharged Gasoline Direct Injection Engine for Transient Strategy Optimization, **SAE Technical Paper** No. 2005-01-0693, 2005.

ANDREATTA C AND PEDERIVA R. Valve Train Kinematic and Dynamic Simulation, **25th SAE Brasil International Congress and Display**, São Paulo, Brasil. SAE 2016-36-0213 E, October 2016.

CAPETTI A. Effects of Intake Pipe on the Volumetric Efficiency of an Internal Combustion Engine (Translation from Italian "Annalidella R. Scoulad'Ingegnoria di Padova, December 1927."), UNT Digital Library, http://digital.library.unt.edu/ark:/67531/metadc65350/, 2012.

CERDOUN M, CARCASCI C AND GHENAIET A. An approach for the thermal analysis of internal combustion engines' exhaust valves, **Applied Thermal Engineering** 102 1095–1108, 2016.

CEVIZ MA. Intake plenum volume and its influence on the engine performance cyclic variability and emissions. **Energy Conversion and Management**, Vol. 48, n. 3, p.961 966, 2007.

CHALET D, MAHE A AND MIGAUD J. et al. A frequency modelling of the pressure waves in the inlet manifold of internal combustion engine, **Appl. Energy** 88(2011) 2988–2994.

CHALLEN B AND BARANESCU R. Engine Reference Book. 2 ed**. SAE International**, 1999.

COSTA RC, HANRIOT SM AND SODRÉ JR. Influence of intake pipe and diameter on the performance of a spark ignition engine, J. Braz. Soc. Mech. **Sci. Eng**. 36 (2014)29–35.

DEB M, BANERJEE R AND MAJUMDER A. et al. Multi objective optimization of performance parameters of a single cylinder diesel engine with hydrogen as a dual fuel using pareto-based genetic algorithm, Int. J. **Hydrogen Energy** 38(2014) 8063–8077.

DOMSCHKE AG AND LANDI FR. Motores de Combustão Interna de Embolo, **Departamento de livros e** Publicações **do Grêmio Politécnico da USP**, 1963.

ENGELMAN HW. Design of a Tuned Intake Manifold, ASME Paper 73-WA/DGP-2, 1973.

GALINDO J, ARNAU J AND TISEIRA A. et al. Solution of the turbo compressor boundary condition for one-dimensional gas-dynamic code, Math. Comput. Model. 52 (2010) 1288–1297.

HARRISON MF AND DUNKLEY A., The acoustics of racing engine intake systems, **Journal of Sound and Vibration** 271 959–984, 2004.

KAKAEE AH, MASHADI B AND GHAJAR M. A novel volumetric efficiency model for spark ignition engines equipped with variable valve timing and variable valve lift Part 1: model development Proceedings of the Institution of Mechanical Engineers, Part D: **Journal of Automobile Engineering,** vol. 231, 2: pp. 175-191, First Published June 8, 2016.

MEZHER H, CHALET D AND MIGAUD J. et al. Frequency based approach for simulation pressure waves at the inlet of internal combustion engine using parameterized model, **Appl. Energy** 106 (2013) 275–286.

MONIR H, SALEHI R AND SALARIEH H. et al. Real-time estimation of the volumetric efficiency in spark ignition engines using an adaptive sliding-mode observer, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, vol. 229, 14: pp. 1925-1933. First Published March 23, 2015.

OCH H, MOURA M AND MARIANI C. et al. Volumetric efficiency optimization of a single-cylinder D.I. diesel engine using differential evolution algorithm, **Applied Thermal Engineering** 108 (2016) 660–669.

PROVASE IS, PIMENTA MM AND MARIANI ALC. Analysis of Flow Detachment in the Intake Manifold of a Formula Sae Vehicle. Escola Politécnica Da Universidade De São Paulo, XXIII **SIMEA Simpósio Internacional de Engenharia Automotiva**, 2014.

RIBEIRO SY, SILVA JA AND MOURA M. Eficiência Volumétrica em Motor Monocilíndrico de Combustão Interna a Gasolina. **VI Congresso Nacional de Engenharia Mecânica CONEM**, agosto de 2010, Campina Grande, Paraíba, Brasil, 2010.

ROSTEK E, BABIAK M AND WRÓBLEWSKI E. The influence of oil pressure in the engine lubrication system on friction losses, **TRANSCOM 2017**: International scientific conference on sustainable, modern and safe transport, Procedia Engineering 192 771 – 776, 2017

SILVA FG AND ALVES MAC. Ensaio e Determinação de curvas características para o Motor Eh17-2, Trabalho realizado como parte dos requisitos para aprovação na disciplina de Laboratório de Motores de Combustão Interna, **Universidade Federal de Juiz de Fora** UFJF, 2017.

TORREGROSA J, GALINDO J AND GUARDIOLA C. et al. Combined experimental and modelling methodology for intake line evaluation in turbocharged diesel engines, **Int. J. Automot.** Technol. 12 (2011) 359–367.

WINTERBONE DE AND PEARSON RJ. Design Techniques for Engine Manifold: Wave Action Methods for IC Engines. Warrendale: **Society of Automotive Engineers**, Inc, 1999.

ABBREVIATIONS

LES - Lotus Engine Simulation.
IVO - Intake valve opening
IVC - Intake valve closing.
EVO - Exhaust valve opening.
EVC - Exhaust valve closing.
TDC - Top dead center
BDC - Bottom dead center