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OTIMIZATION OF VOLUMETRIC EFFICIENCY BY CHANGING THE PIPES LENGHTS OF SINGLE-CYLINDER INTERNAL COMBUSTION ENGINES USING DIFFERENTIAL EVOLUTION

Stephan Hennings Och
Luís Mauro Moura
Viviana Cocco Mariani
Pontifícia Universidade Católica do Paraná - PUCPR, Rua Imaculada Conceição 1155, 80215-901, Curitiba-PR, Brasil
stephan.och@pucpr.br, luis.moura@pucpr.br, viviana.mariani@pucpr.br

José Antonio Velásquez
Universidade Tecnológica Federal do Paraná - UTFPR, Av. Sete de Setembro 3165, 80230-901, Curitiba-PR, Brasil
velasquez@utfpr.edu.br

Eric Domingues
CORIA UMR CNRS 6614, Avenue de l'Université 76800, Saint Etienne du Rouvray, France
eric.domingues@coria.fr

Abstract. *Environmental standards and specific targets of efficiency and consumption become one of the more critical phases in the design of internal combustion engines. To analyse the behaviour of engines, engineers and scientists can use experimental techniques, but they are expensive in cost and analysis time. On the other hand, the simulation of thermal processes of combustion engines by digital computing allows reduction in time and cost and offers wealth of details. Three dimensional flow simulations of internal combustion engines are becoming more common and may contribute in valuable information about the displacement of the fluids in different parts of the engine and evolution of combustion within the cylinder. But despite showing a more detailed flow of fluid, such simulations provide a relatively high computational cost when compared with one-dimensional models. The one-dimensional model is simpler to be used and provides reasonable accuracy in global quantities, such as volumetric efficiency. The volumetric efficiency is a function of gas flow properties, geometry of engine parts and operating parameters. Normally, this function can be nonlinear, discontinuous, non-differentiable and multimodal. Because of these characteristics, optimizing the volumetric efficiency with deterministic methods becomes a difficult process. On the other hand, stochastic methods have the advantage of using only the information of the function to be optimized. In this study, results of a technique called the Differential Evolution are presented. Another interesting point of this technique is the simplicity to implement and to parallelize, since the individuals of a generation are all independent. This technique requires a large number of objective functions calls, so it was decided to work with a one-dimensional model. This paper presents the domain of design variables, and the optimization is done for different engine speeds. This work allows the analysis of the influence of physical phenomena in the intake process of an internal combustion engine and their evolutions with engine speed.*

Keywords: *IC engines, Optimization, Admission Process, One zone model, Differential Evolution*

1. INTRODUCTION

The purpose of internal combustion (IC) engines is power production through the chemical energy contained in fuel. The work performed by the fluid is transferred to mechanical components and will be converted in rotation mouvement on the shaft. These engines are found in transportation (land, sea and air) and electricity production, because they are simple, robust and have a high power/weight ratio. The design phase has become one of the most critical in the manufacturing process of an engine due the tightening on pollutants and increased targets for efficiency and fuel consumption, thus stimulating research on this subject (D'Errico *et al.*, 2011). The goal of intake and exhaust process is to remove burned gases at the end of the expansion phase and admit a new charge for the next cycle. Thus, introducing and maintaining the maximum mass within the cylinder is the first objective of the gas exchange process. The efficiency of IC engines depends on inertial, pulsating and friction phenomena that occur in intake and exhaust manifolds. The optimization of the gas exchange in IC engines necessarily involves a thorough analysis of different variables involved in this process. Thus, for example, intake and exhaust systems can be sized and have their geometrical configurations defined according to the desired focus characteristics, such as fuel consumption, torque and volumetric efficiency. The IC pipes optimization can be made through tests on a physical engine or through computer simulation. The physical tests are expensive and require the engine prototype in an advanced stage of development (Torregrosa *et al.*, 2011), which goes in the opposite direction to industry needs. However in the last decades, computer simulation developed rapidly, and these techniques permited

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very accurate results, allowing to test settings at the beginning of the project, reducing cost and design time.

Multi-dimensional simulations have been used to improve the fluid flow through various devices, such as ducts and combustion chamber, among others. Despite the power of computational resources increased in recent years, the simulation of an entire IC engine needs a high computing time. The 1D fluid dynamic model simulates the transient movement of pressure waves in normal direction of duct system. This model is simpler than the multi-dimensional model and thus drastically reduces the simulation time and usually does not require high-performance computers. This justification becomes more significant in the case of mathematical optimization that requires a long time to simulate the thermal operation of the engine.

At higher speeds, the inertia of the gas in the inlet duct means that there is input of mass into the combustion chamber even when the piston is already moving at TDC. To take advantage of the ram effect (inertia), the inlet valve closes around 40 to 60 degrees after BDC. This is effective at high speed, but not at low speed because the piston has a lower average speed, which makes the inertia effect of the gas lower. Thus the force due to gas pressure within the combustion chamber is larger than the inertial force of gas flow which causes undesirable backflow at the intake valve.

The parameter used to measure the effectiveness of the admission process is the volumetric efficiency. It is defined as the ratio of the mass flow rate of the intake air and the mass flow rate to fill the volume displaced in local conditions. The volumetric efficiency is a function of the intermittent nature and properties of the pulsating air flow in the intake duct, which can produce resonances at certain speeds. This can improve performance in some engine speeds but can reduce others, depending on the dimensions and shape of ducts and valves timing.

In common internal combustion engines, the lenght of ducts and the control of the intake and exhaust valves is fixed. Pipelines and lift valves curves are fixed and allow to optimize the performance only in a specific rotation speed (usually high speed). The idea of changing these parameters is to work at optimal points in various speed ranges. A practical application of this are the urbain vehicles. Normally the engine is used at low speed, or in conditions where it is not efficient.

These parameters are used in experimentals and theoretical works to increase the volumetric efficiency. Kesgin (2005) propose a theoretical study of the effect in engine performance of changing the length and the diameter of ducts sytems and valve timing and valve lift profile. Sergio de Morais Hanriot (2001) show the effect of inserting a Helmholz resonator in intake duct. Ceviz and Akın (2010) make experimental test in a multicylinder SI engine for differents intake duct length and they propose a new intake manifold with variable length. D'Errico *et al.* (2011) applied some multi-objetive optimization techniques choosing the length ducts and valves timings as design variables.

The objective of this work is to apply a mathematical model of the processes of intake, compression, combustion, expansion and exhaust of single cylinder CI (Compression Ignition) naturally aspirated engine and operating with diesel. The specific objective is to find better lengths to maximize volumetric efficiency at typical speeds of CI engines. The optimization will be done through the stochastic technique called Differential Evolution proposed by Storn and Price (1997).The purpose of this work is not dedicated directly to the reducing of emission, but experimental studies show that the variation of the length of the duct does not affect the exhaust emissions effectively. However, it reduces fuel consumption by engine power and therefore decreases the total production of exhaust emission (Ceviz and Akın, 2010).

2. MATHEMATICAL MODEL

2.1 Fluid dynamics model in pipes lines

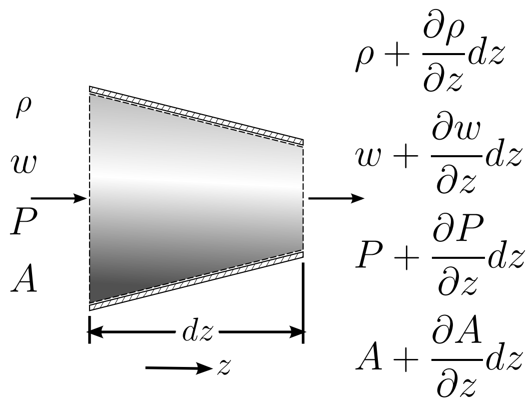

Figure 1. Control volume in pipe section.

The process of combustion systems and catalytic after-treatment can be described by the fundamental laws of conservation of mass, momentum and energy. This description consists of a set of coupled partial differential equations between

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the transport process and the complex chemical process that requires high computational resources and has little use in systems with the goal of optimization. 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 (Pearson and Winterbone, 1997). It is necessary to consider the friction and heat transfer between the flow and the walls of the tube (Payri *et al.*, 2004). The variation of specific heat ratio of the fluid, k, can be neglected. Applying the equations of conservation of mass, momentum and energy in the control volume shown in Figure 1, leads to the following systems of equations:

$$\frac{\partial U}{\partial t} + \frac{\partial F}{\partial z} = f \tag{1}$$

where:

$$U = \begin{bmatrix} \rho \\ \rho w \\ \dfrac{P}{k-1} + \dfrac{\rho w^2}{2} \end{bmatrix}, \quad F = \begin{bmatrix} \rho w \\ \rho w^2 + P \\ w\left(\dfrac{kP}{k-1} + \dfrac{\rho w^2}{2}\right) \end{bmatrix}, \quad f = \begin{bmatrix} -\dfrac{\rho w}{A}\dfrac{dA}{dz} \\ -\dfrac{\rho w^2}{A}\dfrac{dA}{dz} - \rho F_f \\ -w\left(\dfrac{kP}{k-1} + \dfrac{\rho w^2}{2}\right)\dfrac{1}{A}\dfrac{dA}{dz} + \rho\dot{q} \end{bmatrix} \tag{2}$$

Where ρ, w, A, P, k, F_f, $\dot{q}$ are density, velocity, cross-section area, pressure, specific heat ratio, friction force and heat transfer per mass, respectively. 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 (Kadalbajoo and Kumar, 2006). 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.

$$U_{i+1/2}^{n+1/2} = \frac{U_{i+1}^n + U_i^n}{2} - \frac{\Delta t}{2\Delta z}\left(F_{i+1}^n - F_i^n\right) + \frac{\Delta t}{4}\left(f_{i+1}^n + f_i^n\right) \tag{3}$$

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

$$U_i^{n+1} = U_i^n - \frac{\Delta t}{\Delta z}\left(F_{i+1/2}^{n+1/2} - F_{i-1/2}^{n+1/2}\right) + \frac{\Delta t}{2}\left(f_{i+1/2}^{n+1/2} + f_{i-1/2}^{n+1/2}\right) \tag{4}$$

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

$$\Delta t \leq \frac{\Delta z}{(|w| + c)_{max}} \tag{5}$$

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. (El-Rahman *et al.*, 2004).

2.2 Combustion chamber model

The working fluid composed of combustion gases is considered to be homogeneous on the entire volume of the combustion chamber. This hypothesis is closer to reality when engines operate at high speed and produce high intensity vortices, wich allows rapid gases mixture. Therefore, in this work we neglect the gradients of pressure and temperature, atomization and vaporization of the fuel, ignition delay. This methodololy is called zero-dimensional model. The reaction equation is given below:

$$x_{13}\left[C_nH_mO_lN_k + \frac{n + \frac{m}{4} - \frac{l}{2}}{\phi}\left(O_2 + \alpha N_2 + \beta CO_2 + \gamma H_2O + \delta Ar\right)\right] \rightarrow x_1H + x_2O + x_3N + \\ + x_4H_2 + x_5OH + x_6CO + x_7NO + x_8O_2 + x_9H_2O + x_{10}CO_2 + x_{11}N_2 + x_{12}Ar \tag{6}$$

Applying the mass and energy balance equations and the ideal gas equation, we obtain the following system of differential equations:

$$\frac{dm}{d\theta} = \sum \frac{dm_i}{d\theta} \tag{7}$$

$$\frac{dP}{d\theta} = P\left[\left(\frac{1}{T} + \frac{1}{R}\frac{\partial R}{\partial T}\right)\frac{dT}{d\theta} + \frac{1}{m}\frac{dm}{d\theta} - \frac{1}{V}\frac{dV}{d\theta} + \frac{1}{R}\frac{\partial R}{\partial \phi}\frac{d\phi}{d\theta}\right]\left(1 - \frac{P}{R}\frac{\partial R}{\partial P}\right)^{-1} \tag{8}$$

$$\frac{dT}{d\theta} = \frac{\frac{dV}{d\theta}\left(\frac{A}{V} - \frac{P}{m}\right) - \frac{d\phi}{d\theta}\left(\frac{\partial u}{\partial \phi} + \frac{A}{R}\frac{\partial R}{\partial \phi}\right) + \frac{1}{m}\left[\frac{\delta Q}{d\theta} + \sum (h_i - u - A)\right]\frac{dm_i}{d\theta}}{\frac{A}{T} + \frac{\partial u}{\partial T} + \frac{A}{R}\frac{\partial R}{\partial T}} \tag{9}$$

where:

$$A = \frac{P\frac{\partial u}{\partial P}}{1 - \frac{P}{R}\frac{\partial R}{\partial P}} \tag{10}$$

In Equations(7-10), P, V, T, m, u, h, θ, R, ϕ, and Q are the pressure, volume, temperature, mass, internal energy, enthalpy, crankshaft angle, gas constant, fuel/air equivalence ratio and heat transfer, respectively. The indice i refers to the admission or exhaust valves or injector. The rate of heat transfer is obtained by Woschni correlation (Woschni, 1967), which is the most used today (Ma *et al.*, 2008). 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 c_d, obtained experimentally (Shapiro, 1954) . The step of crankshaft angle is obtained from the time step of gas flow solution in the intake and exhaust ducts.

It is considered that the fuel burns instantaneously when entering the combustion chamber and evolution of combustion reaction is given by two Wiebe functions, one for the premix combustion and another one for the diffusive phase, the Wiebe function is given by:

$$\chi = 1 - exp\left[-6.908\left(\frac{\theta - \theta_{ig}}{\Delta\theta_d}\right)^{\xi+1}\right] \tag{11}$$

The volumetric efficiency is calculed in compression phase with Equation 12.

$$\eta_v = \frac{m_c}{\rho_0 V_{disp}}\left[\frac{1 - \phi_c/\phi_{cycle}}{1 + \phi_c\,(F/A)_{st}}\right] \tag{12}$$

In Equation 12, η_v, m_c, ρ_0, V_{disp}, ϕ_c, ϕ_{cycle}, $(F/A)_{st}$ are the volumetric efficiency, cylinder mass, air density, displaced volume, equivalence ratio of mixture, equivalence ratio of the cycle and stoichiometric fuel-air ratio, respectively.

3. DIFFERENTIAL EVOLUTION METHOD

Optimization methods can be classified into deterministic and stochastic. In the deterministic method, the solution obtained at each iteration moves in the direction of the gradient and ends when the gradient is near zero. As it depends of gradient, we should know the derivatives or give them by numerical schemes. Unfortunately these methods may converge to a local optimum. The stochastic methods only need to evaluate the objective function that can be: discontinuous, non-differentiable, multimodal and difficult to represent.

The evolutionary algorithms (EA) are stochastic optimization algorithms inspired by Darwinian natural evolution. According to the Darwinian Theory, only best adapted individuals survive natural selection and reproduce from one generation to another. One terms of optimization, the evolution is reflected by an iterative process of searching an optimal point in the design variables domain. The evolutionary optimization process begins by initialization step: an individual's finite amount, x_i, generally randomly chosen within the field of design variables forms the initial population, P_0. Then applies the operator of variation (mutation and crossover) that allows to create a new set of individuals, called the children

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population (this step is entirely stochastic). These children will be evaluated and combined with their parents in order to decide which ones will replace certain parents and will be part of the next generation (selection stage).

Note that in most cases of real-world applications, the cost of evolutionary algorithms depends essentially on evaluation step. For example, if you want to evolve a few tens individuals during a several tens generations, some thousands of calculations of the objective function must be performed by a rating usually expensive.

The method of Differential Evolution was proposed by Storn and Price (1997). This method differs significantly from other evolutionary algorithms in the information about the distance and direction of the current population used to guide the search process. This method is divided into three steps: mutation, crossover and selection.

For each target vector $x_{i,G}$ with $i = 1, 2, ..., NP$ (NP is the number of individual), the vector mutation is created by the following equation:

$$v_{i,G+1} = x_{r_1,G} + F.\left(x_{r_2,G} - x_{r_3,G}\right) \tag{13}$$

with random indices r_1,r_2,r_3 $\in 1, ..., NP$ are distinct from each other and $F > 0$. The choice of random indices r_1,r_2,r_3 is different from the current index, i. Therefore, NP must be greater than or equal to four. F is a positive real value between 0 and 2 that controls the amplification of $(x_{r_2,G} - x_{r_3,G})$. One schema of this process can be visualized in Figure 2.

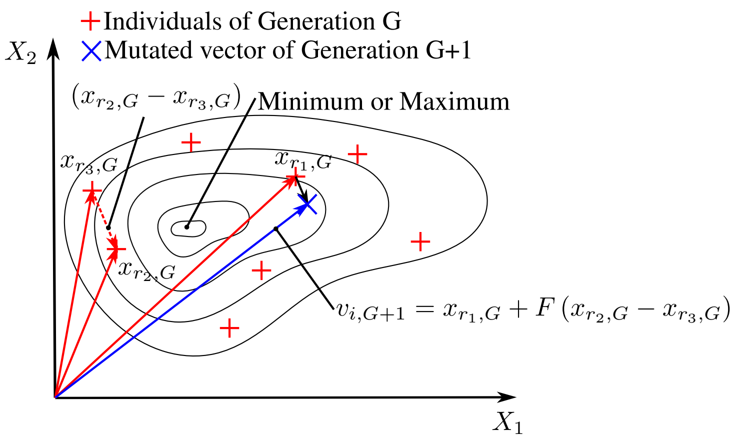

Figure 2. Schema of Differential Evolution method.

The objective of crossing step is to increase the diversity of the population, disrupting the vector mutation. Thus the experiment vector $u_{ji,G+1}$ is calculated as:

$$u_{ji,G+1} = \begin{cases} v_{ji,G+1} & \text{if}\,(\text{randb(j)} \leq \text{CR}) \quad \text{or} \quad j = \text{rnbr}\,(i)\,, \\ x_{ji,G} & \text{if}\,(\text{randb(j)} > \text{CR}) \quad \text{and} \quad j \neq \text{rnbr}\,(i) \end{cases} \qquad j = 1, 2, ..., D. \tag{14}$$

In the above, $randb\,(j)$ is the jth term in a uniform random number generator between 0 and 1. CR is the crossover constant between 0 and 1 chosen by the user. $rnbr(i)$ is to randomly select the index $\in 1, 2, ...D$, which this ensures that $u_{ji,G+1}$ receives at least one parameter $v_{i,G+1}$.

Selection consists to decide which individual will become a member of the generation $G+1$. The experiment vector $u_{i,G+1}$ is compared with the target vector $x_{i,G}$ using the criterion ambitious. If the vector $u_{ji,G+1}$ produces a function value less costly than $x_{i,G}$ then $x_{i,G}$ is equal to $u_{ji,G+1}$, otherwise, the value is kept equal to $x_{i,G}$. There are several strategies for crossover, in this paper we used the ED/rand/1/bin (Storn and Price, 1997) given by Equation (13) with multiplier factor mean difference, F, equal to 0.8, the crossover probability also equal to 0.5. The number of generations is equal to 100 and 20 experiments are done.

Currently we face what is usually called "the end of Moore's Law" which predicts an exponential increase of computing capacity that can be obtained on one processor. One solution to this problem in the case of calculations that require high computational resource is the parallelization of the algorithm on multiple processors (Yagoubi, 2012). This methodology was applied in this work, using the MPI library, where each process is responsible to calculate the objective function of an individual of the population.

4. RESULTS

For this study we define an hypothetical engine with caracteristics representative of a real engine. Main data from this engine are show in Table (1). Note that this table shows the values of the lenghts of the intake and exhaust ducts, these

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values are taken as a reference for later comparison with the optimas values obtained by this methodology.

Table 1. Main engine specifications.

Engine type	Compression Ignition
Compression Ratio	16:1
Bore	93 mm
Displacement	170 mm
Stroke	103 mm
Injection	Direct
Air metering	Naturally aspirated
Intake duct length	380 mm
Exhaust duct length	1500 mm
Number of admission valves	2
Admission valve open (avo)	10° BTDC
Admission valve close (avc)	37° ABDC
Diameter of admission duct	48 mm
Number of exhaust valves	1
Exhaust valve open (evo)	42° BBDC
Exhaust valve close (evc)	10° ATDC
Diameter of exhaust duct	58 mm

In Figure 3 is shown the result of volumetric efficiency due to the length of the intake duct for some typical speed of CI engines. These data were obtained from the model presented in mathematical model section. In this figure we can see the variations of volumetric efficiency due to the length of the intake duct for diferent rotations speed of local maxima and minima are observed. This already allows predicting that optimization methods based on derivatives may present difficulties and stop the iterative process in these points. Other information that can be extracted from this graph is that each point of rotation has different maximum. In the case of low speed (1000 rpm) it can be noticed that the maximum point is located in a high length of duct. And as increasing the optimal length rotation speed decreases. This decrease is not linear, as shown by the Figure 3. This figure also allows to extract the range of the design variable, which was chosen between 0.1 and 3 m.

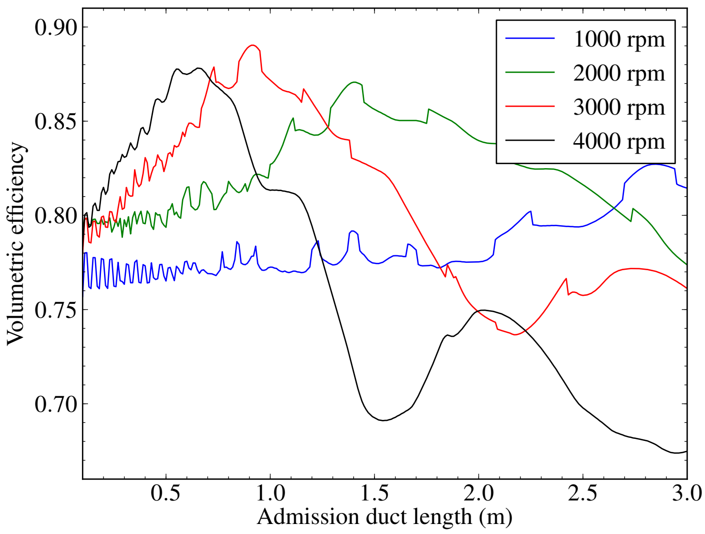

Figure 3. Volumetric efficiency versus admission length for various rotation speed.

Table 2 shows results of optimization using only one design variable (length of intake duct) runner for several rotations speeds. The minimum, mean, maximum and standard deviation values refer to 20 simulated experiments for each rotation speed. In maximum and minimum values are also shown the values of the duct length (desing variable). A quick comparison of the results shown in Table 2 and Figure 3 shows that for tested case, the Differential Evolution method can capture the global maximum value within the range of design used. In the results for the 1500, 2000, 2500, 3000, 3500 and 4000 rpm, it can be observe that all experiments converged to points close to the maximum. It was also observed

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that the volumetric efficiency maximized increase with speed up to 3500 rpm. For the 4000 rpm speed the volumetric efficiency maximized was lower at 3500 rpm.

Table 2. Volumetric efficiency optimization results with the admission length duct as parameter.

Speed (rpm)	Minimum		Maximum		Mean	Std. Dev.
	η_v	$L_{adm}(m)$	η_v	$L_{adm}(m)$	η_v	η_v
1000	0.827198	2.842779	0.827431	2.862864	0.827224	51×10^{-6}
1500	0.851696	1.888264	0.851815	1.887902	0.851712	26×10^{-6}
2000	0.870777	1.403188	0.871751	1.402372	0.870912	257×10^{-6}
2500	0.872137	1.110794	0.872892	1.110793	0.872229	179×10^{-6}
3000	0.890389	0.915987	0.890413	0.920036	0.890399	6×10^{-6}
3500	0.892486	0.770752	0.892543	0.771024	0.892492	13×10^{-6}
4000	0.878490	0.660094	0.878509	0.660138	0.878492	4×10^{-6}

In Figure 4 are shown pressure values in cylinder and in inlet valve port. The curve at 1000 rpm is in left side and the curve at 3000 rpm is in the right side. Results of pressure in the valve port and into the cylinder for the reference case (L_{adm}=0.38 m) and for the optimum length are shown. It can be notice that in both curves the optimization find a way where the pressure in the inlet end is higher than the pressure for the reference case. Namely it makes the wave to have a specific format that upon closing of the inlet valve pressure is close to the maximum pressure wave.

On left curve we observe that the frequency wave tube pressure reference is greater than the frequency of the wave optimized. This is due to the fact that the reference tube is short, ie the distance traveled by the wave is less causing multiple reflections exist in the same time period.

The optimal point "tries" to fit a period of oscillation of the wave inside the opening time of the valve and also "try" to get at the opening of the intake valve the pressure is in its phase of decrease. Therefore, the pressure wave is in phase with the rate of change of the volume inside the cylinder creating a constructive wave with greater pressure drop. This higher pressure drop travels along the tube and is reflected at the open end as a wave of pressure which returns to the cylinder. It should naturally come in moments before the valve closes.

For 1000 rpm the inertia of the gas in the intake duct is small and therefore does not influence significantly the filling curve, but at 3000 rpm but its importance is greater causing the filling of the cylinder is not only dependend to the frequency of air in duct and phase of the pressure wave.

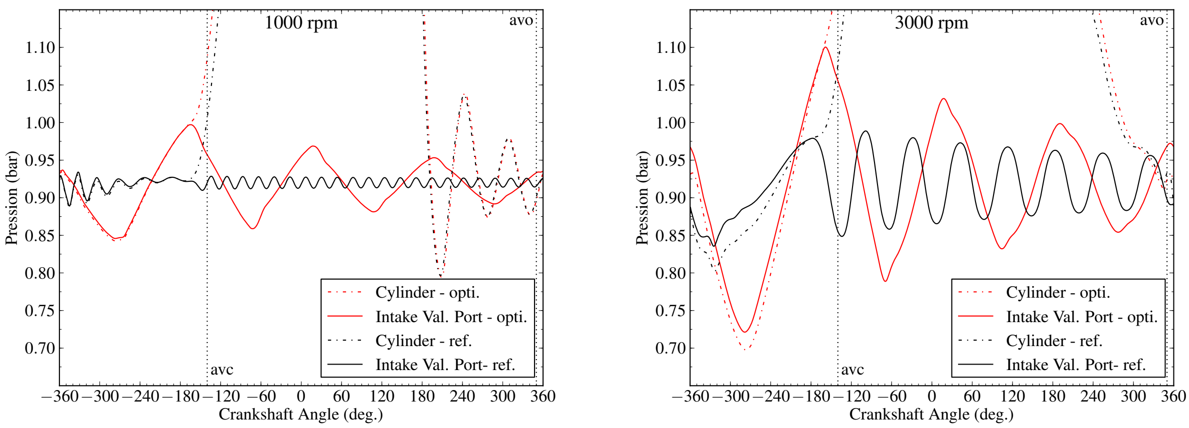

Figure 4. Cylinder and inlet port pressure curve versus rotation speed at 1000 (left side) and 3000 rpm (right side).

In Figure 5 are shown instantaneous curves of air mass flow in the intake valve port, at 1000 rpm (left side) and at 3000 rpm (right side). Note that early admission at 1000 rpm, the case begins with optimized mass flow positive and reference case shows the exit gas inlet valve. In much of the flow curve of the optimized case is higher than in the reference case, except two peaks of mass flow due to two rapid fluctuations in pressure. In this case the volumetric efficiency of a gain of 8.28%. It is also observed a reduction in the reflux inlet valve. The same points seen previously can also be seen in the figure on the right at 3000 rpm. However reflux inlet valve to the optimal case and decreased gain in volumetric efficiency was 9.04%.

Table 3 shows the results of optimization with two design variables. The first is the length of the intake duct, and the second is the length of the exhaust duct. The results shown were also obtained between the rotation speed of 1000 and

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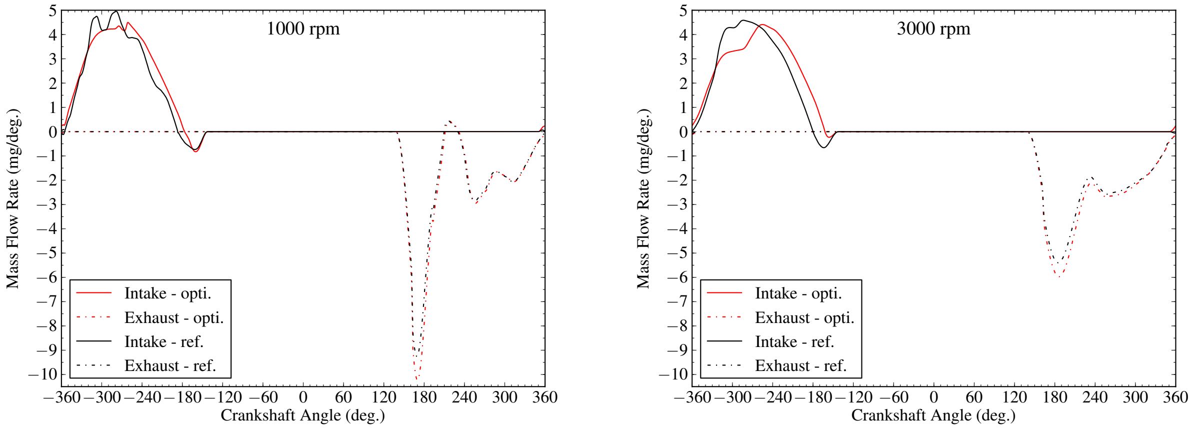

Figure 5. Mass Flow Rate in intake and exhaust port at 1000 (left side) and 3000 rpm (right side).

4000 rpm. Comparing the values in this table with the results of Table 2 it can be noticed that naturally there was an increased volumetric efficiency with the highest gain around 1.4% at 2500 rpm. Note also that the values of the intake duct were found close to the optimal values found in Table 2.

These small differences can be explained by the short period crossover valves (20 degrees) at which fails to generate conditions in the exhaust duct for increasing the mass flow at the intake valve. In any case these results verify that you can change the exhaust lenght duct for a higher volumetric efficiency.

Table 3. Volumetric efficiency optimization results with the admission and exhaust length duct as parameter.

Speed (rpm)	Minimum			Maximum			Mean	Std. Dev.
	η_v	$L_{adm}(m)$	$L_{exh}(m)$	η_v	$L_{adm}(m)$	$L_{exh}(m)$	η_v	η_v
1000	0.831744	2.861986	3.000000	0.832400	2.860515	2.849999	0.832152	126×10^{-6}
1500	0.854695	1.888512	1.792048	0.854959	1.888218	1.782830	0.854731	60×10^{-6}
2000	0.871218	1.402479	2.706448	0.872648	1.403515	1.254813	0.872552	314×10^{-6}
2500	0.883809	1.111032	2.132519	0.885111	1.111754	0.965337	0.884918	461×10^{-6}
3000	0.890971	0.915217	1.736653	0.891729	0.915532	0.783623	0.891629	224×10^{-6}
3500	0.892598	0.770951	1.419878	0.892634	0.772125	0.593017	0.892619	14×10^{-6}
4000	0.889451	0.660618	1.119721	0.889462	0.662030	1.120901	0.889456	2×10^{-6}

It can be see that the standard deviation of the experiments for all results in this paper showed small values, which allows to say that the experiments converge near the point of maximum change. The difference between the maximum and minimum length values of admission duct obtained in all the experiments is a few millimeters.

5. CONCLUSION

In this work we presented a numerical methodology to increase the volumetric efficiency. First, it was showed that the mathematical model used to simulate the thermal processes and fluid flow of a single cylinder IC engine is able to predict the change in performance due to changes in the lengths of the pipes.

The results of optimization done by Differential Evolution method proved capable of finding the global maximum. The standard deviations are small, which indicates that all experiments approached global maximum. The difference in pipes lengths found in each experiment is small. This allows to say that for this specific study, the number of experiments may be reduced and thus the computation time.

It was shown at high speed, the the length of the intake duct must be lower. This observation is completely in line with results reported in literature. This is because the duration intake valve is open decreases with rotation speed and therefore the frequency of the pressure wave must be increased. And this frequency is dependent on the inverse of the length. At higher speeds, the inertia of the gas in the duct increases and it start to affect the shape of the pressure wave. Another significant aspect is to ensure higher values of volumetric efficiency, the pressure wave present in the intake manifold must be in phase with the rate of volume change whitin cylinder, thus allowing increase the amplitude of wave.

It was seen that the length of exhaust duct has a small influence on optimizing the volumetric efficiency, because the overlap is small.

It was observed that optimization of the intake duct is an interesting strategy to increase the income of engines de-

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pending on engine speed. It was noted that this approach reduces the flow of gas through the intake valve, but not totally exclude. In this case, strategies such as VVT (Variable Valve Timing) are more appropriate.

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