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INTAKE AND EXHAUST SYSTEMS PRESSURE WAVES: THE EFFECTS ON ENGINE PERFORMANCE

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Abstract. *The wave phenomena in the intake and exhaust manifolds affect the residual gas fraction inside the cylinder, the volumetric efficiency and the indicated performance of an internal combustion engine. These pressure waves are created in the inlet and exhaust manifolds in response to sudden changes of speed of the gaseous mass in motion due to the pulsed operation of the engine valves. Extensive experimental research has been done to achieve a "tuning" of these pressure waves to increase the engine performance, always taking into account the real geometry of the intake and exhaust manifolds. The objective of this work is to present an analysis of the effect of the pressure waves on the indicated performance of a spark-ignited engine and how a variable valve system can be useful to take advantage of these waves, both for intake and exhaust systems. Differences in the volumetric efficiency can be dramatic when we compare the maximum positive synergy among the pressure waves and the inlet valve timing, with the maximum negative synergy. This is especially important to inlet valves at high engine speed: volumetric efficiency can vary from 0.63 to 0.81. For exhaust valves, the effect is smaller, but yet relevant: volumetric efficiency can vary from 0.79 to 0.815.*

Keywords: *Internal combustion engines, volumetric efficiency, Waves in manifolds, Engine performance simulation*

1. INTRODUCTION

The parameter employed to measure the effectiveness of an engine's induction process is the volumetric efficiency. The volumetric efficiency is defined as the ratio between the mass of air (m_a) sucked by the engine and a reference mass (m_{ref}), usually thought to be equal to the mass of air that could fill the cylinder displacement in the conditions of pressure and temperature present at ambient conditions. Sometimes, this efficiency is referred to the conditions of the inlet port. In this work the prior definition is adopted.

$$\eta_V = \frac{m_a}{m_{ref}} \tag{1}$$

From Eq. (1) it is clear that to maximize the volumetric efficiency it is necessary to increase the amount of air introduced into the cylinder (ma). Intake valves and the induction process are determinants to obtain the best possible volumetric efficiency, and this subject is presented in all internal combustion engines literature, be it on fundamentals (Benson and Whitehouse, 1983; Heywood, 1988), be it dedicated to engine modeling (Ramos, 1989). The intake and exhaust processes are always relevant: for four-stroke and two-stroke engines, spark-ignition and compression ignition engines, naturally aspired of supercharged engines.

The unsteady nature of the fluid flow in the manifolds and valves induces pressure waves both in the intake and exhaust manifolds which affect the flow through valves and consequently the volumetric efficiency. When the intake valve is open, the motion of the piston in the intake stroke creates a depression wave which propagates along the suction duct, with speed equal to the speed of sound. When the wave of depression reaches the outlet section of the duct, or other geometric boundary condition, it is reflected as an overpressure wave which again traverses the suction duct, at this time with a direction concordant with the air current, with the same sonic speed. If this wave of overpressure arrives in the intake region when the valve is still open, it is observed an effect of dynamic supercharging, increasing the amount of air sucked by the engine. Being the wave propagation speed independent of the rotation speed of the engine, it is possible to achieve a positive effect of dynamic supercharging only for some speeds of the engine. However, if a depression wave arrives at the intake region, the effect is the opposite: in this case, the pressure wave does not help the intake stroke, and therefore there is a worsening of the filling process that causes a decrease in the volumetric efficiency.

The determination of the pressure waves in the intake and exhaust manifold can be made by unsteady flow analysis methods. One of the most popular methods of analysis for the study of the manifold waves is the 1-D method of the characteristics. Although known for a long time, Benson (1982) detailed the method, studying the effects of the changes in the intake and exhaust systems geometry on the boundary conditions necessary to the method of the characteristics. Since then, many researchers (Bulaty and Niessner, 1985; De Risi et al, 2000; Varsos, 2010) and also various commercial software for engine simulation adopts this method when the objective is to obtain the pressure in the intake and exhaust ports as a function of crank angle, to evaluate the volumetric efficiency for a given engine speed.

Also a 1-D method, Harrison and Stanev (2004) proposed an acoustic model to study the pressure waves in the intake manifold. With a similar acoustic model, but with emphasis in experimental results, Cavalieri (2014) made a characterization of the waves inside an intake manifold. Chalet et al. (2011) also proposed an alternative model for the manifold waves, based on a mass-spring-damper system, with parameters determined by an experimental apparatus. They were seeking to reduce the computational time associated with numerical solutions for the unsteady flow, but preserving the wavy nature of the pressure in the intake and exhaust processes.

The multi-dimensional methods are adopted when there is the need to obtain the velocity and pressure fields inside the manifolds or to evaluate the turbulence pattern inside the cylinder. Pioneer examples of this kind of studies can be seen in Bicen et al (1985) and Ramos (1989). Since then, there are many 3-D studies made by different researchers and engine manufacturers to define and optimize the inlet and exhaust system geometry. As an example, the PhD Thesis of Montorfano (2012) studied the intake and exhaust processes including a diesel particulate filter (DPF) with a 3-D code. This kind of studies is very time-consuming, due to the nature of the flow: unsteady, turbulent, compressible and viscous. These characteristics make the use of 3-D models suitable to study particular problems associated with the intake and exhaust flow, but not suitable for the global engine simulations.

The use of variable valve timing is an interesting form to increase the volumetric efficiency and can be explored to adjust and optimize the valve timing with the wave phenomena for different engine speeds. The use of camless valve systems (Lie et al, 2008; Mohamed, 2012), associated with electronic control is another important perspective which can help this optimization.

The main goal of this work is to understand the influence of a variable pressure in the intake and exhaust manifolds on the performance of the engine. To this aim, there are also two aspects that greatly affect the volumetric efficiency and explored in this study: the use of a variable valve actuation and the choice of the valve lift model.

2. SIMULATION METHODOLOGY

2.1 Simulation models for valve elevations

The traditional valve lift system is composed by a fixed valve timing camshaft and springs. A usual way to model the conventional valve elevation is through parabolic curves (Sherman and Blumberg, 1977). The adimensional valve elevation (elevation/valve diameter) for a conventional camshaft system can be seen in the Figure 1 for exhaust and intake valves. However, for the new valve lift devices, camless, there is a need of a new valve lift model. The model presented here for a camless system adopts a modified form of the logistic function. For the construction of the exponential profile, the valve position is divided into three parts. The first part of consists of the valve lift and is described by the following equation:

$$y = 1 - exp\left[-a * \left(\frac{\theta-\theta_i}{\Delta\theta}\right)^{M+1}\right] \qquad (2)$$

where $\Delta\theta$ is the duration (in crank angles) of the valve opening period. The choice of this parameter is given by a compromise between the reduction of the stresses on the valve, and the reduction of the opening duration $\Delta\theta$, in degrees, until the maximum lift. The extremes of this function are the zero and the unity. So, to obtain the instantaneous elevation of the valve the logistic function y must be multiplied by the maximum valve elevation. Adjacent to this curve, it follows a part where the valve has already reached its maximum lift and remains full open. The final part of the valve movement is its closure, which also follows the exponential equation symmetric to the Eq.(2). The Fig. 1 also shows the adimensional valve lift profile described by the proposed exponential law. The valve stays for a long period full open; this increases the volumetric efficiency at full load.

2.2 The engine simulation model

The engine was simulated by a thermodynamic model, based on mass and energy conservation, developed by the authors. The ideal gas hypothesis was adopted. The mass and energy balances are presented by Eq. (3) and (4), respectively.

$$\left(\frac{dm}{d\theta}\right)_{CV} = \left(\frac{dm}{d\theta}\right)_{in} - \left(\frac{dm}{d\theta}\right)_{out} \qquad (3)$$

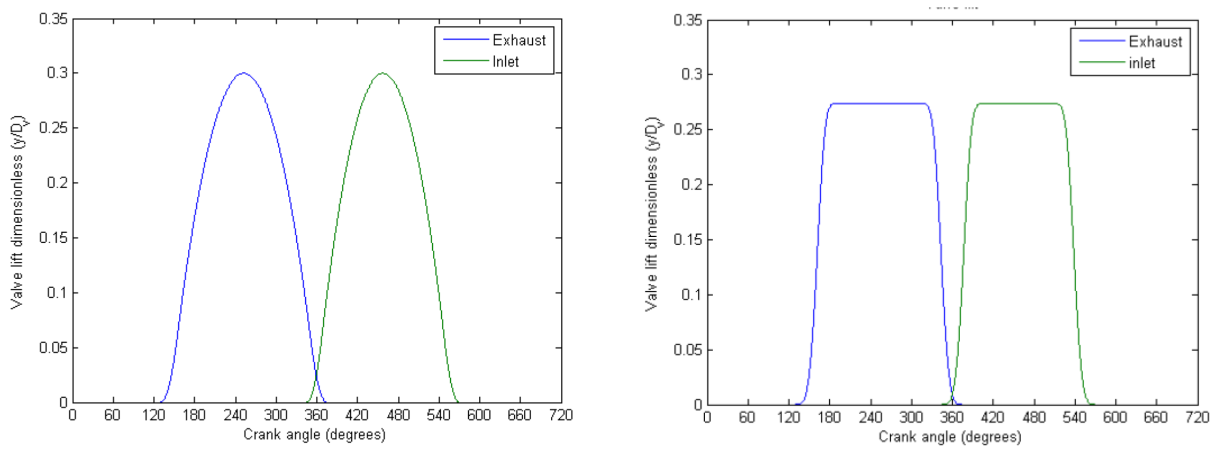

Figure 1 – Adimensional valve elevation, for the parabolic and exponential models.

$$\frac{dQ}{d\theta} - p\frac{dV}{d\theta} = \frac{dU_{CV}}{d\theta} + h_{out}\left(\frac{dm}{d\theta}\right)_{out} - h_{in}\left(\frac{dm}{d\theta}\right)_{in} \tag{4}$$

The term dQ includes the heat transfer process between the gases contained in the cylinder and the walls (cylinder liner, piston head and cylinder head) as well as the energy released by the fuel during the combustion. The heat transfer is evaluated by the conventional Newton equation for heating, with the film coefficient given by the Hohenberg's correlation (1979). The combustion heat rate is given by a Wiebe function (Ghojel, 2010).

For the open part of the cycle, the mass flow through valves was calculated with the "quasi steady-state" model. Assuming an adiabatic and compressible flow in steady-state condition and isentropic, it can be shown that the mass flow is given by the Eq. (5):

$$\frac{dm}{dt} = A.P_1.\sqrt{\frac{2}{RT_1}.\frac{\gamma}{\gamma-1}.\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}}.\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \tag{5}$$

where A is the throat area, and subscripts 1 and 2 define the conditions upstream and downstream, respectively. The "quasi steady-state" model adopts this same equation, but with instantaneous values for each variable. The Eq. (5) is valid only when the Mach number is below the unity. Otherwise, the choke flow condition appears. The choke condition prevails when:

$$\frac{p_2}{p_1} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \tag{6}$$

then, the mass flow associated with the choke condition is given by:

$$\frac{dm}{dt} = A.P_1.\sqrt{\frac{2}{RT_1}.\frac{\gamma}{\gamma+1}.\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{(\gamma-1)}}} \tag{7}$$

To correct the Eq. (7) for irreversibilities, an experimental discharge coefficient (as a function of the adimensional valve elevation) was introduced by Kastner et al. (1964).

The Table 1 presents the engine model parameters adopted as base case in this study. Some of these parameters are geometric, some are operational. The engine is a four-stroke single cylinder, spark-ignition. The fuel is gasoline. The characteristics of the engine are the same used in other studies (Damasceno and Gallo, 2013; Gallo, 1992), to preserve the possibility of direct comparisons among them.

Figure 2 shows results for the mass flow through exhaust and inlet valves, as well as for the mass trapped in the cylinder, for a given engine speed, for constant and mean pressures in the intake and exhaust manifolds, that is, without the consideration of the pressure waves in the intake and exhaust processes. As can be seen in the Fig.(2), the exhaust flow presents a big mass flow in the blow-down period, with sonic flow in the exhaust valve. The inlet flow presents a backflow at its beginning, due to the unbalance among the pressure inside the cylinder and the intake manifold pressure. The mass trapped into the cylinder can also be seen in the Fig. (2b).

Table 1 – Parameters of the modeled engine

Parameters	Values	Parameters	Values
Cylinder diameter	80.0 mm	Combustion duration	60º CA
Piston stroke	79.5 mm	Combustion begins at	340ºCA
Connecting rod length	128 mm	Intake valve diameter	30.93 mm
Compression ratio	8	Exhaust valve diameter	28.27 mm
Fuel heating value	43.4 MJ/kg	Intake valve stem diameter	7.733 mm
Air fuel ratio (by mass)	15	Exhaust valve stem diameter	7.068 mm
Atmospheric pressure	1.0133 bar	Cylinder wall temperature	520 K
Atmospheric temperature	25º C		

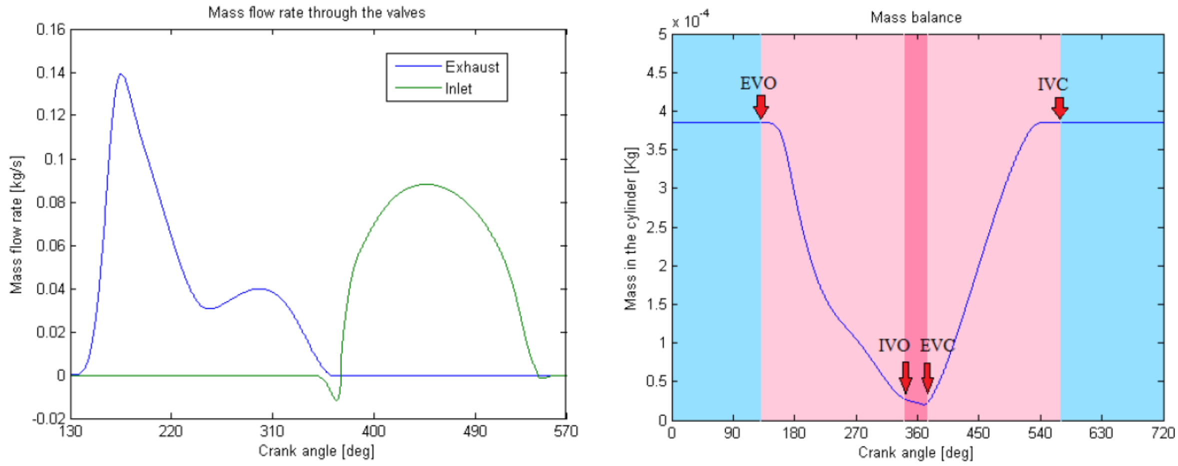

Figure 2 – a) Exhaust and intake mass flow for 2800rpm. b) Mass trapped in the cylinder. Model with constant intake and exhaust pressures in manifolds.

2.3 The pressure wave model

The determination of the instantaneous pressure acting in the intake and exhaust manifold can be obtained either from experimental measures, or by a 1-D calculation method. However, in both cases they depend on the actual geometry of the intake and exhaust systems: diameters, curves, manifold branching, plenums and so on. Since we are modeling a hypothetical engine, there is no way to obtain the pressure profile in the inlet of the intake valve, or in the exit of the exhaust valve. To model the pressure waves, it was assumed a typical pressure pattern shown in the Fig. (3). The wave profile is periodic, but attenuated. The same profile was imposed for the intake and exhaust valves, but centered in the mean value of the pressures for intake manifold (0.86 bar) and exhaust manifold (1.15 bar). To explore the effect of this wave profile at different engine speed, the amplitude of the waves were increased by a scale factor as the engine rpm increased, as is verified from different studies (Heywood, 1988).

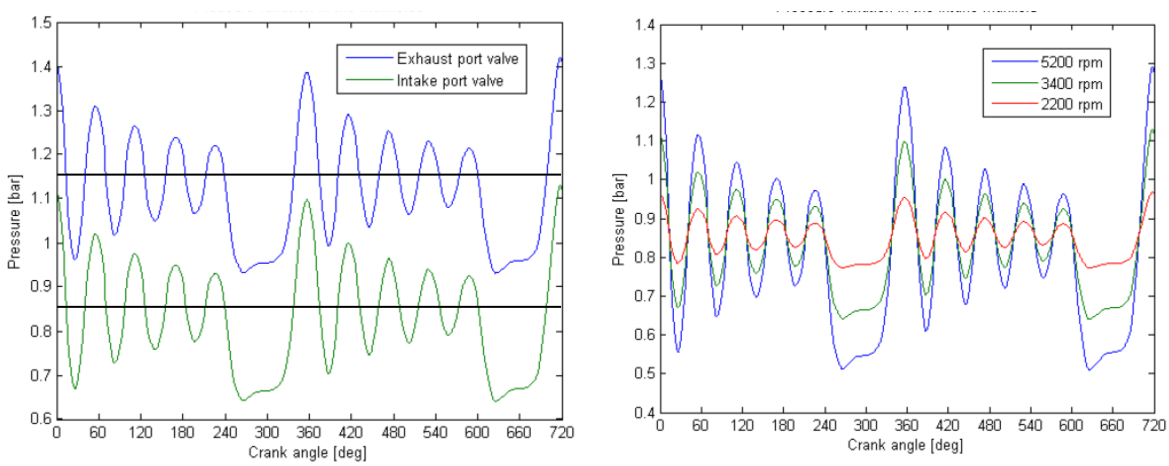

Figure 3 – a) Wave pattern for exhaust and intake ports. The wave oscillates around the mean exhaust or intake pressures. b) Effect of the engine speed on the wave amplitude.

With this wave pattern it is possible verify how much the volumetric efficiency is affected by the presence of the waves, and compare it with the volumetric efficiency for constant pressures at manifolds. Rather than determine the real volumetric efficiency for a given wave pattern, the objective is to obtain how great this effect can be. Since the position of the peaks of the wave can increase or decrease the volumetric efficiency, the wave profile was moved for the right side and to the left side of the it's initial position relative to the crank angle, to promote positives and negatives synergies and quantify the variation of the volumetric efficiency.

3. ANALISYS OF THE RESULTS

3.1 Comparison: performance of the engine with and without pressure waves in the intake and exhaust ports

Figure 4 presents the indicated diagram for the simulated engine, with an exponential-type valve elevation optimized for fixed pressures at intake and exhaust manifolds. The same mean intake and exhaust pressures at 3400 rpm were adopted for both simulations – with waves (variable pressure) or without it (fixed pressure). The very small difference in the cylinder pressure in the beginning of the compression process is responsible for an increase of near 2 bar in the maximum pressure of combustion, and the effect of the intake port wave in this case gives an increase in the volumetric efficiency and engine indicated power. It is interesting to observe the wavy nature of the pressure in the cylinder for exhaust and intake strokes as compared with the pressure inside the cylinder for constant manifold pressures. In this particular case the pumping work was decreased, also contributing to an increase in the indicated power.

To analyze effects of the intake and exhaust waves with variable engine speeds, this parameter was changed from 2200 to 5200 rpm and the results can be seen in the Figure 5. The wave pattern was maintained at the same crank angle position for intake and exhaust ports, although this would not be the case in an actual engine. The objective was to analyze the sole effect of the engine speed with a fixed wave pattern and position.

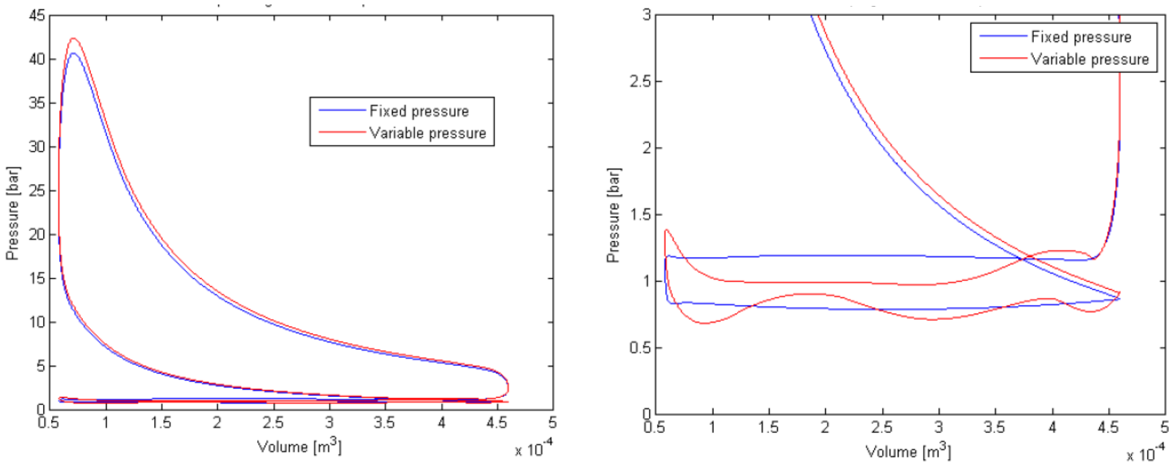

Figure 4 – a) Indicated diagram with fixed or variable pressures. b) Low pressure end of the indicated diagram

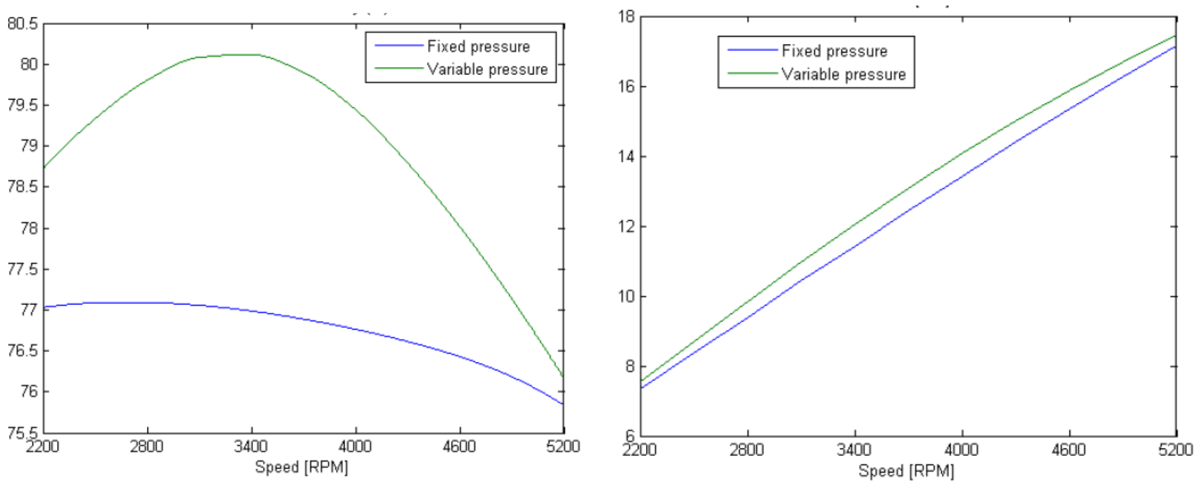

Figure 5 – a) Volumetric efficiencies as a function of engine speed. b) Indicated power [kW].

The position of the waves related to the crank angle seems to present an optimum value of volumetric efficiency for an engine speed near 3300 and 3400 rpm. For low engine speeds, the difference of the fixed pressure model and the wave model in the manifolds is somewhat reduced; however, for high engine speeds the difference is quite small. This shows that the variable valve mechanism is not "tuned" with the wave effects and could be modified to increase the volumetric efficiency and indicated power. The absolute value of the volumetric efficiency varies some 3,9% in the best case (3400 rpm) and 0,4% in the worse case (5200 rpm).

3.2 Effect of the phase shift of the wave: intake port.

Figure 6 shows what happens if the intake port wave pattern is phase-shifted in relation to crank angle, for the engine speed of 3400 rpm. For example, this could be accomplished by a variable length intake runner (variable geometry intake system). The phase shift shows that the original position was close to the optimum. The first part of Fig.(6) shows the volumetric efficiency values, and the second part shows the percent losses or gains compared with the base case (position zero crank angle in the Fig. 6). The losses can be as high as 25% if the intake system produces a wave which has negative synergy with the valve opening. The same analysis was made for a low engine speed (2200 rpm) and for a high engine speed (5200 rpm), and results are shown in the Fig. (7). The higher the engine speed, the higher are the losses if the intake system is not "tuned".

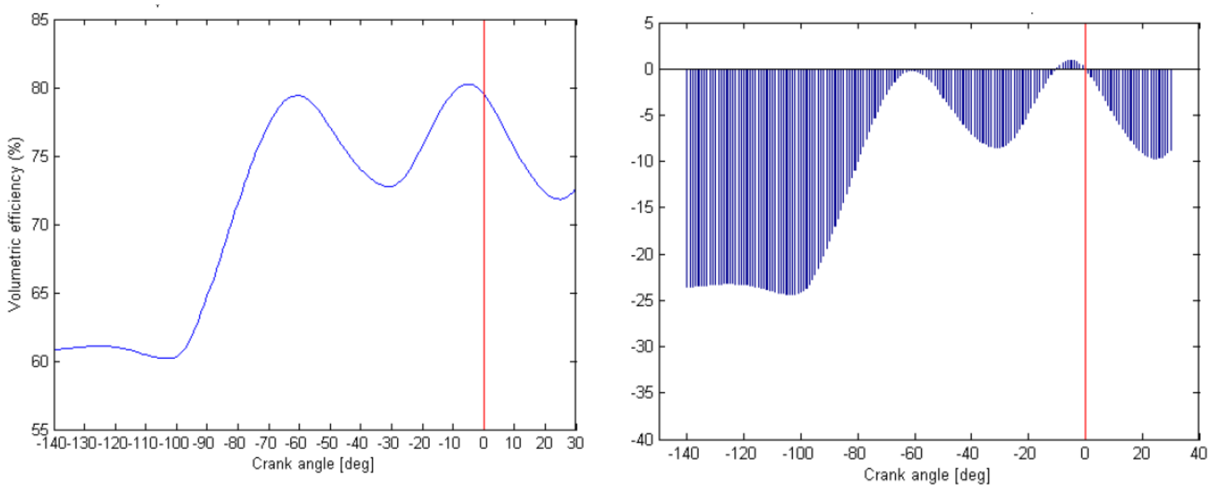

Figure 6 – a) Volumetric efficiencies as a function wave phase shift for 3400 rpm, intake manifold. b) Percent gains or losses in volumetric efficiency from the base case (zero).

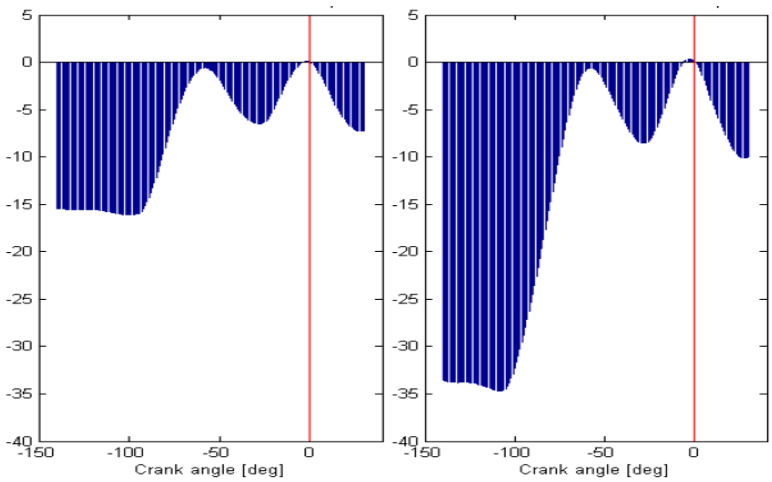

Figure 7 – a) Intake manifold: percent gains and losses of volumetric efficiency as a function wave phase shift for 2200 rpm. b) Percent gains or losses for 5200 rpm

3.3 Effect of the phase shift in the wave: exhaust port.

The same analysis made for the waves in the intake port was repeated for the exhaust port. Results are in the Fig.8.

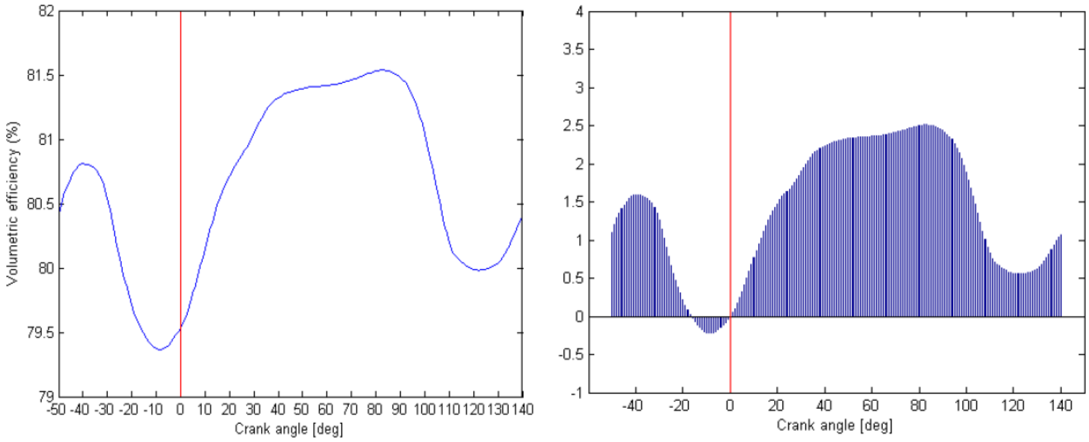

Figure 8 – a) Volumetric efficiencies as a function wave phase shift for 3400 rpm, exhaust manifold. b) Percent gains or losses in volumetric efficiency from the base case (zero).

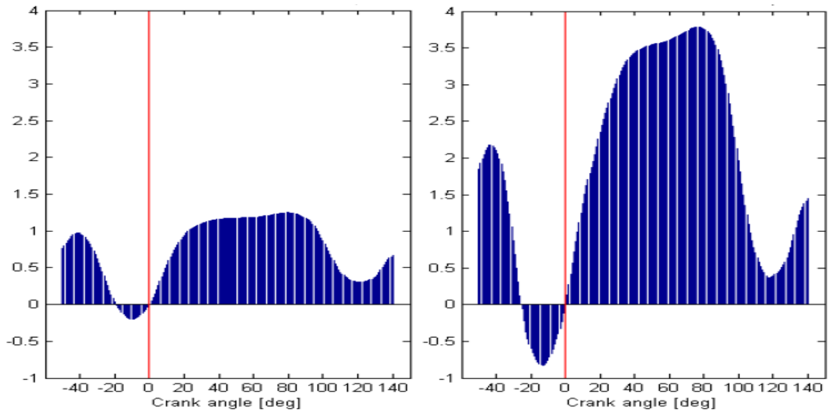

Figure 9 – a) Exhaust manifold: percent gains and losses of volumetric efficiency as a function wave phase shift for 2200 rpm. b) Percent gains or losses for 5200 rpm

Figure 9 shows the results for 2200 and 5200 rpm. Contrary to the observed in with the phase shift of the waves in the intake port, for the exhaust port the position of the waves in the base case were detrimental to the engine performance. So, the gains to be obtained – for any speed – are high. The exhaust system is not "tuned" in the base case (the zero in the Figures 8 and 9). To increase the engine performance, modifications should be made in the exhaust system to increase the volumetric efficiency. However, the gain is not so spectacular, ranging from near 1% (2200 rpm) to a maximum of 3.8% (5200 rpm).

3.4 The combined effect of the wave phase-shift: intake and exhaust ports

The Figure 10 shows, for the engine speed of 3400 rpm, the combined effect of the waves in the intake and exhaust port which produces the best and the worst volumetric efficiencies. Intake and exhaust systems not tuned with the engine speed can present very high penalties in the volumetric efficiency and, consequently, in the engine performance. The worse case shows (blue lines, volumetric efficiency of 0,6008) where the waves are as related with the valve events. On the other side, if the intake and exhaust systems are tuned for a given engine speed (red lines), the volumetric efficiency is much higher (maximum value of 0.8195).

4. CONCLUDING REMARKS

The relation that links the variation of pressure in the inlet and exhaust manifolds to the volumetric efficiency was already known, but this study quantified how great this influence can be. The simulation model permits to fix a determined shape and vary some other parameter, which is not possible in an actual engine. As expected, the intake system has greater influence on the volumetric efficiency than the exhaust system.

To "tune" the valve events with the waves in the intake and exhaust ports, a variable camless valve system can play an interesting role, increasing the volumetric efficiency and the indicated power in a wide range of engine speeds.

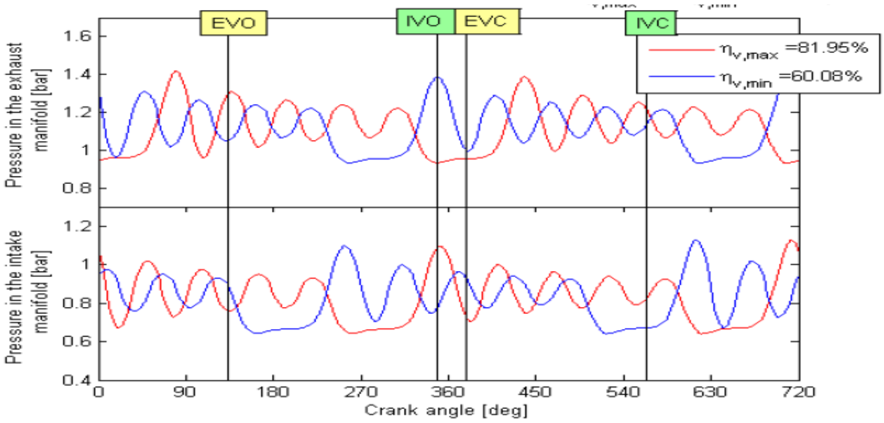

Figure 10 – Best (red) and worst (blue) wave phase shift in the intake and exhaust systems for 3400 rpm

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