Numerical and experimental analysis of the Fluid Flow in Two Intake Pipes with a Junction

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Abstract. *The project of engine intake systems involves optimization of parameters such as the pipe length and diameter, junctions, accessories and the intake and exhaust valves opening and closing timings. The correct sizing leads to an increase of the air mass admitted by the cylinders at the desired engine operational conditions. In the present work, pressure variations caused by the valve movement were investigated numerical and experimentally considering an intake system composed by two intake pipes, a junction and a single duct connecting the junction to the atmosphere. The objective was to study the flow characteristics at varying junction position in the intake system. For this purpose, curves of mass flow rate and the dynamic pressure at several locations of the intake system were presented. The results shown that the junction location in the intake system affects the air mass flow rate.*

Keywords*: Gas Dynamics, Intake System, Pressure Wave, Internal Combustion Engine.*

1. INTRODUCTION

The elementary function of the intake system is to improve breathing capacity of the engine, while keeping pressure losses to a minimum. The basic idea is that the mass flow must be distributed equally to all the cylinders, which imposes a fluid dynamic symmetry layout for the system. Since the flow is periodically interrupted by the valves, it is clear that each portion of the engine cycle is accompanied by acoustical vibrations of the gas concerned. A large proportion of the total noise generated by the engine is due to the pressure waves that propagate from the intake and exhaust systems.

Due to the piston and valve movements, the core gas inside the intake system presents an oscillating pressure. The transient pressure caused by the valves and pistons can be used to improve cylinder charging, through optimization of the intake system. The geometry of the intake manifold has an effect on the frequency and amplitude of the pressure waves. The intake system components are often designed to improve the volumetric efficiency and attenuate the amplitude of the pressure waves on a specific frequency. The challenge is to obtain the desired radiated noise spectrum without producing deleterious effects on the management of the waves which enhance the engine performance. Tuning of the intake manifold is a difficult task due to the periodically pulsating gas flow and to the practical difficulties associated with the measurement of the unsteady flow field in the intake system (Hanriot, 2006).

2. LITERATURE SURVEY

Developing an efficient powertrain and engine management can be very challenging task. The number of variables and their inherent nonlinear and interactive properties makes the solution to the powertrain development problem very complex and nontrivial (Dvorak, 2003). Designers have long been aware that the geometry of the intake system substantially affects reciprocating internal combustion engines performance.

This is because the air charge to the cylinder is a function of the frequency of the pressure pulses in the intake system. Intake systems can be ‘tuned’ to give improved cylinder charging at a particular engine speed and variable geometry manifolds exploit this phenomenon to enhance the engine volumetric efficiency across the whole operational speed range. Considering the nature of the induction process, the basic tuning mechanism harnesses spatial and temporal variations in the manifold flow dynamics to increase the pressure at the inlet valve in the critical period around bottom-dead-center of the induction stroke (Winterbone, 1999, 2000). It is therefore essential that techniques that capture unsteady flow effects be employed in the design of tuned intake systems.

Morse et al (1938) were one of the first to study the pulsating pressure in intake manifolds. They studied the vibrations in the intake system of a four-stoke, single cylinder gasoline engine. They showed that the periodically interruption from the gas flow by the valves sets up standing waves in the intake manifold.

Winterbone (1990) showed a comparison between calculated and measured pressure and air mass flow through a family of intake manifold geometries. A non-linear wave action calculation technique based on the method of characteristics was employed, to predict variation of pressure in the manifold over a broad range of engine speed. No comparisons of mass flows were given, and two different modes of tuning were presented: organ pipe tuning and Helmholtz resonator effects. It was shown that these tunings occur during the periods when the inlet valves are closed and open, respectively. Both tuning modes are important, but the Helmholtz resonator seemed to have a bigger influence on volumetric efficiency due to its effect on cylinder pressure at the time of intake valve closing.

Benajes (1997) presented a pre-design model for intake manifolds in internal combustion engines. The model was based on the acoustic-wave theory, and was capable of calculating the overall dimensions of an optimum intake manifold, with the aim of improving the gas exchange process in the engine. It was shown that the place where the pressure pulses are reflected is important to the design of the intake systems.

Curves and junctions are placed where the reflected pulses can exist. The reflected pulses are in part superposed by the primitive pulses, and the resultant pressure pulse is a combination of these two pulses. The pressure inside the pipe can be divided into two components: one called primitive pulse and another called reflected pulse. The sum of these two pulses produces the instantaneous pressure that effectively exists in the intake pipe. One of the factors that affect the mass flow through the cylinder is the junctions presented at the inlet manifolds. This means that an engine cylinder is very susceptible to interference from other cylinders, which can disrupt the gas flow through them, resulting in poor scavenge and backflow into the inlet and causing fouling.

In all flows in junctions there is a pressure loss. Sometimes that loss is very small and may be neglected, while at other times it can affect the calculated results. Deciding whether there will be a large effect is difficult in marginal cases, but, as a guideline, pressure losses should be included in high-speed engines where the gas velocities are high. The designer hampered by a lack of generalized data and it is often necessary to test the flow at each single junction, sometimes cutting the manifold into sections to allow the problem to be discretized, but it is necessary to observe that the effects from other branches can be important. In fact, the engine designer has three possible sources to obtain the data:

- Steady flow tests can be performed on the junction;
- Previous experimental results from literature;
- An empirical or analytical expression can be used to estimate the loss coefficients.

The steady flow pressure loss coefficients for a junction are usually established experimentally. This is a time-consuming process, which requires the junction to be manufactured before its loss characteristics can be measured. Unfortunately there is scant junction pressure loss data available in the literature, and some of these data are only applicable to junctions of certain types and generally cover only four of the six possible flow types (Winterbone, 2000). It is convenient to classify junctions into two distinct types: ‘T’ junctions and ‘Y’ junctions. ‘T’ junctions consist of a straight duct of uniform cross-sectional area, intersected by a lateral branch, inclined at an angle to the main duct. The 90^o equal area ‘T’ junction is a sub-test of the general ‘T’ junction. ‘Y’ junctions are formed by a main duct, which bifurcates into two side ducts that intersect the main duct at the same angle. Winterbone (2000) describes how the measurement of the junction pressure loss coefficients has to be made.

In this paper, the effects produced by a 20^o ‘Y’ junction between branches were studied. The main objective was to verify the influence of the location of the junctions in the intake manifold at oscillating pressure and the mass curve rate versus the engine rotational speed. The experiments were conducted in a flow rig, as described in the next section.

3. PULSATING PHENOMENA IN INTAKE MANIFOLD

The dynamic flow process along the intake manifold can be characterizes by the instantaneous evolution of the pressure at a point downstream from and close to the valve. In this point, the pressure is a consequence of the proper emptying process and the dynamic response of the manifold. In a intake system the dynamic interaction between originated pressure pulses from the intake valves and reflected pulses from the different boundaries is complex.

Benajes (1997) showed that the case of a simple pipe open to the atmosphere is relevant for single or multi-cylinder engines with independent intake pipes, either because they induce air from the atmosphere or from a very large plenum, where pressure is unaffected by the engine operation. In this case, the solution in terms of frequency of the first harmonic is:

$$f = \frac{a}{4L} \tag{1}$$

where $f=\omega/2\pi$ and ω is the angular frequency, L is the effective length of the pipe and a is the speed of sound.

Benajes (1997, 1998) showed a model for the case of four intake manifold (four primary pipes, an intermediate chamber and one secondary pipe). The system for this configuration can be reduced to a single implicit equation:

$$4\tan\frac{w.L_1}{a} = \frac{A_2}{A_1}\cot\frac{w.L_2}{a} - \frac{w.V}{a.A_1} \tag{2}$$

where ω is the angular frequency, a is the speed of sound, L_1 is the primary length, L_2 is the secondary length, A_1 is the cross section area from primary pipe, A_2 is the cross section area from secondary pipe and V is the volume of intermediate chamber between primary and secondary pipes.

The angular frequency solution of this equation is not explicit and it can be obtained by an iterative calculation introducing the values of the manifold dimensions. The numerical value of w represents the natural frequency of the whole intake manifold, from the valves to the open extreme of the manifold. A similar interpretation was given by Ohata and Ishita (1982). In the more general case of a manifold with Z_1 primary pipes and Z_2 secondary pipes, equation (2) is written as:

$$Z_1\tan\frac{wL_1}{a} = Z_2\frac{A_2}{A_1}\cot\frac{wL_2}{a} - \frac{wV}{aA_1} \tag{3}$$

where Z_1 is the number of primary pipes (before junction) and Z_2 is the number of secondary pipes (after junction);

Aside from the dynamic response of the whole manifold, a pressure reflection phenomenon at the junction takes place similar to that described for the single-cylinder engine. This means that in such engines two natural frequencies can be exploited for tuning purposes: the natural frequency of the complete intake system, obtained from equation (2), and the higher natural frequency of the primary pipe, given by the equation (1).

Theoretical studies made by Morse (1938) developed a parameter called "Frequency Parameter" Q, which relates the natural frequency of the intake system to the frequency of the engine:

$$Q = \frac{f_{system}}{f_{engine}} \tag{4}$$

Morse (1938) showed that optimum system manifolds should feature a dynamic response equivalent to producing a Q value in the range between 3 and 5. Results published by Ohata and Ishita (1982) allowed deriving a linear expression for the optimum value of Q in terms of the engine speed, N (in rpm):

$$Q_{optimum} = 2{,}7 + \frac{N}{3333} \tag{5}$$

Which yields values between 3 and 4.5, if N varies from 1000 to 6000 rpm.

4. ANALITICAL METODOLOGY AND NUMERCAL SIMULATION

The equations used to evaluate the unsteady flow of gas in engine manifolds are showed in this item. The analytical methodology concentrates on techniques that are based on the equations of one-dimensional inviscid fluid flow, multi-dimensional viscous models.

The problem of calculating unsteady compressible flows in the manifolds and pipes of engines can be sub-divided into two parts: the flow in the pipes themselves, and the flow interactions at the boundaries.

The governing equations, which relate to the control volume, for the one-dimensional flow of compressible fluid in a pipe with area variation F, wall friction f, and heat transfer q are (Winterbone, 1999, 2000):

$$\frac{\partial V}{\partial t} + A\frac{\partial V}{\partial x} + C = 0 \tag{6}$$

where:

$$V = \begin{bmatrix}\rho \\ u \\ p\end{bmatrix},\ A = \begin{bmatrix} u & \rho & 0 \\ 0 & u & \frac{1}{\rho} \\ 0 & a^2\rho & u \end{bmatrix} \qquad C = \begin{bmatrix}\rho u \\ 0 \\ a^2\rho u\end{bmatrix}\frac{1}{F}\frac{dF}{dx} + \begin{bmatrix} 0 \\ G \\ (\kappa-1)\rho(q+uG)\end{bmatrix} \tag{7}$$

where x is the distance, t the time, p the pressure, u the flow velocity, ρ the density and a the speed of sound. F is the cross section and d the pipe diameter and q the rate of heat transfer per unit mass.

The term G presented into eq. (7) is defined as:

$$G = 4 f_w u|u| / 2d \tag{8}$$

where f_w is the wall friction coefficient and the term $u.|u|$ is used to ensure that the pipe wall friction always opposes the fluid motion.

Rearranging the equations to Method of Characteristics (Winterbone, 1999), the equations (7) can be transformed from partial differential equations with respect to space and time, into ordinary differential equations in time by considering lines in the space-time plane defined by:

$$dx/dt = u \pm a \tag{9}$$

and

$$dx/dt = u\ . \tag{10}$$

4.1 – Numerical Simulation

Aside the analysis and description of the pulsating pressure interactions in the manifold, the numerical study has been based by a calculation code. This program is able to describe numerically the gas flow at intake manifold under a pressure fluctuation. The model needs the input experimental conditions, such as the number of cylinders and valves, speed of the engine and camshaft, the ambient temperature and pressure, geometric lengths of the intake system and the difference of pressure between atmosphere and the reservoir.

The numerical simulation to study pressure fluctuations and unsteady flow uses the Method of Characteristics, which is a mathematical technique to solve hyperbolic partial differential equations. The equations are solved numerically using a rectangular mesh in time and space.

The gas pressure, density, temperature and flow speed can be determined from the conservation, continuity and momentum equations, conjugated with the ideal gas equation. The initial and boundary conditions were obtained from the experimental data. The intake system was divided into individual tube sections that are connected through junctions. The mesh is specified for each tube section between junctions.

Initially, a conduit containing a gas at atmospheric pressure and temperature is considered. The initial state of the gas is considered the reference state, and represents the stationary condition in which the fluid perturbation propagates inside the pipe.

Numerical simulations were done to verify the flow characteristics under a junction presence. The software was developed at Fiat Research Center, in Italy. The methodology considered the valves shifted from 180^o.

The most important input conditions for the software are the atmospheric pressure, temperature, valve profile, intake system geometry, number of valves per cylinder, engine rotational speed and wall friction factor.

The numerical data used for the software input are showed in Table 1.

Table 1 - Input data for the software

Input data for the software	
Atmospheric Pressure (bar)	0.920
Atmospheric Temperature (K)	294
Pressure Difference between atmosphere and the plenum (bar)	0.32
Pipe diameter before the junction / after junction (mm)	25/25
Wall friction factor	0.01
Intake head loss	1

5. EXPERIMENTAL SET-UP AND PROCEDURES

The flow bench (Fig. 1) is an apparatus used for gas flow studies in the intake and exhaust systems of internal combustion engines, under steady or unsteady conditions. The apparatus allows for flow rate measurements through the intake or exhaust pipe, with the valve in movement, at a constant pressure drop throughout the system (Hanriot, 2001).

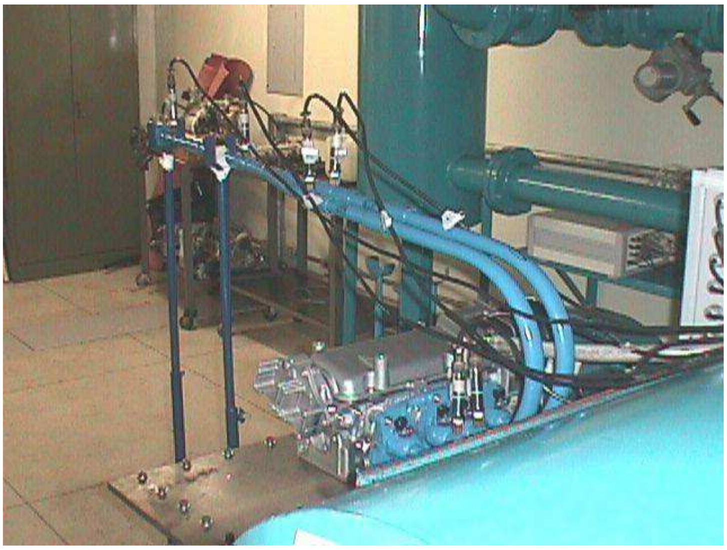


Figure 1 - View of the experimental set-up.

The constant pressure drop is obtained through a big reservoir to which the intake system is connected, for equalization of the pressure at that point. At the other end, the reservoir is connected to a blower, which produces a constant pressure difference between the atmosphere and the reservoir. The blower works at constant rotational speed, and the pressure drop is obtained through the valves between the blower and the reservoir. The reservoir has a volume of around 350 liters, and it eliminates pressure pulsation originated from the valve movement. The valves are moved through an electric motor, which rotational speed is adjusted through a frequency converter. The electric motor used a maximum power of 30 kW and a maximum speed of 3500 rpm.

Two laminar flow meters are used to determine the mass flow rate. The instantaneous pressure is measured by piezoresistive pressure transducers, with a working range of ± 1 bar. The temperature sensors are of the platinum resistance type, for use between 0 and 60 ^{o}C. A four-cylinder, 1.0 liter engine cylinder head was fixed to the dumping tank. Only the intake valves of the second and third cylinders were operating, while the other valves remained closed. A 22.30 mm internal diameter straight tube containing a junction was connected to the valve ports. The steel tube plays the role of the intake system.

Six pressure transducers were used, distributed along the intake pipe. Transducers named P1 and P2 were located at the nearest position to the valves of cylinders two and three, respectively. The others transducers were connected as shown in Fig. 2. Five junction's positions were tested. The configurations were called *`A`,`B`,`C`,`H` and `J`*. The configurations were divided into L1, L2 and L3 lengths. Table 2 shows the lengths of the configurations tested.

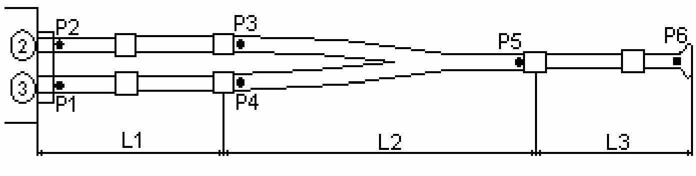


Figure 2 - Schematic drawing of the transducers and the intake system.

Table 2 – Configurations Tested (mm)

Configuration	L1	L2	L3	Total Length	(*)Natural frequency (Eq.1) (Hz)	(**)Natural frequency (Eq.1) (Hz)	Natural frequency (Eq. 3) (Hz)
A	468	315	331	1114	76	142	59.8
B	468	315	830	1613	52.6	142	41.9
C	468	315	1332	2115	40	142	32.6
H	1010	315	830	2155	40	74.6	32.9
J	1509	315	331	2155	40	51.9	29.4

(*) Total length; (**) Primary length

The schematic drawing of the configurations tested are showed in Fig. 3. One of the set of the configurations tested (A, B and C) has the same junction position, but the primary pipe intake system changes. In the second series (C, H and J) the intake system has the same total length, but the junction position changes.

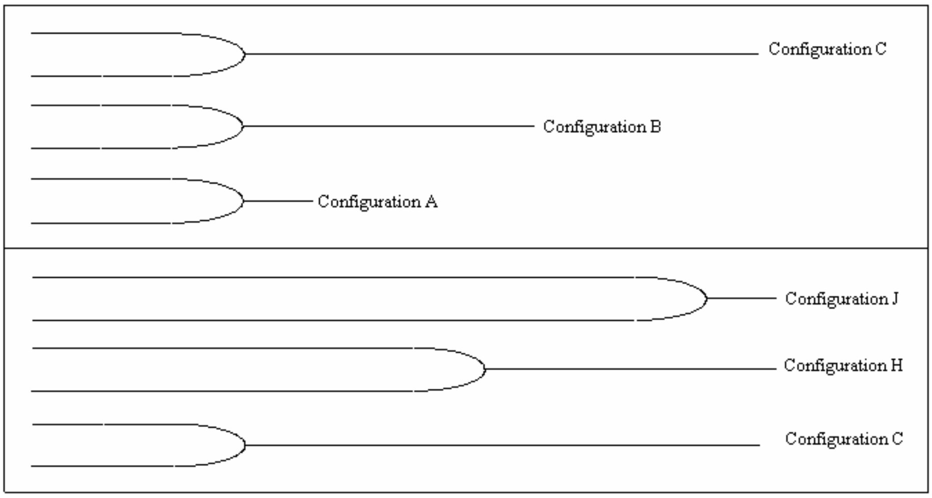


Figure 3 – Schematic drawing of the configurations tested

The tests carried out allowed for the data acquisition of the following parameters:

- pressure wave data throughout time, including the location at the valve ports; and
- mass flow rate through the camshaft revolution.

The acquired data was recorded for each single cycle of a four-stroke engine, corresponding to 720 camshaft degrees.

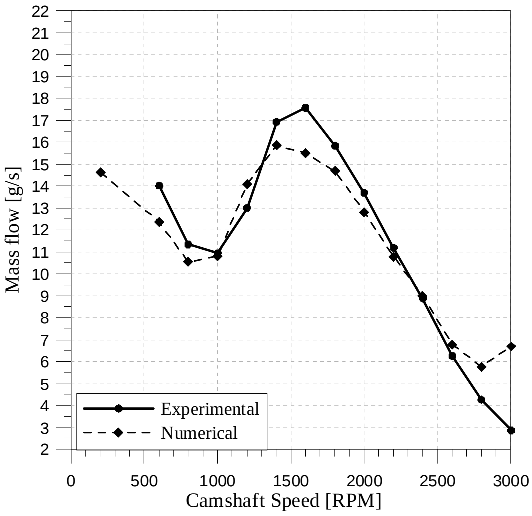
An inductive rotational speed sensor was connected to the camshaft, allowing the acquisition of pressure data per cycle. The experiments were carried out for camshaft rotational speeds in the range from 200 to 3000 rpm, with 200 rpm steps.

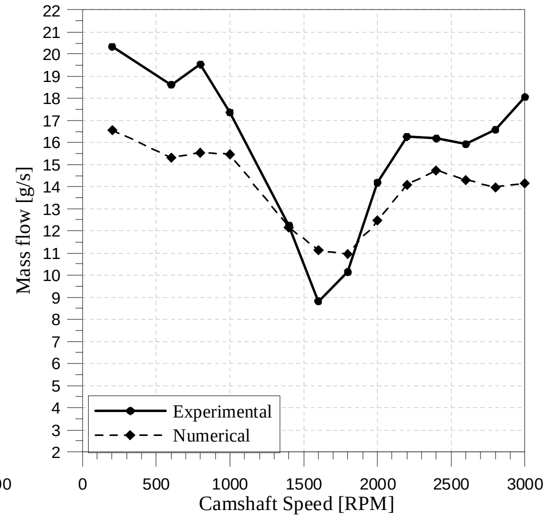
6. RESULTS AND DISCUSSION

In the experiments, the room temperature was 294 K and the pressure drop between the room and dumping tank (plenum) was 0.32 bar. The tank pressure was bellow the atmospheric pressure and virtually constant during the tests.

Figure 4 shows the mass flow rate through the intake system versus camshaft revolution for the configurations A and J. Both experimental and numerical results are shown. These configurations were chosen to analyze and validate the numerical simulation and because they represent the locations were the junction is closer to the valve port (configuration A as example) and longest with it (configuration J).

It is apparent that the simulated curve reproduces very closely the shape of the measured data along the whole range of speeds.

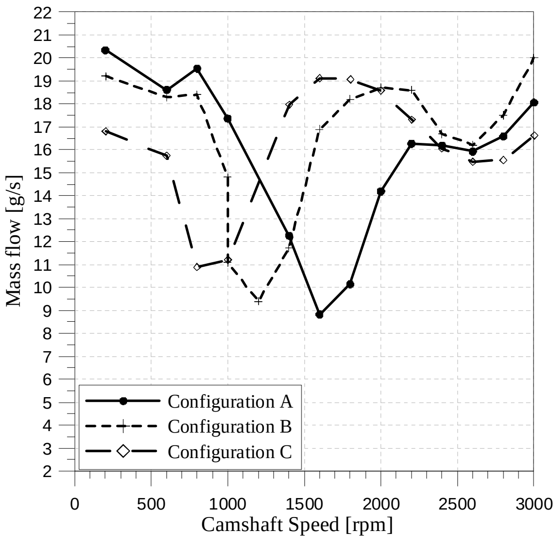


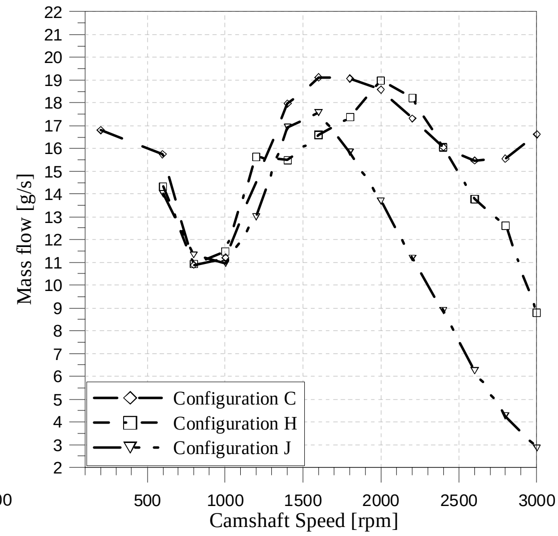


(a) Configuration J (b) Configuration A

Figure 4 – Mass Flow Rate versus Camshaft Revolution

Figure 5 shows the experimental mass flow rate versus camshaft revolution for all the configurations tested. In Fig. 5 (a), the junction has the same position closer to the valve port (according Fig. 3 and Table 2), but with the primary intake pipe with different lengths. In fig. 5 (b), the intake pipe has the same total length, but the junction has different positions. It can be seen that the junction position and the total length affect the mass flow rate with the camshaft speed.





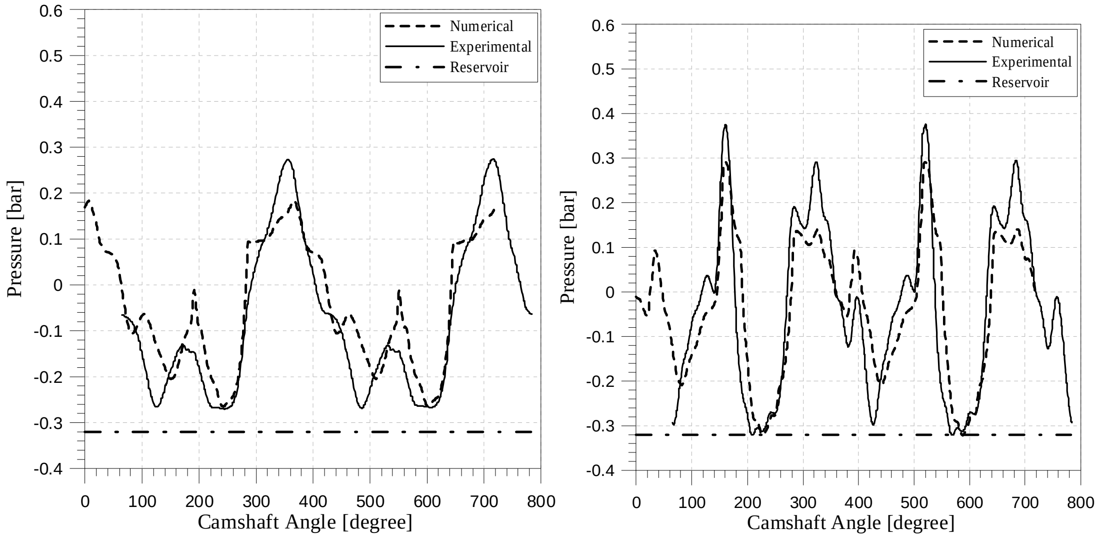
(a) Configurations A, B and C (b) Configurations C, H and J

Figure 5 –Mass Flow Rate versus Camshaft Revolution

In Fig. 5 (a), it can be seen that the mass flow rate versus camshaft speed of the tree configurations are different until around 2400 rpm. After this speed, the configurations are very similar and follow the same behavior. As an explanation for this situation, it can be noticed that the primary intake length doesn't changed for the tree configurations. Thus, the natural frequencies (according Eq.1) of the system are the same. On the other hand, for the camshaft speed lower than 2400 rpm, it can be observed how minimum mass flow rate has been shifted towards lower engine speed values, clearly related with the different lengths of the intake pipe system.

Figure 5 (b) shows the mass flow rate versus camshaft revolution for the configurations C, H and J. This configurations have the same total length, but with different positions where the junctions is. It can be seen that the behavior of the tree configurations are very similar until 1400 rpm and the minimum mass flow rate are the same for all configurations (around 900 rpm). After this speed, the configurations are different and can be observed that the maximum mass flow rate has been shifted tower the higher engine speeds values.

Figure 6 presents the experimental and numerical curves of pressure variation along camshaft angle at 1000 and 1600 rpm for configuration J at the valve port. The results exhibit a very similar behavior between them and validate the numerical simulation.

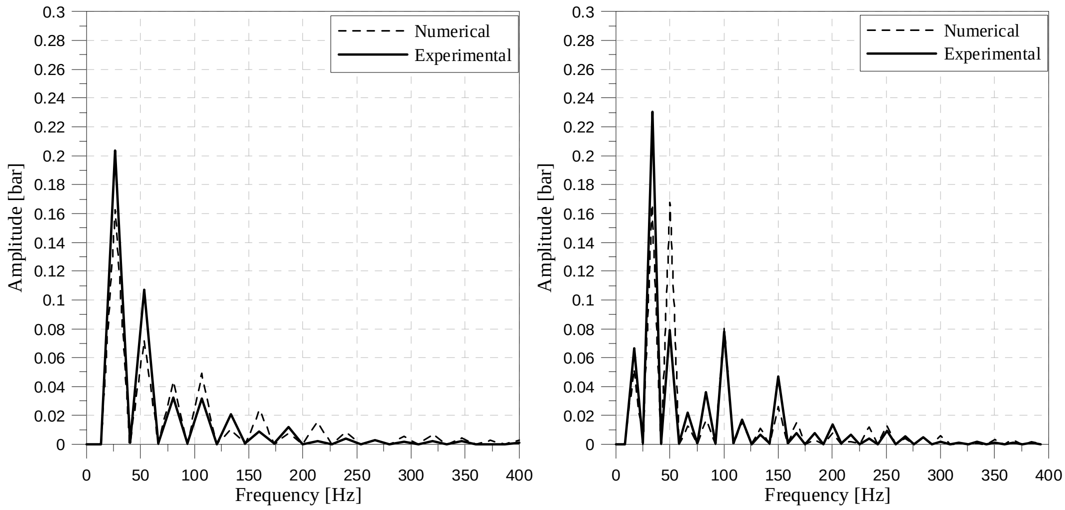


(a) First maximum at 1600 rpm

(b) First minimum at 1000 rpm

Figure 6 –Mass Flow Rate versus Camshaft Revolution for configuration J

A spectral analysis was made for the pressure waves at 900 and 1600 rpm for configuration J. Figure 7 shows the spectrum analysis at the pressure waves at those speeds for configuration J. The geometry of configuration J has the primary length equal to 1639 mm and the secondary length, i.e., the distance between the junction location and the pipe end, 516 mm. The resonance frequency for the primary pipe, considering an open-end aperture, is 69.4 Hz, while the resonance frequency for the secondary pipe is 91.4 Hz. The resonance frequency for the full pipe is 40 Hz. An interesting aspect for the 900 rpm camshaft is that the first harmonic has an amplitude level higher than the fundamental frequency. The reason for this is that the first harmonic gets into resonance with the system frequency of 29.4 Hz (according Eq. 3). The explanation for the high level for the second harmonic is that its frequency is nearest to the resonance frequency for the primary pipe.

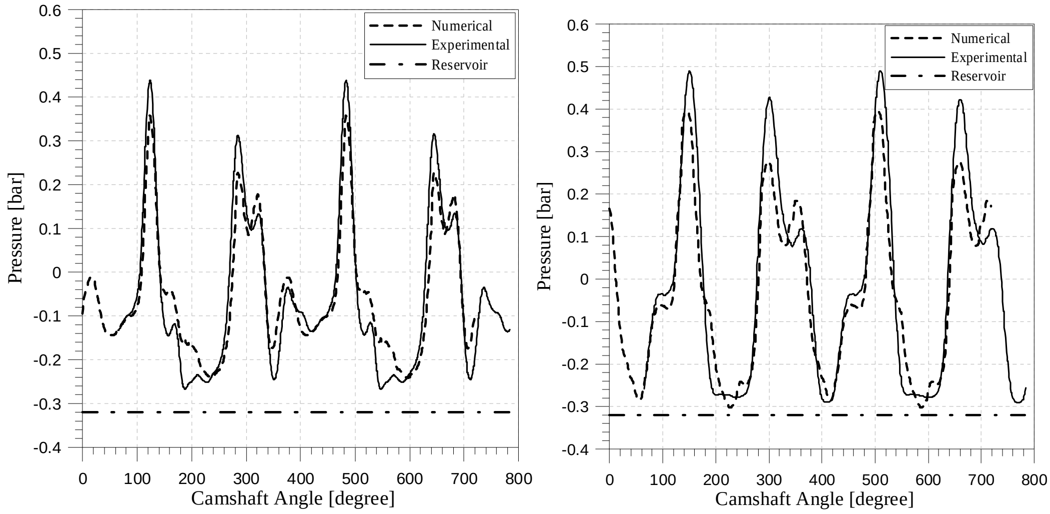


(a) 1600 rpm camshaft

(b) 1000 rpm camshaft

Figure 7 –Amplitude versus frequency for configuration J

Figure 8 presents the experimental and numerical curves of pressure variation along camshaft angle at 1000 and 1600 rpm for configuration A at the valve port. The results exhibit a similar behavior between them and validate the numerical simulation.



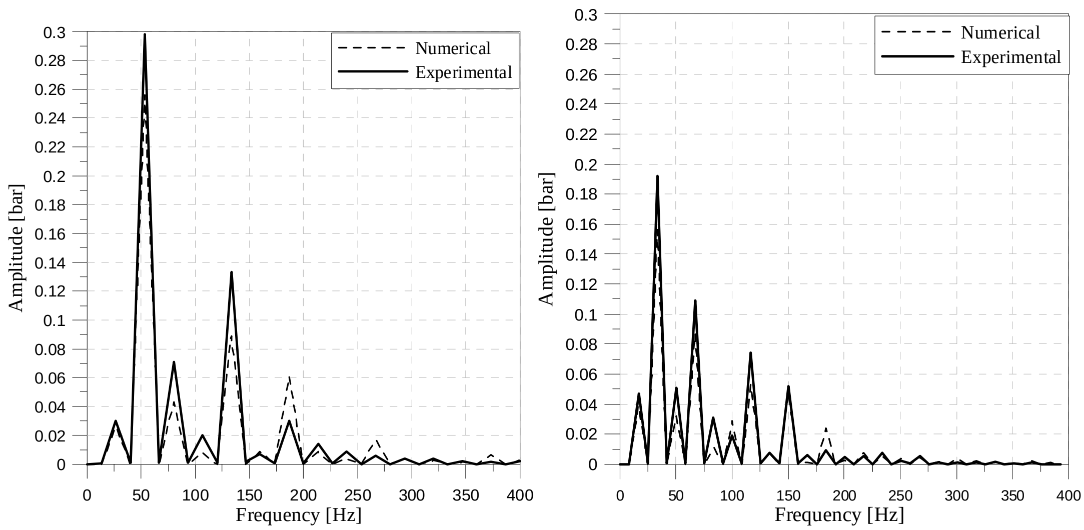
(a) First maximum at 1000 rpm

(b) First minimum at 1600 rpm

Figure 8 –Mass Flow Rate versus Camshaft Revolution for configuration A

Figure 8 shows that the pressure waves are affected by the junction position and its oscillation can increase or not the mass flow rate at the intake system. Others results can be seen in Hanriot (2006).

A spectral analysis was made for the pressure waves at 1000 and 1600 rpm for configuration A. Figure 9 shows the spectrum analysis at the pressure waves at those speeds for configuration A. The geometry of configuration A has the primary length equal to 598 mm and the secondary length, i.e., the distance between the junction location and the pipe end, 516 mm. The resonance frequency for the primary pipe, considering an open-end aperture, is 76 Hz, while the resonance frequency for the secondary pipe is 164.7 Hz. The resonance frequency for the full pipe is 142 Hz. An interesting aspect for the 1000 rpm camshaft is that the first harmonic has an amplitude level higher than the fundamental frequency. The reason for this is that the first harmonic gets into resonance with the system frequency of 59.8 Hz (Eq. 3). The explanation for the high level for the second harmonic is that its frequency is nearest to the resonance frequency for the intake system.



(a) 1600 rpm camshaft

(b) 1000 rpm camshaft

Figure 9 –Amplitude versus frequency for configuration A

7. CONCLUSIONS

The phenomena of pressure pulse propagation and reflection at the presence of junction in the intake manifold engines have been presented and discussed. Experimental and numerical results were obtained for different junction locations. It was shown that the location where the junction is connected at the intake system affects the mass flow rate along the rotational engine speed. There can be an optimum inlet pipe length at a given engine speed. The junction produces a damping effect on the pressure wave and affects the inlet air mass.

8. ACKNOWLEDGEMENTS

The authors wish to thank FAPEMIG, CAPES and FIAT-GM Powertrain Ltda., for the financial support to this project.

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