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HELMHOLTZ RESONATOR EFFECTS ON ENGINE INTAKE AIR MASS FLOW RATE

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Abstract. *The present work investigates an engine intake manifold pulsating flow characteristics and the effects of a Helmholtz resonator on the air mass flow rate into the cylinder. Experiments were carried out in a straight intake pipe connected to a production engine cylinder head mounted in a flow test bench, specially designed for fluid dynamic studies of engine intake systems with unsteady flow conditions. A Helmholtz resonator of varying frequency was installed on different locations in the pipe. The results showed that resonator size and location affect the intake air mass flow rate. The highest intake air mass flow rate was obtained with the longer neck length resonator located closer to the intake valve.*

Keywords: Internal Combustion Engine, Helmholtz Resonator, Intake System, Pulsating Flow, Gas Dynamics

1. INTRODUCTION

An adequate intake manifold design should provide satisfactory engine intake air charge while keeping pressure losses along the intake system to a minimum. The intake air mass flow must be equally distributed through the engine cylinders, which requires optimized fluid dynamics and system layout. Due to piston and intake valve movement, the gas flow in the inlet manifold oscillates. The pulsating gas motion can be used to improve cylinder charge, and the optimum design is affected by the natural frequencies of the intake system (Chalet et al, 2011; Morse, 1938; Winterbone & Pearson, 1999). However, taking the systems to optimum operation is often a difficult task due to practical difficulties associated to measurement of intake pipe unsteady flow field. The application of geometric modifications in the intake manifold to improve engine breathing has found positive results from different strategies, from variable intake plenum length (Ceviz, 2007 ; Ceviz & Akın, 2010) to the use of Helmholtz resonator – HR (Bortoluzzi et al, 1998).

Helmholtz resonator is widely used for noise reduction in vehicle intake and exhaust systems. Intake system noise is generated by pressure pulses that arise from pistons and valves movements, and is generally influenced by the low frequency related to the engine speed. Many authors have discussed the effect of HR location on intake system noise (Kastner, 1947; Dupère & Dowling, 2005).

Helmholtz resonators works on a similar principle to that of vibration absorbers, but in the acoustic domain, it aims at reduced response of acoustic cavities (Brads, 1979; Ih, 2009). Resonators have a defined natural frequency, which depends on parameters such as neck length, cavity volume and neck area (Pogorevc & Kegl, 2006).

Nair et al (2010) studied the use of HR for noise reduction of targeted low frequency acoustic modes of cylindrical cavities. The coupled resonator–cavity system was analyzed using numerical models and validated with experiments. Both numerical modeling and experiments have confirmed that the introduction of a resonator tuned to the cavity mode

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introduces two coupled modes asymmetrically placed about the original cavity mode and the spacing between two coupled modes increases with increasing resonator–cavity volume ratio.

The use of HR tuned with passive control to damp unstable combustion systems was investigated by Zhao and Morgans (2009), with particular emphasis on systems that exhibit multiple unstable modes. In this study the HR was tuned to the measured modal frequencies by varying its neck area. It has been shown that tuned passive control of combustion instabilities can be achieved by tuning HR neck area.

Winterbone and Pearson (1999) described the use of HR to reduce sound pressure levels and mentioned a very interesting discussion about the mechanism of reflection and transmission of sound waves at geometrical discontinuities, like pipe junctions and abrupt area changes, to control the acoustic power generated by the source and transmitted downstream along the pipe-system. A numerical simulation of unsteady flows in internal combustion engine silencers and the prediction of tailpipe noise were presented. The authors showed that modeling silencers were possible by use of (i) the typical set of quasi-steady boundary conditions; (ii) suitable acoustically equivalent duct systems and (iii) end correction for each geometrical discontinuity.

Ballester et al (2008) studied the influence of burner aerodynamics on the characteristics of the flame by means of detailed laboratory measurements in a gas-fired furnace. The authors considered the geometry of the combustion chamber and the exhaust duct like a HR. In this situation, if the combustion chamber and the chimney are assumed to be, respectively, the cavity and duct of a simple HR, the resonant frequency can be calculated by applying the general equation of this acoustic system. Although the geometry of the primary and secondary air ducts is not exactly that of a simple HR, it enabled the estimation of the resonance frequencies associated with both air injections. The resonance frequencies were 185 and 345 Hz for the primary and secondary air injections, respectively, which are relatively close to some of the pressure spectra peaks (220 and 370 Hz).

Selamet et al (2001) investigated the use of HR in the engine induction system. The sound attenuation characteristics of the HR were determined in terms of insertion loss. In this study, it was verified that the HR insertion changes the pressure wave form significantly in the intake pipe at the engine speed with fundamental firing frequency corresponding to the resonance frequency of the resonator.

Stone and Etminan (1992) studied the use of HR to improve engine intake air mass flow rate. The use of resonator volume systems was discussed and the various ways of modeling these systems were compared. The interpretation of the data from the tuned induction systems that incorporate Helmholtz resonators was not so clear-cut. According to the authors, this may in part be due to the complex nature of the flow and pressure pulsations. It was suggested that a Fourier analysis should be undertaken, so that the frequency components present in the flow could be analyzed.

The effect of tunable resonators on the volumetric efficiency of single-cylinder engines was investigated by Bortoluzzi et al (1998). The primary aim was to investigate the possibility of varying the optimum filling frequency by adjusting the volume or the duct of the resonator. The results showed that the intake system equipped with an in-series resonator could be regulated when the duct between the cylinder and the resonator is short; i.e., the Helmholtz frequency of the cylinder and the first duct alone occurs in the region of high frequencies.

The present work investigates experimentally the use of HR to improve engine intake air mass flow rate applying a flow test bench. The main purpose is to produce more insight to engine intake system optimization for increased volumetric efficiency.

2 FUNDAMENTAL CONCEPTS

The dynamic flow process through the intake manifold of an engine can be characterized by the instantaneous evolution of the local pressure downstream the intake valve and close to the intake valve. In those locations, flow pressure is a consequence of both the discharging process and the manifold dynamic response. In an intake system the dynamic interaction between the pressure pulses originated from the intake valves and the reflected pulses from the different boundaries is a complex phenomenon. Benajes et al (1998) showed that the study of a simple pipe open to the atmosphere is relevant for the development of single or multi-cylinder engines with independent intake pipes. That is because the pipes either intake air from the atmosphere or from a very large plenum, which pressure is unaffected by engine operation. In this case, the solution in terms of frequency f of the first harmonic is:

$$f = \frac{\omega}{2\pi} = \frac{c}{4L_p} \tag{1}$$

where ω is the angular frequency (rev/s), Lp is the pipe effective length (m), and c the sound speed (m/s).

The theoretical HR natural frequency is (Benajes et al, 1998):

$$f = \frac{1}{2\pi}\left(\frac{c^2 A}{L_N V}\right)^{1/2} \tag{2}$$

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where A is the resonator neck cross-section area (m^2), V is the resonator volume, L_N is the resonator neck length (m) and c, the sound speed (m/s).

The theoretical frequency can differ from the measured one due to model simplifications. The main simplification is that neck friction effects are not considered in Eq. (2). The frequency can be adjusted introducing an equivalent neck length.

Benajes et al (1998) developed a model based on the wave acoustic theory capable to calculate the natural frequencies (w) of an intake manifold. It can be shown that the location where the wave reflections take place is important to intake system design. Equation (3) shows the system acoustic response in the presence of junctions and intermediate chambers:

$$Z_1 \tan\frac{wL_1}{c} = Z_2 \frac{A_2}{A_1} \cot\frac{wL_2}{c} - \frac{wV}{cA_1} \tag{3}$$

where Z is the number of primary (1) and secondary (2) conduits, w is the intake system natural frequency, L and A are the pipe length and cross-section area in the conduits, respectively, and V is the intermediate chamber volume.

Besides the dynamic response of the whole manifold, pressure reflection takes place at the junction similarly 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 Eq. (3), and the higher natural frequency of the primary pipe, given by Eq. (1).

3 EXPERIMENTAL SECTION

Experiments were conducted in a flow test bench specially designed for fluid dynamic studies of engine intake systems with unsteady flow conditions to compare with the numerical results. The intake system was submitted to a constant pressure difference between the ambient and a 350-liter chamber, used to attenuate pressure pulses from a blower of 0.2 m^3/s capacity. Air flow was controlled through valves locates between the blower and the chamber. The intake valves are driven by an electric motor, which rotational speed is adjusted through a frequency converter.

The flow test bench was fitted with pressure transducers, thermocouple, flow meter and an encoder, being the readings recorded by a computer. The advantage of such test rig is that reliable results are easily obtained in comparison with warm engine tests (Hanriot, 2001).

However, it must be recognized that the tuning effects obtained are not exactly the same as those achieved on operating engines, because there are no moving pistons and no combustion. The major advantage of the flow test bench is that it enables the tuning frequencies of manifold designs, and also it can be used to establish how accurately the flow in the manifold can be simulated.

Initial tests were performed with a PVC-built intake pipe of 1964 mm length and 35 mm diameter, with a natural frequency of 43.7 Hz calculated from Eq. (1). The pipe was connected to one engine cylinder, being the other three cylinders blocked by a metal plate. Camshaft rotational speed was varied from 600 to 2400 rev/min, in steps of 100 rev/min. Nine pressure transducers were installed along the pipe to measure pressure fluctuations. Pressure difference between the chamber and the ambient was kept constant at 230 mmHg (0.306 bar). Table 1 shows the position of the pressure transducers in the intake pipe, with reference to the intake valve.

Further tests were carried out with a HR installed on the intake pipe (Fig. 1).

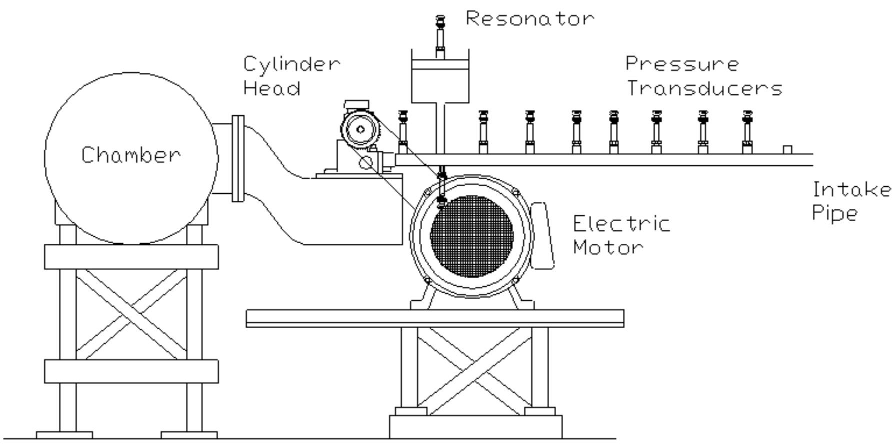

Figure 1 - Schematics of intake pipe with Helmholtz resonator.

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The cylindrical HR was built in acrylic, and its internal volume could be adjusted through a mechanical system (Fig. 2). The resonator chamber internal diameter (DR) was 145 ± 1 mm. The resonator neck length (LN) and diameter (DN) were 251 ± 1 mm and 26 ± 1 mm, respectively. In order to verify the HR response frequency, two resonator internal cavity lengths (LR) were used in the experiments: 30 ± 1 mm (114 Hz resonance frequency, calculated from Eq. (2)) and 246 ± 1 mm (40 Hz resonance frequency, calculated from Eq. (2)).

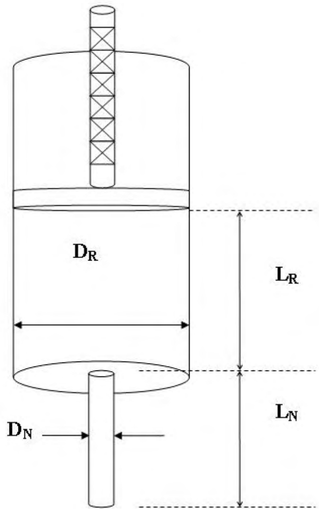

Figure 2 – Helmholtz resonator details.

The lower frequency corresponds to the intake pipe natural frequency, and the higher frequency was chosen considering the mechanical system limit. The resonator was initially installed at position 2 and, then, at positions 6 and 9 (see Table 1). These positions represent the initial, middle and end pipe sections.
Valve flow discharge coefficient was measured with the engine cylinder head connected to the flow bench and varying valve lift. A constant pressure difference was set up between the ambient and the chamber. According to Benson (1982), the mass flow rate through the valve, $\dot{m}$, can be calculated from standard orifice equations for compressible fluid flow:

$$\dot{m}_{real} = C_D \frac{A_{ref}\, p_0}{\sqrt{RT_0}} \left(\frac{p}{p_0}\right)^{\frac{1}{\kappa}} \sqrt{\frac{2\kappa}{\kappa-1}\left[1-\left(\frac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}}\right]} \qquad (4)$$

where C_D is the discharge coefficient, A_{ref} is the reference area, p is the chamber pressure, k is the specific heat ratio and p_0 and T_0 are the atmospheric pressure and temperature, respectively.

Table 1- Pressure transducers and HR position with reference to the intake valve port.

TRANSDUCER/HR POSITION	DISTANCE (mm)
1	153
2	353
3	554
4	754
5	953
6	1154
7	1354
8	1555
9	1755

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The reference area considered in this work was the valve head area. The results obtained for the discharge coefficient calculated by Eq. (4) are shown in Table 2.

Table 2- Measured valve discharge coefficients.

VALVE LIFT (mm)	DISCHARGE COEFFICIENT
0	0
1	0.087
2	0.186
3	0.291
4	0.390
5	0.466
6	0.526
7	0.560
8	0.573

The intake valve was open at 159.5 crank angle degrees before top dead center (°BTDC) and closed at 301.5°BTDC.

4 RESULTS AND DISCUSSION

During the tests the intake air temperature was kept at 20.0 ± 2.5 ^{o}C. The results shown in this section are the average of 40 measurements made at each camshaft rotational speed. Pressure data acquisition rate was 10 kHz. Figure 3 shows the dynamic pressure measured by transducer #1, for 600 rev/min camshaft rotational speed. It can be observed that wave amplitudes are attenuated by reflections inside the pipe and by friction effects. The standard deviations were very small, around 0.01 bar.

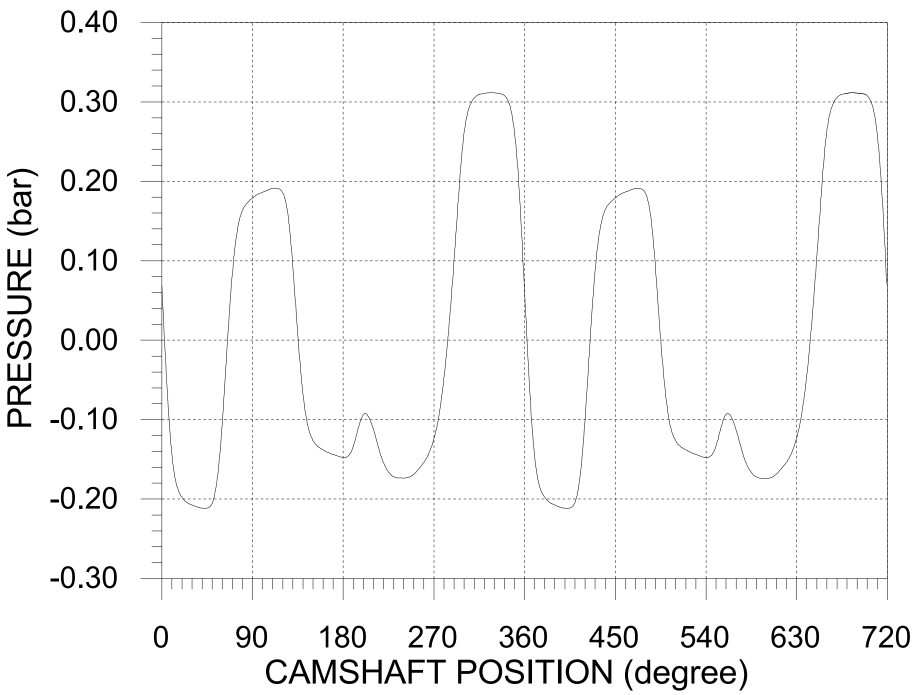

Figure 3 – Pressure fluctuation at 153 mm from the intake valve for 600 rev/min camshaft speed.

Figure 4 shows how the air mass flow rate into the engine cylinder head varies with camshaft rotational speed. In an operating engine the filling process is similar to a volumetric pump, for which the rotational speed is proportional to the intake air mass flow rate. In the flow test bench, with increasing rotational speed the depression inside the cylinder increases and affects system response. The mean mass flow rate measured with pipe without resonator was 13.8 g/s. The maximum and minimum mass flow rates observed are related to the wave phase that arrives at the valve port. Minimum flow rates are recorded at 1200 and 2400 rpm, and maximum flow rates are noticed at 900 and 1600 rpm. The mass flow rate uncertainty for all tests was ± 0.1 g/s.

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As mentioned by other authors [2,3], the intake air mass flow rate is governed by the average pressure difference upstream and downstream of the intake valve, as well as by pressure fluctuations in the intake pipe. Most of the pressure fluctuation (Fig. 3) and mass flow rate (Fig. 4) trends can be explained considering the behavior of a quarter wavelength mode.

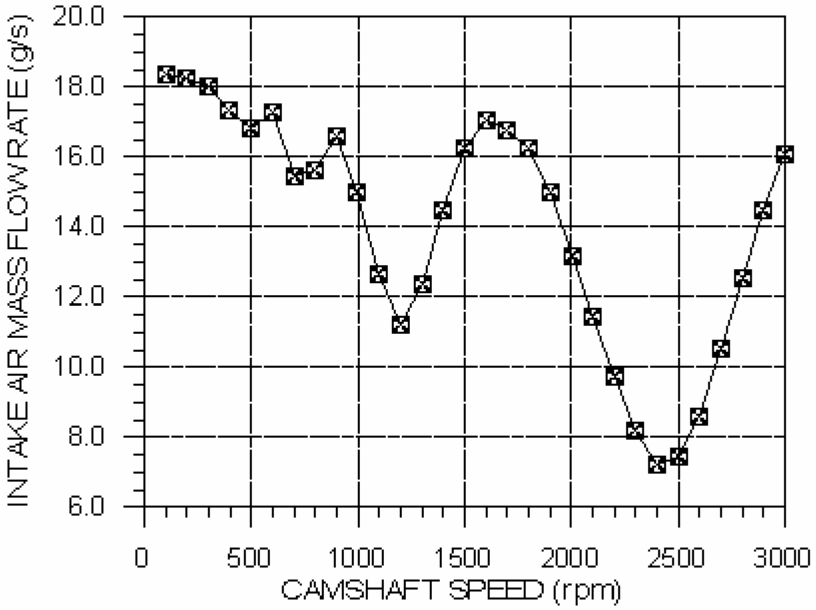

Figure 4 – Intake air mass flow rate for varying camshaft speed.

The fundamental frequency and the harmonics were obtained from spectral decomposition of the pressure fluctuations. As an example, Fig. 5 shows the pressure fluctuation amplitude along the pipe for 2400 rev/min camshaft speed. There is an indication of the existence of a stationary wave with fundamental frequency and high amplitude close to the valve. The wave has a frequency close to the quarter wavelength, since the angular period the valve remains closed (218^{o}) is longer than the period its remains open (142^{o}). It has been observed that the 2nd, 3rd and other harmonics of higher order have low amplitude and do not significantly affect the pulsating phenomena.

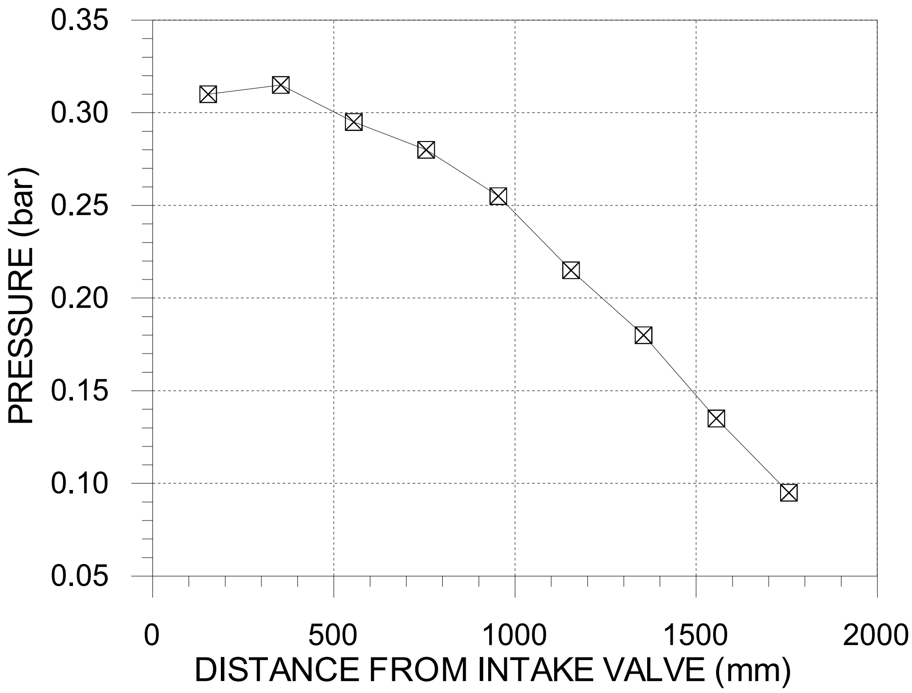

Figure 5 – Pressure fluctuation amplitude along the pipe for 2400 rev/min camshaft speed.

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Figure 6 shows the intake air mass flow rate variation with camshaft rotational speed using the 114 Hz HR at positions 2, 6 and 9. It can be noticed that, for positions 2 and 6, the resonator modifies the pipe mass flow rate behavior. Moreover, the resonator location affects the response of the valve-pipe system. The valve is the excitation source. The results obtained from resonator located at position 9 (close to pipe end) indicate that it does not act at this position. This is probably due to the inability of the pressure waves to interact with the resonator in that location. This means that the HR cannot capture the interaction of the pressure waves generated by the valve, due to the fact that these waves have low amplitude.

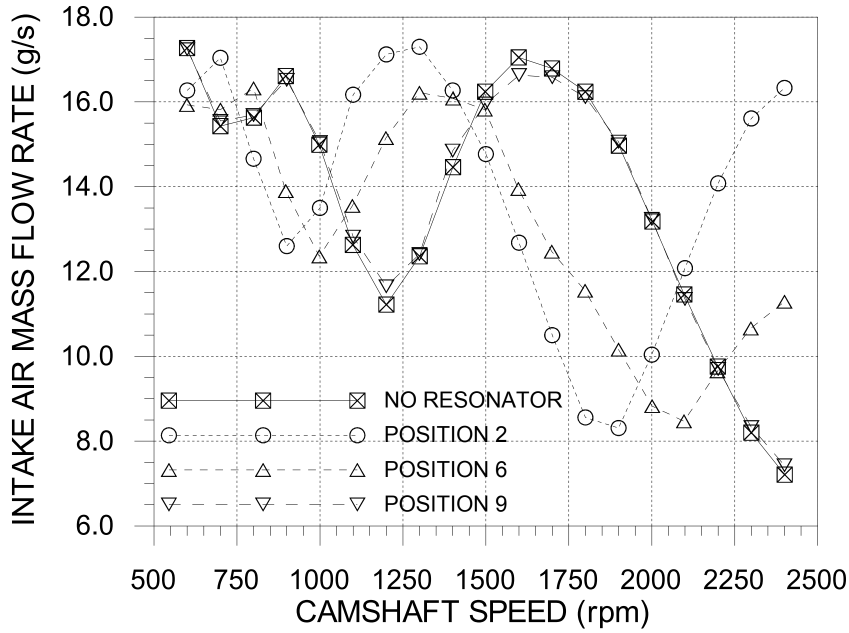

Figure 6 – Variation of intake air mass flow rate with camshaft speed for 114Hz Helmholtz resonator at three different positions.

The mean mass flow rates recorded with HR at positions 2, 6 and 9 were 13.9 g/s, 13.1 g/s and 13.6 g/s, respectively. The results indicate that the resonator closer to the valve port has the higher intake air mass flow rate. The resonator position changes the basic tuning mechanism in the manifold flow dynamics, caused by the unsteady and periodic nature of the intake process. As a consequence the inlet valve pressure is altered, thus changing the mass flow rate.

Figure 7 shows the intake air mass flow rate variation with camshaft rotational speed using the 40 Hz HR at positions 2, 6 and 9. As mentioned before, this is the intake pipe natural frequency. Comparing with Fig. 6, it is observed that the 40 Hz resonator also alters the intake air mass flow rate. It can also be seen that when the resonator is located close to the valve the highest mean mass flow rate is obtained, of 16.6 g/s. When the HR is located at positions 6 and 9 the mean mass flow rates are equal to 14.8 g/s and 14.3 g/s, respectively.

The results indicate that the resonator position and its natural frequency are important parameters that are related with the quarter wavelength mode of the stationary wave. From these results, aiming at increased engine intake air mass flow rate, the Helmholtz resonator should be positioned at the locations of high standing wave amplitude.

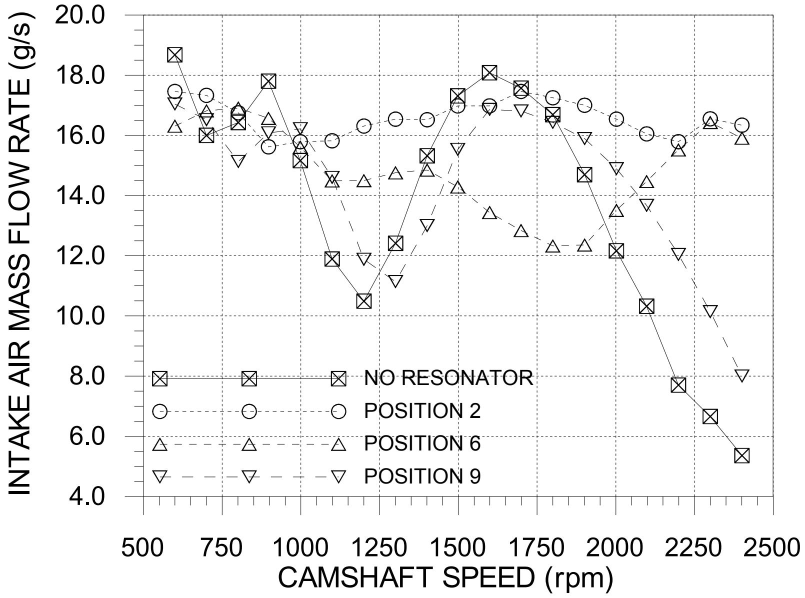

Figure 7 – Variation of intake air mass flow rate with camshaft speed for 40Hz Helmholtz resonator at three different positions.

5 CONCLUSIONS

From the results obtained, the following conclusions can be drawn.

The pulsating flow that occurs in the engine intake manifold is originated from the periodic movement of the intake valves and from the pressure difference between the atmosphere and the cylinders. The pulsating flow can either increase or decrease the air mass flow rate into the cylinder. The wave amplitude is attenuated by reflections inside the pipe and by friction effects.

The maximum and minimum mass flow rates are related with the wave phase that arrives at the valve port. The wave has a frequency close to the quarter wavelength, since the angular period the valve remains closed is longer than the period its remains open. The quarter wavelength is the dominant mode in the intake system.

Both the Helmholtz resonator natural frequency and its pipe location affect the engine intake air mass flow rate. The resonator changes the system response, creating an effective length different to the actual pipe length. The resonator with 40 Hz resonance frequency produced an average intake air mass flow rate higher than that produce by 114 Hz resonance frequency.

The resonator placed near the valve can increase the mass flow at its resonance frequency. The closer the Helmholtz resonator is located to the valve port, the higher the engine intake air mass flow rate. The resonator position changes the basic tuning mechanism of the manifold flow dynamics resulted from the unsteady and periodic nature of the intake process.

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