Effects of Insertion of Resonating Chambers in the Intake System of Internal Combustion Engines on the Air Charge

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Abstract: The present work describes an experimental and numerical analysis of the pressure oscillations caused by the valve movements in the presence of resonating chambers in the intake system, with the objective to determine the conditions at which the intake air charge can be increased. The mass flow rate and pressure in the intake system were evaluated by a computational program that simulates the pressure wave action in the intake pipe using the method of characteristics. The experimental investigation was carried out in an engine intake system with a straight intake pipe where the resonators were located in several different positions. The intake system was connected to an air supply system consisted by a blower, a flow measurement device and a reservoir chamber. The camshaft was rotated by an electrical motor, allowing for investigation of a large speed range. The results showed that the resonating chamber locating at the intake system affects the mass flow through the intake valve.

1. Introduction

The present work has as a main objective a study on the influence on the pressure waves due to the introduction of a resonating chamber in the intake system of an internal combustion engine, with the aim to increase the engine volumetric efficiency through an increase of the intake air mass charge. Although resonating chambers are vastly employed to reduce sound pressure levels, their usage to improve the intake air mass charge to the engine still deserves further investigation. The pressure oscillations can also be awkward to the volumetric efficiency if the interaction between the resonating chamber and the intake pipe is not conveniently dimensioned. The project of intake conduits also involves parameters such as the pipe length and diameter, accessories and the opening and closing periods of the intake and exhaust valves.

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. Most of the automobile engines are naturally aspirated and operate on the four-stroke cycle, in which distinct piston movements are used to induce air and exhaust burned gas. These intake and exhaust strokes enable the engine to pump gas through itself, and this process can significantly be affected by the design of the intake and exhaust systems. The requirement for lower noise and pollutant emissions levels has further increased the importance of the design of the intake and exhaust systems. 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 manifold has an effect on the frequency and amplitude of the pressure waves, which originate noise. The intake system components are often engineered to attenuate the amplitude of the pressure waves and can be designed to act 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.

2. Literature Review

Morse et al. (1938) were among the first to study pressure pulses in the intake system originated from the reciprocating valve movements. The authors showed that pressure fluctuations can improve or reduce the engine volumetric efficiency. In the intake system, the rarefied pulse originated from the valve port is pushed to the pipe inlet and is reflected back to the cylinder from some point in the intake pipe. The exact location of the point from where the pressure pulse is reflected is of particular interest to the project of the conduit geometry. The pipe ends and junctions are typical examples of locations where the pressure wave is reflected. When that happens, the pressure pulse shape and development along the intake conduit are of some complexity.

In a single cylinder configuration, the pressure signal next to the intake valve is rapidly increased during the valve closing. From the pressure increase to the valve closing, typical pressure waves from the wave quarter resonance are installed in the conduit (Hanriot et al., 1999, 2000). The wave pressure is the main physical phenomenon of practical interest in the transient flow regime in the intake pipe. The average mass flow rate behavior is related to the excitation frequency, which is, in this case, the intake valve frequency, and to the pipe natural frequency.

Usage of resonating chambers to improve the intake air mass charge to the cylinder is not vastly mentioned in the literature. Benson (1971, 1982, 1986) was one of the precursors in the application of a similar device for this purpose. Nishio et al (1991) presented experimental results obtained from a so-called pulse simulator, which is very similar to the bench flow test apparatus used in this work. Other literature sources in the subject are Sassi (1996), Puglisi (1996) and Hanriot (1999, 2000).

3. Experimental Set-Up and Procedure

3.1. Flow Bench

The flow bench is a installation dedicated to the experimental analysis of steady state and transient flow in the intake and exhaust systems of internal combustion engines. The main component of the flow bench is a blower, which maximum volumetric flow rate is 600m^3/h and maximum pressure is 0.7bar above the ambient pressure. The blower is connected to the test apparatus through a set of pipes including mechanical and electrical valves and two plenum chambers to dump pressure oscillations (Fig. 1). The valve set allows for regulation of the flow rate, and to establish the direction of the air flow. Two computers connected to the flow bench; the first one is used to obtain the volumetric and mass flow rates, and the second to acquire and convert data from the several measuring sensors of the flow bench.

(a) (b)

Figure 1 – Front view (a) and side view (b) from the flow bench and test apparatus.

3.2. Resonating Chambers

The resonating chambers are used mainly to noise control, and their length is adjusted to reflect the wave and attenuate the sound propagation. The sound loss, in this case, is basically a function of two parameters: the expansion rate, m, defined as the relation between the pipe and resonation chamber areas ($m = A_1/A_2$); and the ratio of the chamber length, L2, and the wave length λ, which, in its turn, is defined as the ratio between the sound speed and the sound wave frequency. The drop in the sound transmission is given by equation (1) :

$$T_r = 10 \log\left[1 + \frac{1}{4}\left(m - 1/m\right)^2 \text{sen}^2(2\pi L/\lambda)\right] \tag{1}$$

As the incoming sound wave frequency is varied, the chamber resonates as a partially closed pipe, according to λ = 2L, L, 2L/3, L/2, and successively wave lengths. Figure 2 shows a schematic drawing of the intake pipe and resonating chamber used in this work.

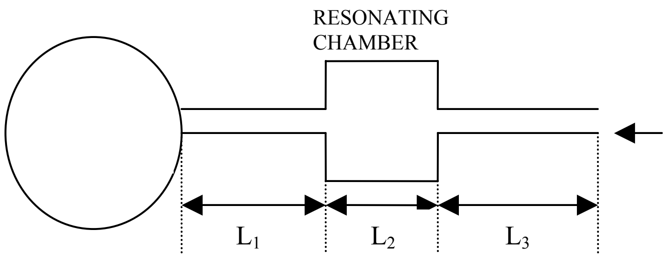

Figure 2 – Schematic drawing of the resonating chamber in the intake pipe.

3.3. Experimental Apparatus

The experimental apparatus allows for application of different pressures in the plenum chamber, by connecting the intake pipe to the blower of the flow bench (Fig. 3). The engine cylinder head was coupled to the equalizing tank, and the electric motor was connected to the pulley of the cylinder head (Fig. 4).

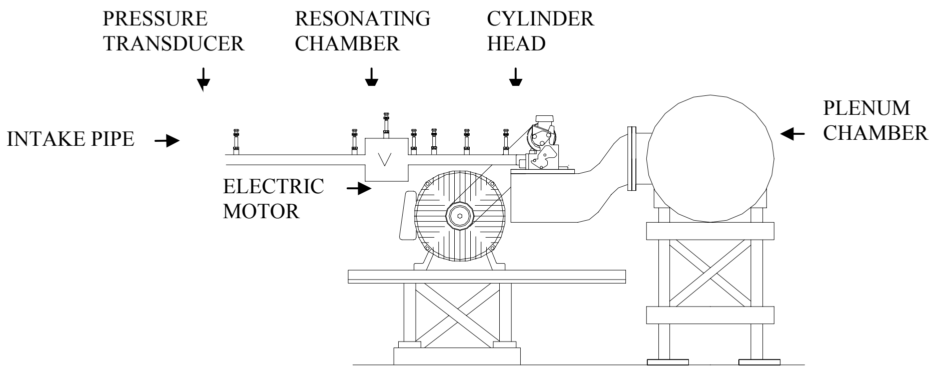

Figure 3 – Side view of the experimental apparatus.

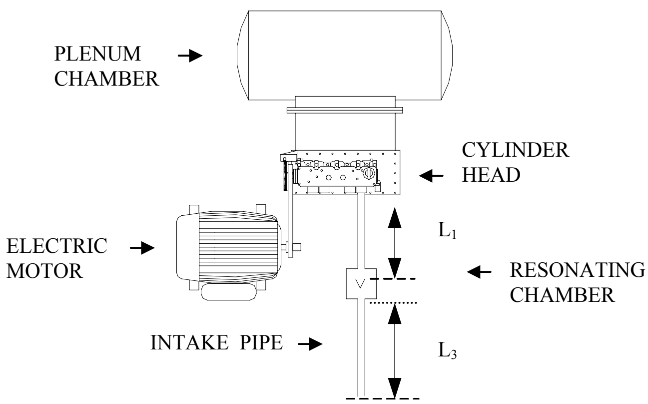

Figure 4 – Top view of the experimental apparatus.

From the tests carried out in the experimental set up, the following information were obtained:

- wave propagation in the intake system in the presence of resonating chambers, and with valve movement for a range of rotational speed;
- pressure wave propagation in pipes of different configurations; and
- temporal variation of pressure and valve position.

3.4. Intake Pipe Length

The intake conduits of internal combustion engines are, in general, of complex geometry, due to the limited space available for the whole powertrain system. This limitation is added to the pressure wave interference coming from the pistons that are not in the intake process. To ease interpretation of the results, the intake conduit used in this work is straight, of 34.0 mm internal diameter, with only one resonation chamber installed. Just one cylinder was considered, the other three being blocked, and the pressure difference between the atmosphere and the plenum chamber to which the cylinder head was connected was 0.306 bar. Three configurations were modeled, with resonating chambers of 1, 2, and 4 dm^3 of volume:

- configuration 1 – resonating chamber next to the intake valve, L_1=220,0 mm; L_2=158,7 mm; L_3=1669,3 mm (4 dm^3); resonating chamber next to the intake valve, L_1=220 mm; L_2=126.0 mm; L_3=1842.0 mm (2 dm^3); resonating chamber next to the intake valve, L_1=220.0 mm; L_2=100.0 mm; L_3=1880.0 mm (1 dm^3);
- configuration 2 – resonating chamber at an intermediate position, L_1=1223 mm; L_2=158.7 mm; L_3=663.3 mm (4 dm^3); resonating chamber at an intermediate position, L_1=1222.0 mm; L_2=126.0 mm; L_3=1669.3 mm (2 dm^3) ; resonating chamber at an intermediate position, L_1=1222.0 mm; L_2=100.0 mm; L_3=879.0 mm (1 dm^3) ;
- configuration 3 – resonating chamber far from the intake valve, L_1=1819.3 mm; L_2=158.7 mm; L_3=70 mm (4 dm^3); resonating chamber far from the intake valve , L_1=1819.3 mm; L_2=126.0 mm; L_3=242.0 mm (2 dm^3), resonating chamber far from the intake valve, L_1=1816.0 mm; L_2=100.0 mm; L_3=282.0 mm (1 dm^3).

A 2m long intake pipe free from resonating chambers was also tested as a reference. Pressure waves and air mass flow rates in the intake system with resonating chambers were compared with those in the intake pipe alone through a range of camshaft rotational speed.

For a straight pipe with a vibrating piston in one end and an opposite open end, the resonance frequency is given by (Kinsler, 1980; Hall, 1987):

$$f_n = \frac{n}{2}\frac{c}{L + \frac{8}{3\pi}a} \tag{2}$$

where L is the pipe length, c is the sound speed, n is harmonic order, and a is the pipe radius. As a is small, the entrance effects can be neglected, and Eq. (2) becomes:

$$f_n = \frac{n}{2}\frac{c}{L} \tag{3}$$

For a straight pipe with a vibrating piston in one end and an opposite closed end, the resonance frequency is given by (Kinsler, 1980; Hall, 1987):

$$f_n = \frac{2n-1}{4}\frac{c}{L} \tag{4}$$

The tuned frequency of the whole manifold is defined by equation (5). This construction effectively consists of one primary pipe connected to the engine cylinder, a plenum chamber, and a further secondary pipe breathing from the atmosphere (Winterbone, 2000):

$$\frac{F_s}{F_p}\cot\frac{wl_s}{a} = \frac{wV}{F_p a} + \tan\frac{wl_p}{a} \tag{5}$$

where F_s is the cross-sectional area of the secondary pipe, l_s is the length of the secondary pipe, F_p is the cross-sectional area of the primary pipe, l_p is the length of the primary pipe, a is the sound speed, V is the volume of the plenum chamber, and w is the angular frequency.

3.5. Experimental Procedure

The intake pipe and each of the four resonating chambers tested, one at a turn, were connected to the flow passage to only one intake port of the 8-valve cylinder head, the other 7 valves being removed and the ports blocked. Thus, the pressure wave propagation to only one engine cylinder was analyzed. The camshaft rotational speed range tested was from 400 to 2600 rev/min, and the valve opening frequency was from 6.67 to 40.0 Hz. The speed was varied in intervals of 200 rev/min through the electronic control board of the flow bench. Electronic data was acquired from piezoresistive pressure transducers located close to the valve port, along the intake pipe, and in the resonating chamber. The mass flow rate was determined from the pressure difference across a viscous flow meter, which were read from two liquid-in-column manometers. The ambient pressure, temperature and humidity were also monitored through a barometer, and a thermo-hygrometer. The flow control valves of the flow bench were positioned at each camshaft rotational speed to maintain a pressure difference of 0.03 bar between the atmosphere and the plenum chamber. The results presented are the average of three tests carried out at the same conditions.

4. Results e Discussions

Figures 5 to 7 show the air mass flow rate variation with camshaft rotational speed for the 2 m long straight pipe and for the intake pipe with 1-liter, 2-liters and 3-lkiters resonating chambers, one at a turn, in several positions, as configurations 1, 2 and 3 described in Section 3.4. One can notice that configuration 1 (Fig. 5), which features the resonating chamber closer to the intake valve, presents the highest mass flow rates for high engine speeds in comparison to configurations 2 and 3 (Figs. 6 and 7). Compared to the straight pipe, it can be observed that utilization of a resonating chamber increased the air mass flow rate for configuration 1 at rotational speeds above 2000 rev/min (Fig. 5). The rotational speed of 2400 rev/min corresponds to the resonance frequency of the 2m long straight pipe, causing a pressure wave opposite to the flow direction at the valve port and, thus, contributing to reduce the mass flow rate. Utilization of a resonating chamber also increased the mass flow rate in configurations 2 and 3 at this speed range, but to a less extent (Figs. 6 and 7).

With respect to the size of the resonating chamber, the biggest chamber, of 4-liter volume, produced the best overall results with configuration 1, where the chamber was located close to the intake valve (Fig. 5). The smallest chamber, of 1-liter volume, produced the worst results at camshaft speeds above 1500 rev/min. For configurations 2 and 3, with the chambers distant from the intake valve, the size of the chamber little interfere in the results (Figs. 6 and 7).

From these results, one could jump to the conclusion that the higher the volume of the resonating chamber, the higher the air mass flow rate to the cylinder, provided that the chamber is located in the closest position to the intake valve. Further investigation on the subject is necessary, as there may be a chamber volume limit above which no substantial increase on the mass flow rate would be observed. For practical purposes, the space available for installation of the intake system in the vehicle will limit the size and location of the resonating chamber.

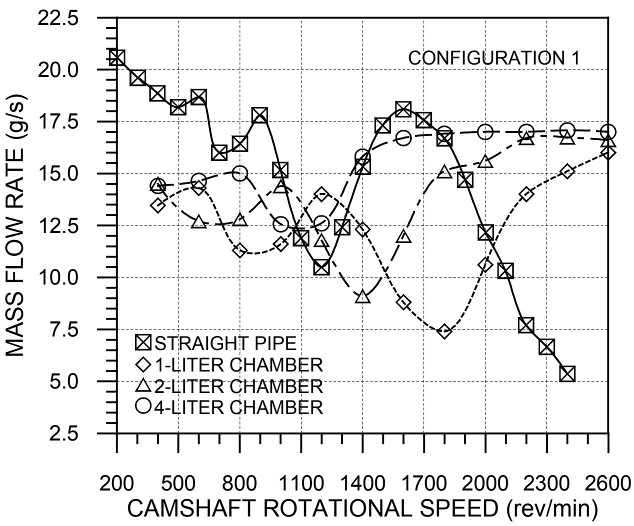

Figure 5 – Intake air mass flow rate variation with camshaft rotational speed for configuration 1.

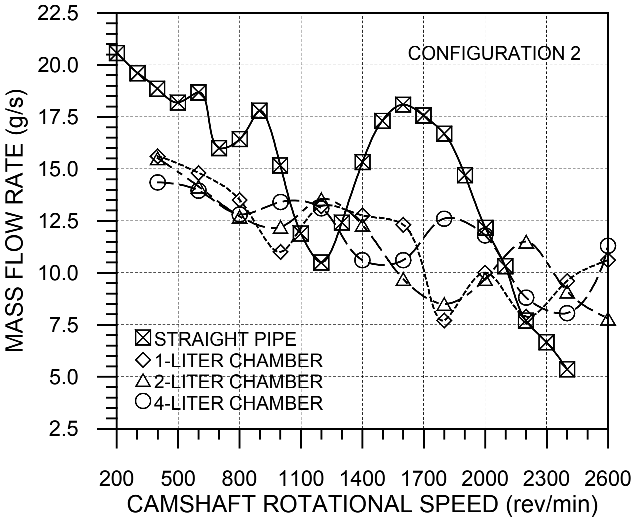

Figure 6 – Intake air mass flow rate variation with camshaft rotational speed for configuration 2.

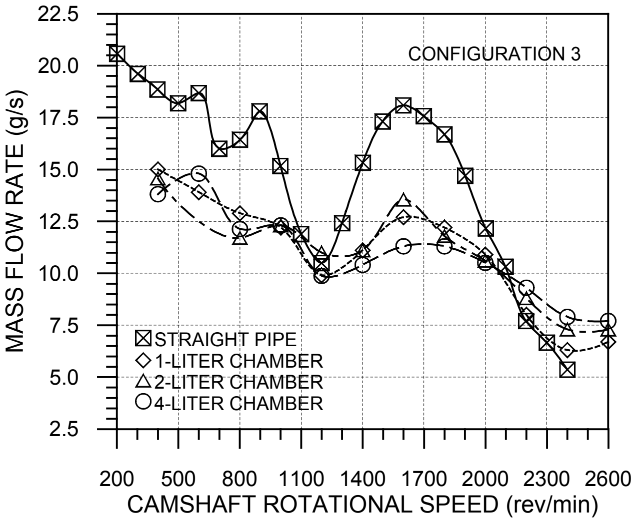

Figure 7 – Intake air mass flow rate variation with camshaft rotational speed for configuration 3.

5. Conclusions

- Utilization of resonating chambers in the intake system of internal combustion engines with the objective to increase the intake air mass charge was investigated.
- Utilization of resonating chambers in the intake system did not provide higher mass flow rate than a simple straight pipe for the whole camshaft engine speed range investigated, but only for engine speed above around 1800 rev/min.
- The results indicate that the closest the location of the resonating chamber to the intake valve, the higher the intake air mass flow rate at a high engine speed.
- From the three resonating chamber sizes evaluated, the biggest chamber produced the highest air mass flow rate, and the smallest chamber produced lowest mass flow rate.
- There may have a resonating chamber size limit above which the air mass flow rate will no longer increase.
- The space available for the intake system in the vehicle will limit the size and location of the resonating chamber.

6. Acknowledgements

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

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