

Analysis of the Fluid Flow in Two Intake Pipes with a Junction

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Abstract. The design of engine intake systems involves optimization of parameters such as the pipe length and diameter, junctions, accessories and the intake and exhaust valve opening and closing timings. The correct dimensioning 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 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 in several locations of the intake system were measured. The experimental data was obtained from the intake system connected to a cylinder head, where the two intake valves are phase shifted by 180°. The cylinder head was installed in an air supply system consisted by a blower, a flow measurement device, and a reservoir chamber. The valves were driven by an electric motor with controlled rotational speed. The results showed 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 primary 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, in order to impose 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 inhale air and exhaust burned gas. These intake and exhaust strokes enable the engine to pump gas through itself, and this process can be significantly 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 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 adverse 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 (Kong and Woods, 1992; Sung and Song, 1996).

2. Literature Survey

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; Winterbone and Pearson, 2000). It is therefore essential that techniques that capture unsteady flow effects be employed in the design of tuned intake systems.

Winterbone and Yoshitomi (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 speeds. 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 et al. (1997) presented a pre-design model for intake manifolds in internal combustion engines. The model was based on the acoustic-wave theory, and made it possible to calculate 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 locations 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 affects the mass flow through the cylinder are 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.

The presence of a junction implies in an additional flow 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, but, as a guideline, pressure losses should be included in high-speed engines where the gas velocities are high. The designer is 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 separated. However, 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:

- Execution of tests to analyze the flow in 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 and Pearson, 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 and Pearson (2000) describe 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 check 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. Experimental Set-Up and Procedures

The flow rig (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).

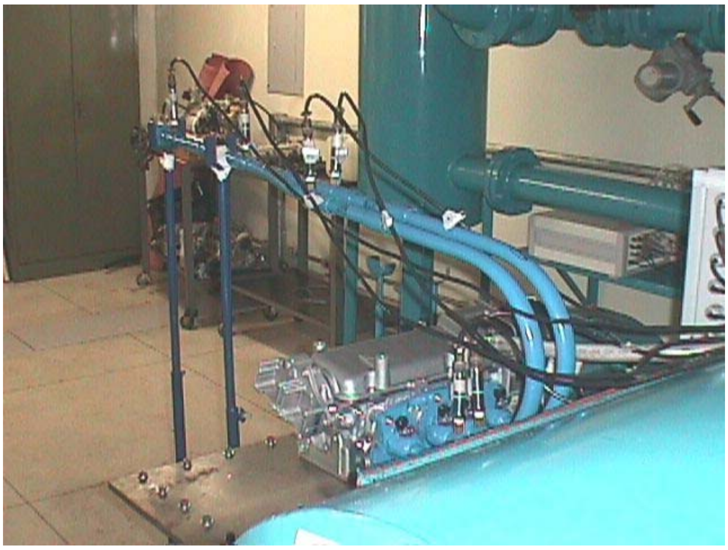


Figure 1 - View of the experimental rig.

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 basic difference from these meters to the orifice plate is that the fluid is forced through small passages that make the flow laminar. Thus, the mass flow rate is directly proportional to the pressure difference through the meter. The instantaneous pressure is measured by piezoresistive pressure transducers, with a working range of ± 2 bar. The temperature sensors are of the platinum resistance type, for use between 0 and 60 ^oC. A four-cylinder, 1.0 liter engine cylinder head was fixed to the dumping tank. Only the intake valve of the second and third cylinders were operating, while the other valves remained closed. A 2.155 m long, 22.30 mm internal diameter straight tube containing a junction was connected to the valve ports. As the pressure waves are one-dimensional and are not affected by the presence of curves in the flow path, the steel tube satisfactorily plays the role of the intake system.

Six pressure transducers were 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. Two configurations were tested, named configuration 'H' and configuration 'J'. The configurations were divided into L1, L2 and L3 lengths.

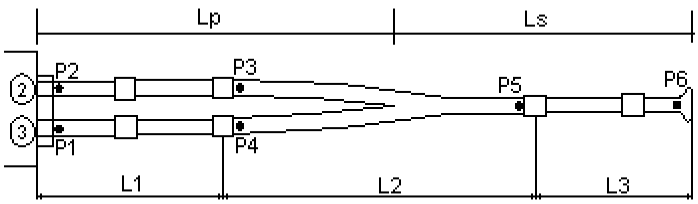


Figure 2 - Schematic drawing of the transducers and the intake system.

Configuration H corresponded to L1 = 1010.0 mm, L2 = 315.0 mm, and L3 = 830.0 mm. The configuration 'J' corresponded to L1 = 1509.0 mm, L2 = 315.0 mm, and L3 = 331.0 mm. 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. A detail of the configuration tested is shown in Fig. 3.

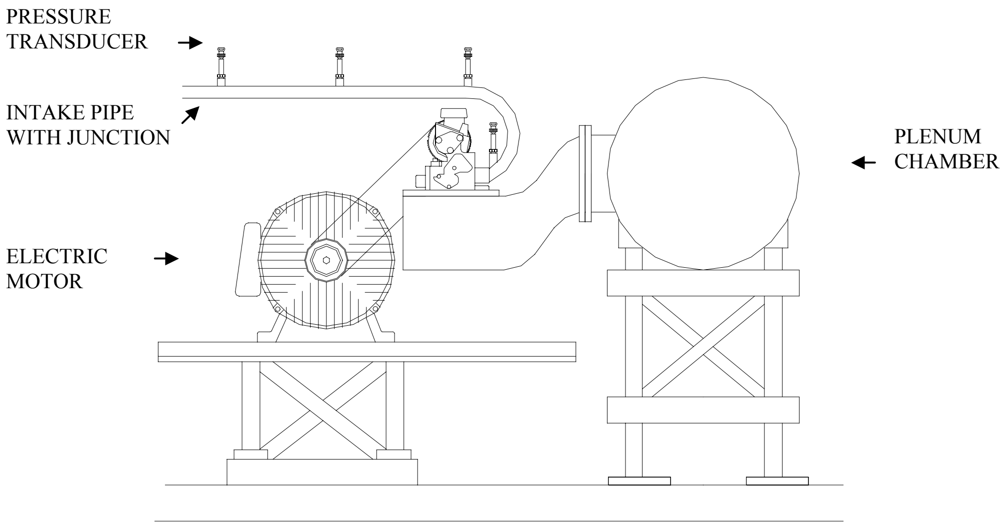


Figure 3 - Schematic drawing of the experimental rig.

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.

4. Results and Discussion

Figure 4 shows the mass flow rate through the intake system versus camshaft revolution for the two tested configurations. It can be seen that the behavior of the two configurations is very similar until around 1600 rpm. After this speed, the configuration H presents a higher level of mass flow rate when compared to configuration J. At 1600 rpm, configuration J presents a maximum mass flow rate and from then on it presents a negative slope. On the contrary, configuration H continues to show increasing mass flow rate and reaches a maximum around 2000 rpm. However, after 2000 rpm both curves present a negative slope, with configuration H presenting a higher mass flow rate in comparison to configuration J. As a possible explanation for this fact, it should be noticed that configuration H is nearer to the valve port than configuration J and, for this reason, presents a higher resonance frequency than configuration J for the primitive pipe. The higher resonance frequency of configuration H makes the maximum flow rate being reached at a higher engine rotational speed.

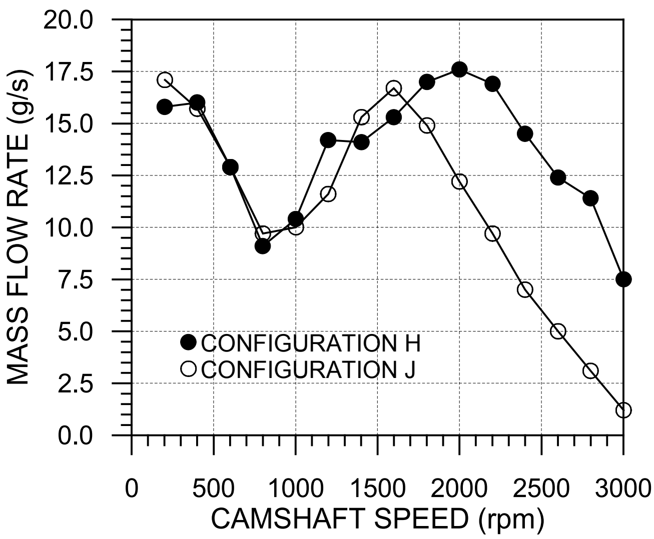


Figure 4. Mass flow rate for different junction positions in the intake manifold as a function of camshaft rotational speed.

The rotational speeds of 1200 and 2000 rpm were chosen to analyze the system behavior. These speeds were chosen because, for configuration H, an inflection point for the mass flow rate is observed around 1200 rpm, and a maximum mass flow rate is observed around 2000 rpm (Fig. 4).

Figure 5 presents the curves of pressure variation along camshaft angle at 1200 rpm for configurations H and J. The results exhibit a very similar behavior between the two configurations for the period when the intake valve is open, i.e., between IVO and IVC, particularly for the transducers P1 and P2, which are located close to the intake valve port. Thus, the mass flow rate should be approximately equal for these configurations at 1200 rpm, which, in fact, is shown by Fig. 4. Besides, from the results shown by Fig. 5 (a) and (b), it can be noticed that transducers P1 and P2 acquired the same pressure waves, shifted by an 180 degrees. This result was expected, once the valves from cylinders 2 and 3 are phase shifted by 180 degrees. A similar behavior happens for the pressure waves for transducers P3 and P4 (Fig. 5 (c) and (d)). Another observation is that the amplitude of the pressure waves from the intake valve port until the entrance of the pipe are attenuated. Viscous effects of pipe and junction cause this attenuation and, as expected, the lowest pressure amplitude is close to the entrance of the pipe, as shown by the pressure waves for transducer P6 (Fig. 5 (f)).

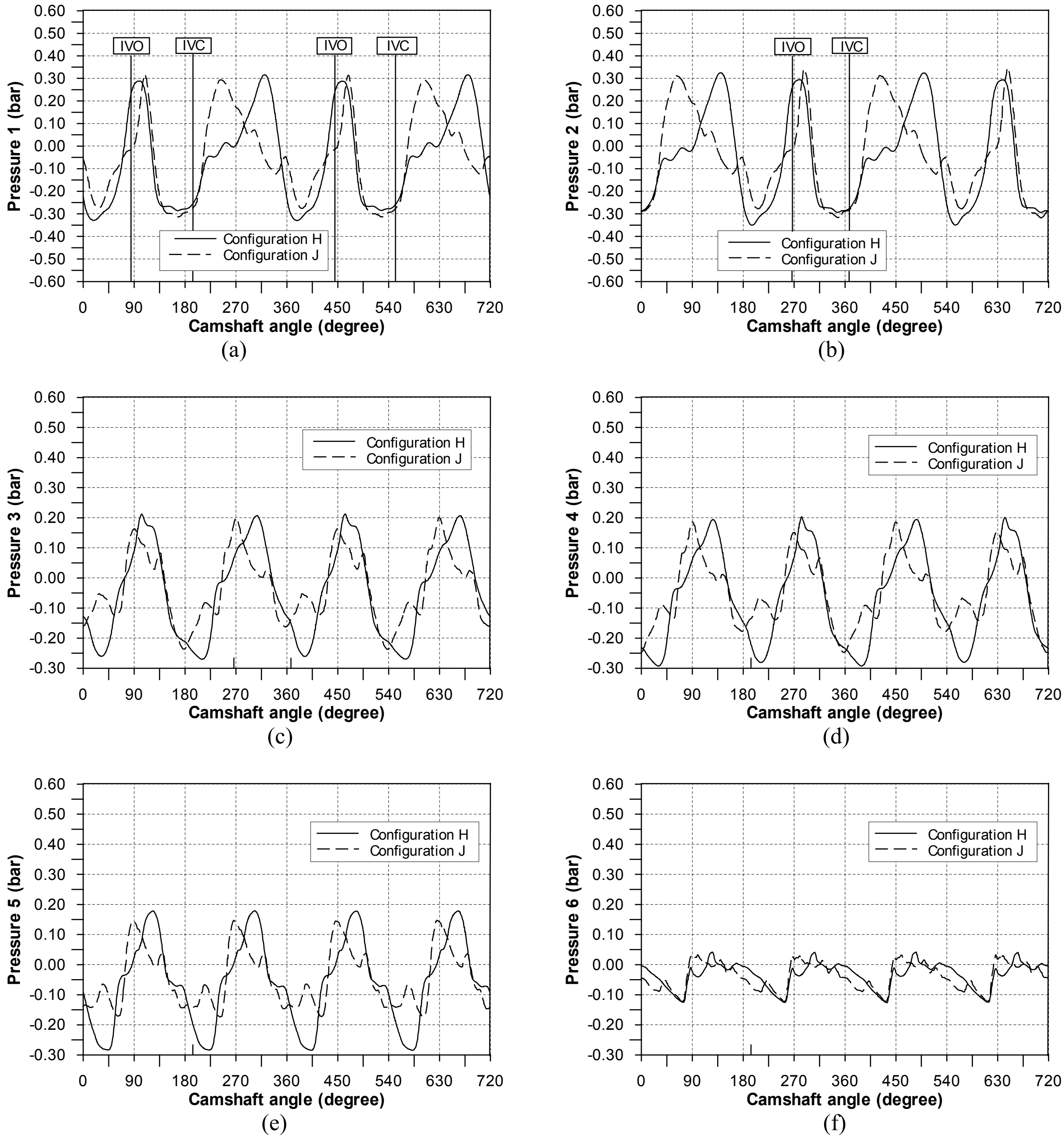


Figure 5- Pressure signals from the pressure transducers P1 (a), P2 (b), P3 (c), P4 (d), P5 (e) and P6 (f) at 1200 rpm camshaft speed.

Figure 6 presents the curves of pressure variation with camshaft angle at 2000 rpm for configurations H and J. The results exhibit a different behavior for configurations J and H, and are distinct from the results at 1200 rpm (Fig. 5), particularly for transducers P1 and P2. As configuration H shows a peak during the valve opening period, its mass flow rate is higher than that of configuration J. The mass flow rate is related to the difference between the pressure wave and the constant pressure at the reservoir. Another observation is that the pressure waves for transducers P1 and P2 exhibit a very similar behavior between the two configurations during the period when the intake valve is open, i.e., between IVO and IVC. Just like the results from Fig. 5, P1 and P2 present similar pressure waves, but shifted by 180 degrees.

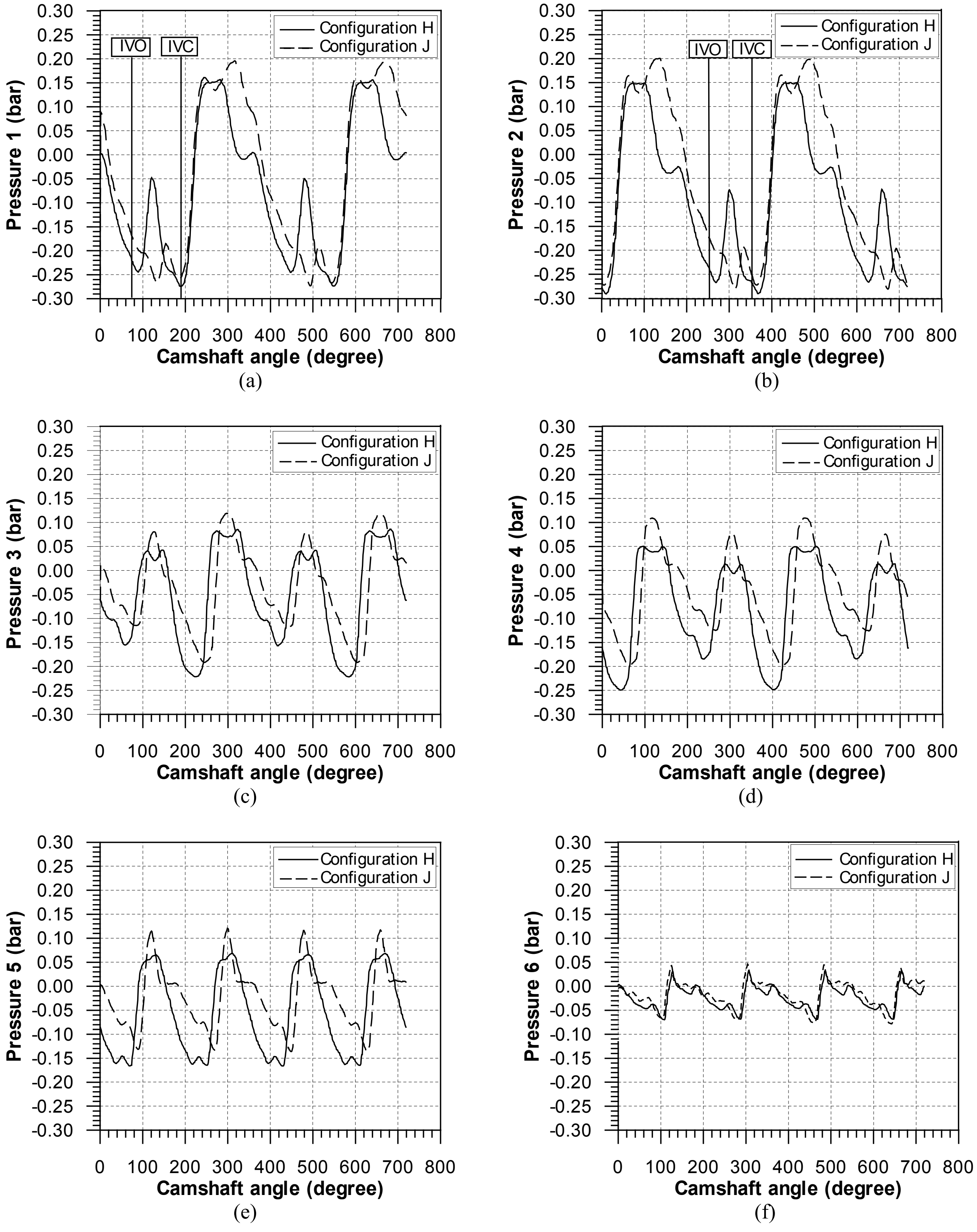


Figure 6- Pressure signals from the pressure transducers P1 (a), P2 (b), P3 (c), P4 (d), P5 (e) and P6 (f) at 2000 rpm camshaft speed.

A spectral analysis was made for the pressure waves at 1200 and 2000 rpm. Figure 7 shows the spectrum analysis at the pressure waves for 1200 rpm for configuration H. The geometry of configuration H has the primary length, the distance from the valve port and the junction (see Fig. 2), Lp = 1225 mm, and the secondary length, the distance between the junction location and the pipe end (see Fig. 2), Ls = 930 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 39.4 Hz. The spectral analysis for P1 (Fig. 7 (a)) shows that the first three peaks are higher than the other harmonics. An interesting aspect 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 full pipe frequency of 39.4 Hz. The explanation for the high level for the second harmonic is that its frequency is nearest to the resonance frequency for the primary pipe.

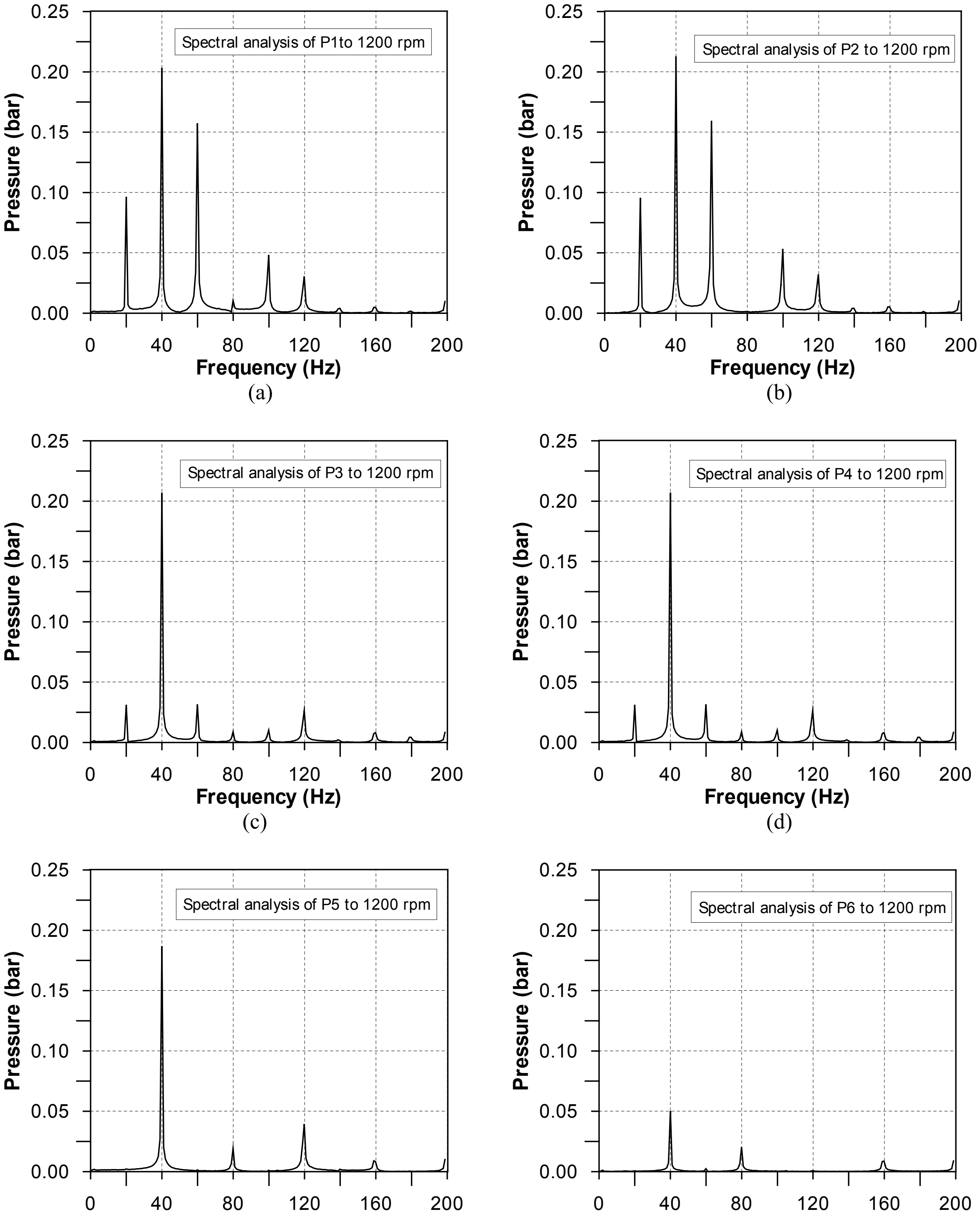
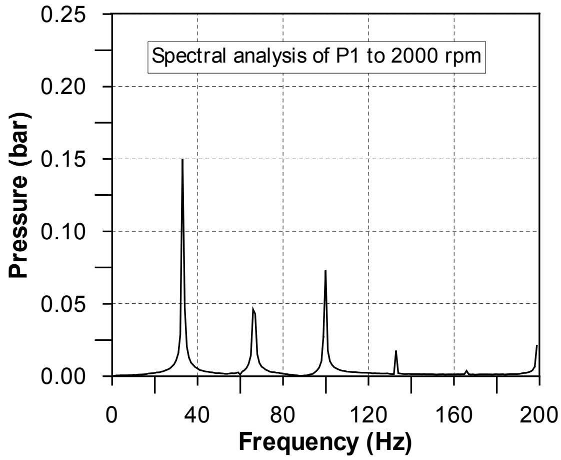


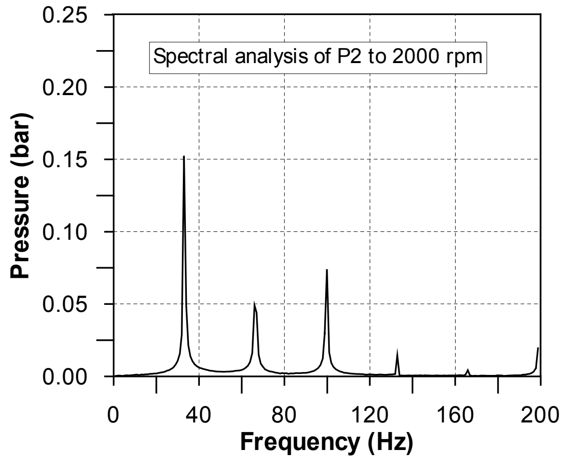
Figure 7- Pressure from transducers P1 (a), P2 (b), P3 (c), P4 (d), P5 (e) and P6 (f) versus frequency for configuration H at 1200 rpm.

Another interesting effect occurs for the spectral analysis referring to Fig. 7 (e). The peak pressure for the harmonic frequency of 60 Hz ceases to exist. As P1 is located at the secondary pipe, it suffers the influence from the two cylinders, and, thus, the response frequency is duplicated. Probably, the cause for this phenomenon is that the frequency for the primary pipe does not affect any more the frequency for the secondary pipe. The spectrum analysis of the pressure waves at 1200 rpm for configuration J is similar to that of configuration H shown by Fig. 7.

Figure 8 shows the spectrum analysis of the pressure waves at 2000 rpm for configuration H. As previously discussed, the resonance frequency for the primary pipe, considering an open-end aperture, is equal to 69.4 Hz, while the resonance frequency for the secondary pipe is 91.4 Hz. The resonance frequency for the full pipe is 39.4 Hz. The spectral analysis for P1 (Fig. 8 (a)) shows the first tree peaks higher than the other harmonics. An interesting aspect is that the fundamental frequency has an amplitude level higher than the fundamental frequency, which differs from the results at 1200 rpm, when the first harmonic had higher amplitude than fundamental frequency. The reason for this is that the engine rotational speed of 2000 rpm has a frequency around 33 Hz, that is close to the resonance frequency for the full pipe. Than, the amplitude pressure for the fundamental frequency was amplified. The amplitude pressure for transducers P1 and P2 are very similar, showing that the only difference from them is really the phase shift. An analogous conclusion can be made for the spectral analysis of transducers P3 and P4.



(a)



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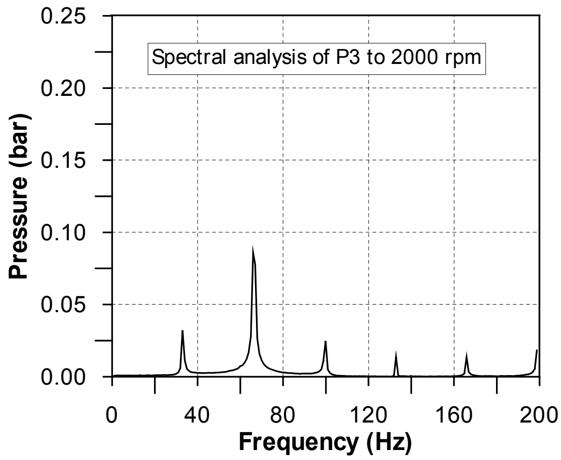
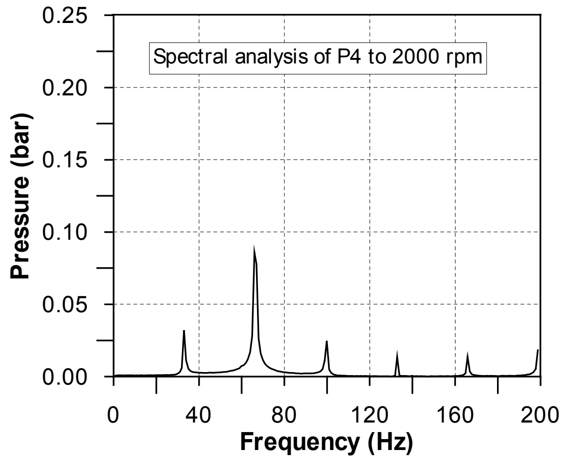


(c)



(d)

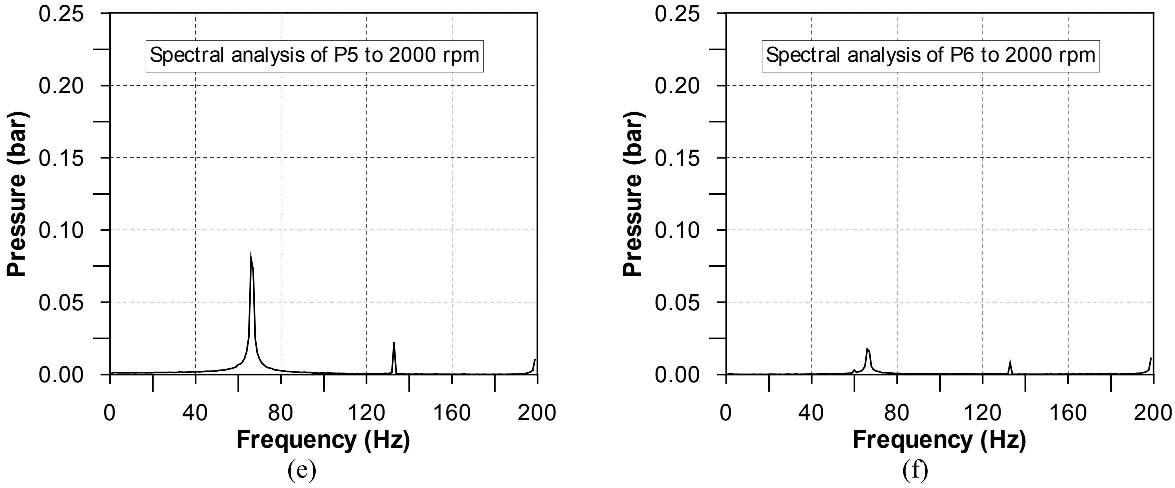


Figure 8- Pressure from transducers P1 (a), P2 (b), P3 (c), P4 (d), P5 (e) and P6 (f) versus frequency for configuration H at 2000 rpm.

7. Conclusions

- The phenomena of pressure pulse propagation and reflection in the presence of junction in the intake manifold engines have been presented and discussed.
- Experimental results were obtained for two 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.
- The junction connected at the intake system closer to the intake valve port presented a higher mass flow rate, especially for camshaft rotational speed above 1500 rpm.

8. Acknowledgement

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