EXPERIMENTAL MEASUREMENTS OF PRESSURE WAVES IN INTAKE SYSTEM OF A CFR ENGINE

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Abstract. *The understanding of the pressure waves behavior in the air intake system of an internal combustion engine has immediate practical interests and is very important to develop the system, increasing its efficiency. The periodic movement of the intake valve originates a pulsating phenomenon in the intake manifold. These pressure waves, which propagate within the system, directly affect the amount of air entering the cylinder. To understand this phenomenon it is important the experimental procedure, both to quantify the behavior and establish results for numerical, or other experimental comparisons in the future. This paper presents experimental results measurements of pressure waves and mass flow that occurs in the intake manifold system of a standard CFR (Cooperative Fuel Research) engine. The engine under consideration has a bore of 82.55 mm and a stroke of 114.30 mm.*

Keywords*: Internal combustion engines, CFR, pressure waves.*

1. INTRODUCTION

The intake manifold of an internal combustion engine plays a very important task concerning the entire system. This component performs the important role of conducting the air from the atmosphere to the cylinder with less pressure losses as possible. The periodic movement of the intake valve originates a pulsating phenomenon in the intake manifold. This pressure waves propagating within the system directly affect the amount of air entering the cylinder. Some studies about this field were already made by some authors in the past.

Morse et al. (1938) was among the first of studying the influence of acoustic vibrations derived from the movement of the valves of an ICE (Internal Combustion Engine). Benson and Winterbone (1989) mentioned that in case of a real engine, when the pressure pulse reaches the intake valve at its closure, there is an increase of amount of air admitted.

Winterbone and Pearson (1999) described construction technics to reduce sound pressure levels, and mentioned a very interesting discussion about the mechanism of reflection and transmission of sound waves at geometrical particularities.

Harrison et al. (2004a), comment on the importance of pressure waves in engine operation and discusses five procedures to make such experimental measurement, either to have good data acquisition for a project or to validate simulations. These methods can be used to measure velocity and pressure fluctuations, or to measure the rate of specific acoustic impedance of the medium. The test bench uses the single-cylinder Ricardo E6 (510 cm ^{3}).

The authors make two important settings in this work. The first affirmation says that the dynamics involved in the admission system does not differ greatly when combustion is present to when combustion is not present. This becomes important to the present work because it performs measurements without combustion. The second affirmation says that the air box placed between the entering of the admission system and the atmosphere has small effect over the system. The air box was not considered in the present work. The five measurement methods and experimental findings are presented, including comments about the strengths and weaknesses of each procedure:

a. Pressure time story analysis: it was used a Kistler 4045A2 piezoresistive absolute pressure sensor and the engine speed set at about 2000 rpm. It was also used an optical position sensor on the crankshaft. Starting from the acquired pressure data, the spectrum was obtained with an FFT of 2048 points, at a frequency of acquisition of 4096 Hz, resulting in a spectrum with a resolution of 2 Hz.

- Strengths: simple and robust;
- Weaknesses: provides information on pressure fluctuations, but does not provide information on speed fluctuations that accompany it.

b. Wave decomposition: about foundation of the method, the pressure field can be described in space and time using the positive and negative components of wave components (p^+ and p^-), and this equation can be written in terms of the frequency and spatial coordinates (k is the wave number).

$$P(\omega, x) = p^+ e^{-ikx} + p^- e^{ikx} \tag{1}$$

Thus, the pressure is monitored in two points next of each other and a pair of equations is constructed and these two can be algebraically solved.

- Strengths: directly measures the rate of specific acoustic impedance;
- Weaknesses: trusted only at low frequencies, only returns rates of pressure and speed and does not distinguish between open and closed valve.

c. Pressure drop across the inlet valve: The unsteady mass flow through the intake valve (kg/s) is commonly calculated by using the well-known model [Winterbone *et al*., 1999].

$$\dot{m} = \frac{P_C A_e}{c_C}\left[\left(\frac{2\gamma^2}{\gamma-1}\right)\left(\frac{P_1}{P_C}\right)^{\frac{2}{\gamma}}\left[1-\left(\frac{P_1}{P_C}\right)^{\frac{(\gamma-1)}{\gamma}}\right]\right]^{1/2} \qquad (2)$$

where P_C is the cylinder pressure, P_1 the intake port pressure, c_C the speed of sound in the cylinder, $A_e = A_m C$ and A_m the valve open area, C the flow loss coefficient (measured on a flow bench). This will normally be part of an engine simulation where P_C and P_1 are also calculated. However, if P_C and P_1 are in fact measured quantities then the unsteady mass flow through the valve may valve may be measured indirectly.

- Strengths: provides story of pressure and velocity as a function of time;
- Weaknesses: it works only during the portion of the cycle in which the valve is open; method is subjected to the deleterious effects of noise on pressure signals; need a very large quantity of empirical data about the engine (coefficient of losses in the valve, curves and dimensions of the valve lift).

d. Hot wire anemometry with pressure measurements: as well known, it can be used to measure the amplitude of a flow, but not its direction.

- Strengths: provides story of pressure and velocity as a function of time;
- Weaknesses: cannot be used when fuel is present in intake ducts or when there is a possibility of particulate fuels or wastes adhere to the wire. The wire has a short life, and to maintain calibration close to perfection is very hard.

e. Bidirectional static Pitot tube: two Kistler 4045A2 piezoresistive pressure sensors are mounted on a same plane, so that one measures the static pressure and the other measures a quantity that is very close to the total pressure. The decay rate of speed in this method is very close to the corresponding data of pressure. This is not true for the method using hot wire anemometer.

- Strengths: robust and good dynamic response and frequency.
- Weaknesses: requires calibration against another method of speed measuring.

Therefore, for the study of pressure waves in engines, at first glance, the first method (analysis of pressure in time) seems more useful since:

- The wave decomposition method does not work along the time period when the valve is closed, that is of great importance for the study of the admission;
- The method of pressure drop across the inlet valve requires a large amount of empirical data and applies only when the valve is open;
- The method of hot wire anemometry is difficult to implement and, if the study consider the combustion, the experiment will be affected;
- The method of bidirectional static Pitot tube, in addition to requiring another method for calibration, the structure exposed to flow cause a number of secondary reflections that can affect the analysis. Furthermore, it was considered important in this study to monitor different points in the domain, which makes the method unviable, being complicated to put several bidirectional Pitot tubes along the geometry.

This paper intends to use different lengths of intake manifolds placed on a single cylinder four stoke motored CFR engine, in order to generate mass flow and pressure data, showing a relation between them by the capture of the pulsating phenomena. Also, intends to generate data for future numerical comparisons and to investigate the influence of a mass flow sensor in the system to capture the transient phenomenon.

2. EXPERIMENTAL METHODOLOGY

The apparatus used was constructed based on a CFR engine, considering that the CFR engine has a well known configuration for being a standard engine. The specifications of the used engine are given in the Table 1.

Table 1. Specifications of the CFR engine

ASTM-CFR – engine	
Bore x Stroke (mm)	82.55 x 114.30
IVO/IVC (^{o} ATDC)	0 / 202
Displacement (cm^3)	611.30

Maximum valve lift (mm)	6.05
Intake air system	Naturally aspirated
Compression ratio	6:1
Rotation (rpm)	600

The engine was instrumented to measure its angular position, intake air mass flow rate and pressure waves. The angular position of the crankshaft was measured with an incremental encoder (Danaher Sensors & Controls, model BA 3022). The encoder was connected to the crankshaft and supplies 1733.33 pulses per revolution, with a resolution of 0.20769 degrees. The pressure waves were measured with an integrated Silicon Pressure Sensor, with signal conditioned, temperature compensated and calibrated (MPX4115), series piezoresistive transducer. The response time, defined as the time for the incremental change in the output to go to 63.2% of its final value when subjected to a specified step change of pressure, is less than 1 ms, which is adequate for these measurements.

The air mass flow rate was measured with an automotive hot film anemometer [Bosch, 2010]. This sensor has an input voltage of 5 and 12 V. The output analogic signal is related to the air flow in the intake process. The incoming air dissipates heat from the hot film, so the higher the air flows, more heat is dissipated. The differential temperature is evaluated by an electronic hybrid circuit. Both the air flow and its direction can be measured. Only part of the air-mass flow is registered by the sensor element. The total air mass flowing through the measuring pipe is determined by means of calibration, known as the characteristic-curve definition [Soriano; Rech, 2011]. The time constant of hot film anemometer, considering dynamic measurement, was determined in another work by authors [Rech 2010], being verified that sensor is 8 times faster than the necessary for these measurements. The characteristic-curve, relating output voltage of the hot film anemometer and its respective air mass flow, was taken by a standard orifice plate.

The data was acquired with a commercial data acquisition board (National instruments 6221) and its original LabView software. The values of voltage and frequency were collected as a function of the encoder angular increment and processed with the correspondent calibration curves of each sensor and converted into units of pressure and angular position.

The schematic picture of the test bench is represented in Figure 1, where the three pressure positions are shown. Three manifolds were used intending to compare three different lengths. These three configurations will be called Long pipe, Medium pipe and Small pipe from now on. When the Long pipe is used (1.36 m), all three sensors can be positioned into the apparatus. When the Medium pipe is used (0.6 m), only the positions 1 and 2 are monitored and when the Small pipe is considered (0.4 m), only the position 1 is monitored. For some analysis a mass flow sensor was used, adding 0.07 m to the system.

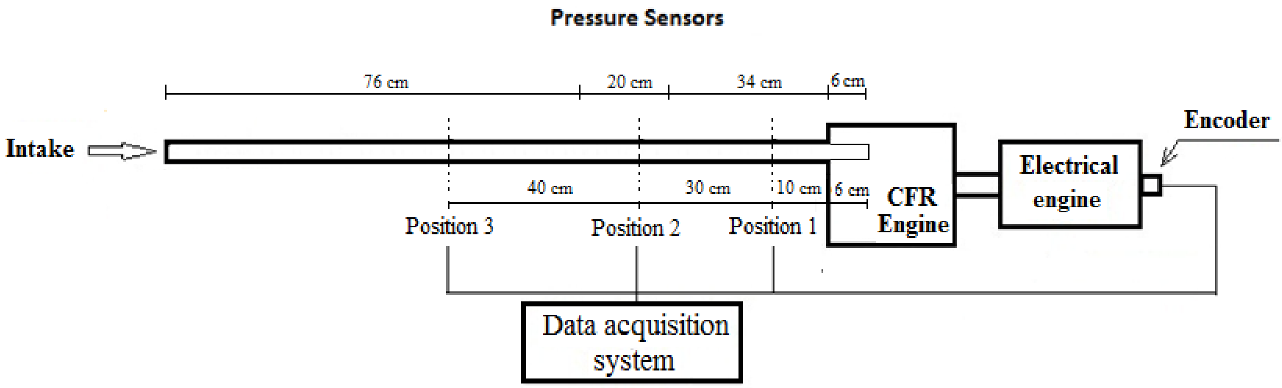

Figure 1 – Schematic picture of measurement system

3. EXPERIMENTAL RESULTS AND DISCUSSION

The pressure sensor implemented in the intake system was effective in the process of measuring the pressure throughout the air intake duct, with rapid response to the phenomena of pressure change, ease of implementation due to their small size, not requiring a dedicated system of signal conditioning. The use of an encoder coupled to the crankshaft in conjunction with a reference top dead center provides the acquisition of all signals of interest linked to any single pulse supplied from the encoder. This ensures that all values are acquired at a given value of the crankshaft angle, and the acquired data are not influenced if there is any irregularity in engine speed during a particular cycle. The Figure 2 shows the average of 30 cycles of the pressure curves versus crank angle, obtained in the Long pipe at the three positions showed in the Figure 1. The maximum mean standard deviation for this case was about 0.5 mbar, ensuring a good repeatability.

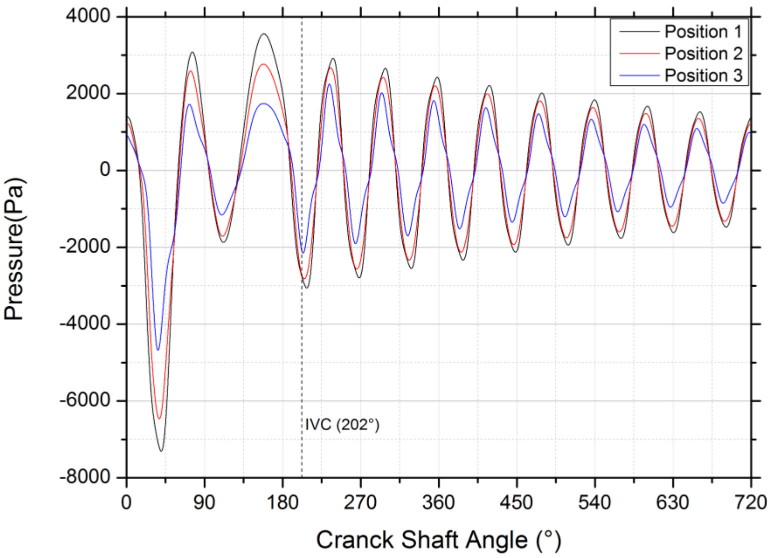

Figure 2 – Experimental results for the average of 30 cycles of pressure measurements in three different positions for Long pipe, at 600 rpm

The sensor of position 1 is closer of the valve than the others, resulting greater amplitude of the pressure behavior at this point. As the monitored position gets farther of the valve (positions 2 and 3), this amplitude diminishes. Considering this same position 1, this point has greater pressure losses in time, since its pressures variations are bigger, for consequence the losses are bigger. The pressure behavior at position 3 shows a perturbation caused by the first harmonic. As the amplitude pressure at this point is smaller, the presence of the first harmonic is clearer.

The Figure 3 shows the frequency spectrum of the pulsating pressure in the intake manifold for the Long pipe, after the inlet valve closing. Considering c the theoretical sound velocity, γ the relation between the specific heats, R the gas constant and T the measured air temperature of 291°K, the following equation provides the theoretical sound velocity value:

$$c = \sqrt{\gamma T R} \tag{3}$$

considering $R = 287$ J/kgK and $\gamma = 1.4$, it results about $c = 342$ m/s.

Based on this result, it is possible to calculate the theoretical resonating frequency of the pressure in the intake manifold [Kinsler, 1980]:

$$f = (2n+1)\frac{c}{4(L+e)} \tag{4}$$

where $n = 0, 1, 3, 5, \ldots$ and represents the harmonic frequencies and L is the length of the pipe. The letter "e" represents the end correction. The reflection in the open end is caused by suction or blowing that happens when the pulse of pressure reaches the open end of the pipe. This reflection doesn't appear immediately when the pulse reaches it, but a little later as it starts to spread out. To give a good approximation, this effect can be calculated by adding the end correction due to the size of the diameter. As the diameter increases, the reflections take more time to occur and then the resonating frequency gets lower. The value for an open-open ended pipe is given in Kinsler (1980), for an unflanged pipe, is 0.6 times the internal diameter. So, for a close-open ended pipe this end correction should be lower, since it has only one opening. The values for this particular correction vary from 0.25 to 0.4 in the literature (Gupta 2001) and in the present work was adopted 0.25 in the calculations. So, considering the Long pipe, the theoretical resonating frequency is 62.5 Hz.

The experimental result in Figure 3 shows the experimental resonating frequency value of 62.4 Hz, very close to the calculated value. The 1st harmonic appears in this figure around 186.9 Hz, that is also very close to the calculated value of 187.4 Hz ($n = 1$).

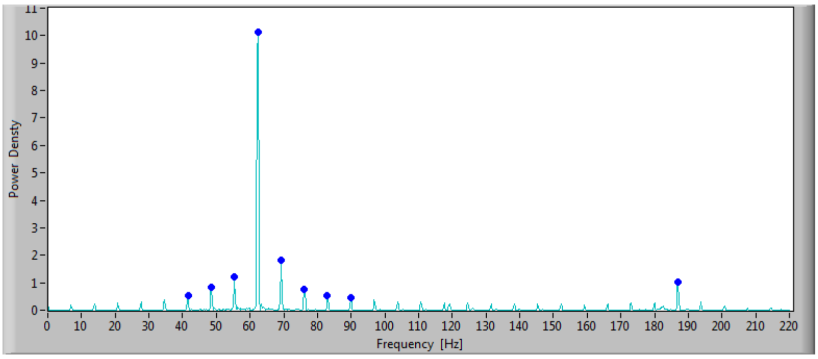

Figure 3 – Experimental results for 30 cycles of frequency pressure measurements in position 1 of the Long pipe, after the closure of the inlet valve

The Figure 4 presents the average of 30 cycles of pressure measures at position 1 considering the three pipes, after closing the valve. As already commented, the three resonating values match the theoretical calculations. Considering the pressure behavior in the three tubes in the same graphic it is possible to analyze the cause of their different behavior. It is verified that the maximum amplitude decreases as the pipe becomes shorter, as already commented, it happens because the inertial effects are smaller and the frequency increases for the fact that the distance through which the wave travels is smaller. The three resonating frequencies are very close to the theoretical values.

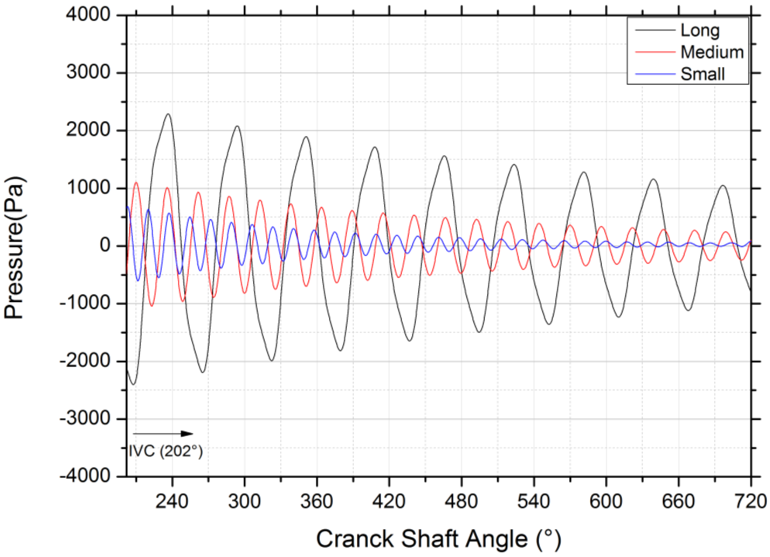

Figure 4 – Pressure at position 1 in the three pipes, after the valve is closed.

It is possible to consider different monitoring points in the same configuration and compare the frequency results intending to found any distortions in the results. If the three different positions are considered to analyze the resonating frequency it is possible to see a good agreement with the theoretical calculation also. The table 2 represents the frequencies of pressure wave propagation after closing the inlet valve to the Long pipe, measured at three positions as described in Figure 1. As previously mentioned, the pipe has dimensions of 1.36 m, and at a temperature of 18°C the speed of sound is approximately 342 m/s. Given these parameters the theoretical frequency wave propagation is 62.5 Hz. Table 2 shows the good agreement between the theoretical results and experimental ones, among the 3 sensors, including a good repeatability.

Table 2. Measurements of propagating pressure wave frequency in the Long pipe

Frequency [Hz]		
position 1	position 2	position 3
62.8	62.6	----
63.0	62.8	62.8
63.3	63.0	63.0
62.3	62.3	63.3
62.3	61.9	62.3
62.1	62.1	62.3
62.3	62.3	62.3
62.6	62.3	62.3

Likewise, the resonating measured frequency in Medium pipe (theoretical 140.5 Hz) and Small pipe (theoretical 209 4 Hz) match the theoretical calculations, showing a good repeatability (Tab. 3 and Tab. 4).

Table 3. Measurements of propagating pressure wave frequency in the Medium Pipe

Frequency [Hz]	
Position 1	Position 2
139.8	139.8
139.8	139.8
140.9	140.9
139.8	139.8
140.9	140.9
142.1	142.1
142.1	142.1
139.8	139.8
139.8	139.8
138.6	139.8
139.8	138.6
138.6	138.6
138.6	139.8
139.8	138.6
139.8	138.6
138.6	140.9
139.8	138.6
139.8	139.8
139.8	139.8

Table 4. Measurements of propagating pressure wave frequency in the Small pipe

Frequency [Hz] Position 1	211.4	206.3	208.8	208.8	208.8	211.4	208.8	211.4	208.8	211.4
	208.8	208.8	208.8	208.6	208.8	206.3	208.8	208.8	206.3	208.8
	206.3	208.8	206.3	206.3	211.4	206.3	211.4	206.3	203.9	

The Figure 5 shows the average of the results of 30 cycles for mass flow and pressure in position 1 into Long pipe. The relation between mass flow and pressure waves can be seen. It is possible to verify the backflow in the intake system when the results of mass flow are negative. The displacement between the two profiles is due to the different positions of the pressure sensor and the mass flow sensor, since the experimental construction requires it. So, the phase difference between them is due to the distance between these two sensors.

While the valve is closed an almost constant frequency can be seen in this pulsating phenomenon. Before its closing, the resonating frequency shows an enlargement. This variable frequency while the valve is opened occurs because the characteristic length of the domain is bigger, since the pressure does not encounter only the valve (since it is opened) and reflect inside the chamber also. After the valve is closed, around 0.05 s or 200° of the crankshaft, this frequency depends on the length and diameter of the pipe, and on the sound velocity relatively to the movement of the gas.

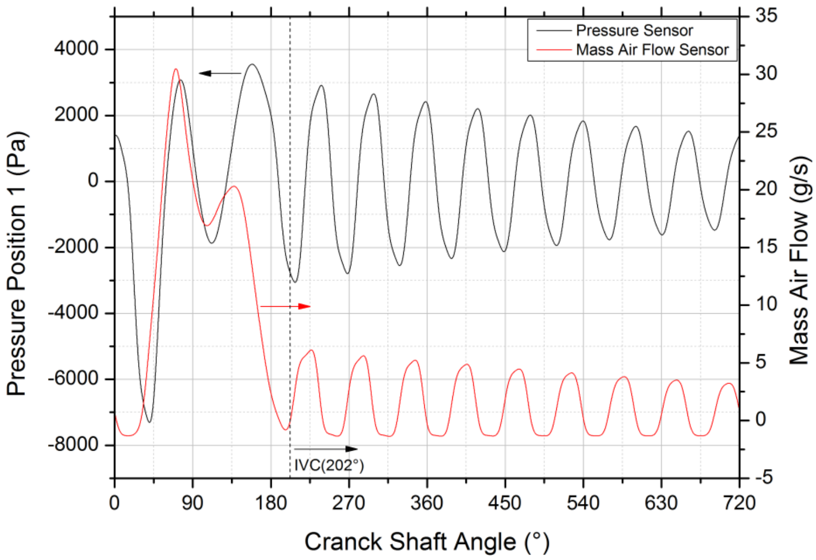

Figure 5 – Experimental results for the average of 30 cycles of pressure measurements and mass flow in the Long pipe.

The placing of the mass flow sensor adds great complexity to the behavior of the pressure since it reflects in several different internal geometric irregularities, changing sections, etc. The influence of the positioning of the mass flow sensor can be seen in Figure 6, where the intake pressure pattern, with and without de mass flow sensor, is plotted in the same graphic. The addition of the mass flow sensor implies an increase of the pipe representative length and, for consequence, there is an increase of amplitude of the pulsating pressure and a decrease of the resonating frequency. The decrease of the resonating frequency happens since it is inversely proportional to the length of the pipe. The increase of amplitude pressure behavior is due to addition of frictional losses and, at the same time, the inertial effects are increased with the elongation of the pipe.

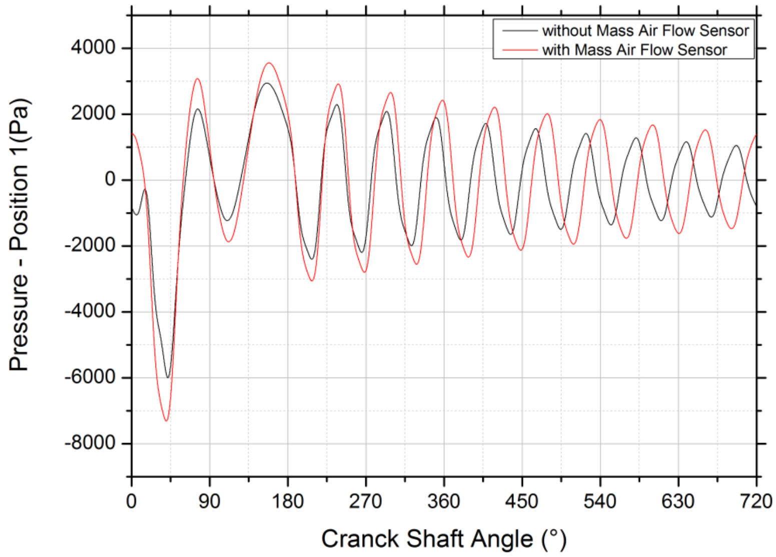

Figure 6 – Pressure behavior with and without the mass flow sensor (pipe C).

With the addition of the mass flow sensor, the representative length increases 0.07 m, and the resonating frequency that, without the mass flow sensor was 62.4 Hz, becomes 59.5 Hz, corroborating the Equation 2.

4. CONCLUSIONS

In the present work, the pressure in the intake manifold of an internal combustion engine was measured. The pressure was monitored in different positions considering different pipe lengths. The theoretical values of resonating behavior were corroborated and the results matched. The exponential decay of the pressure in time was verified as well as the decay of pressure along the pipe.

It was seen that a well-established pulsating mass flow phenomenon is present, even when the valve is closed. The presence of the mass flow sensor changes the behavior of the pressure. The present work is very useful, considering not only the amount of generated data and for consequence a plenty of discussions, but also for future numerical comparisons that will be performed.

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