

INFLUENCE OF TRANSIENT PHENOMENA IN THE DISCHARGE COEFFICIENT THROUGH THE INTAKE VALVE IN AN INTERNAL COMBUSTION ENGINE

Hanriot S.M.* , Queiróz J.M. and Maia C.B.

*Author for correspondence

Department of Mechanical Engineering,
Pontifical Catholic University of Minas Gerais,
Belo Horizonte, MG
Brazil,
E-mail: hanriot@pucminas.br

ABSTRACT

The project of engine intake systems involves optimization of parameters such as the pipe length and diameter, junctions, and opening and closing times of the intake and exhaust valves. The correct sizing leads to an increase of the air mass admitted to the cylinders at the desired engine operational conditions. A suitable design of the intake valves in internal combustion engines is one of the factors that maximize the amount of intake air mass to the cylinder. The parameter that determines the maximization of the mass flow through the valves is called discharge coefficient. The mass flow through the valve is usually described by the compressible flow equation through a restriction, based on a dimensional analysis of an isentropic flow. In the present work, pressure variations caused by the valve movement were investigated experimentally considering an intake system. The objective was to study and compare the dynamic response of the flow through the intake valve. For this purpose, curves of mass flow rate and the dynamic pressure in several locations of the intake system were obtained. The experimental data were obtained from the intake system connected to a cylinder head. 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 correct design of the intake valve affects positively the air mass flow rate.

INTRODUCTION

An appropriate intake manifold design should provide a satisfactory engine intake air charge while reducing pressure losses along the intake system. The intake air mass flow must be equally distributed through the engine cylinders. Due to piston and intake valve movements, 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 [1,2, 3]. 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. Usually the discharge coefficient is obtained under steady state conditions and the inertial effects aren't evaluated [4,5].

Other studies consider a filling and emptying analysis of an internal combustion engine. The simulation of pressure waves in inlet and exhaust manifolds of internal combustion engines was presented by [6]. In this study the system is analysed using a frequency approach, and the inlet system is then characterized by its geometrical characteristics as well as the fluid characteristics. Both numerical and experimental methodologies were used. The authors concluded that a good agreement was obtained between the experimental and the numerical results, and that this approach gives the possibility to reduce the computational times.

In [7] a discussion is made on the discharge coefficient. Sliding mode observer (SMO) scheme has been used for online estimation of discharge coefficient of throttle body at the intake manifold of gasoline engines.

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.

In [8] it was 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.

NOMENCLATURE

\dot{m}_{real}	[kg/s]	Real mass flow rate
C_D	[-]	Discharge coefficient
\dot{m}_{isen}	[kg/s]	Isentropic mass flow rate
A_{ref}	[m ²]	Reference area
k	[-]	Specific heat ratio
P_T	[Pa]	Pressure downstream of the intake valve
P_o	[Pa]	Pressure upstream of the intake valve
R	[J/kg.K]	Gas constant
T	[K]	Temperature
T_o	[K]	Ambient temperature
D_v	[m]	Valve diameter
L_v	[m]	Valve lift
β	[-]	Expansion ratio coefficient
ρ	[kg/m ³]	Air density
P_{ATM}	[Pa]	Atmospheric pressure
P	[Pa]	Pressure
\dot{m}	[kg/s]	Mass flow rate
ΔP	[Pa]	Pressure difference
n	[rpm]	Engine frequency
c	[m/s]	Velocity of sound
f	[Hz]	Frequency
L	[m]	Pipe length
M	[kg]	Mass of fluid
a	[m/s ²]	Acceleration
V	[m/s]	Flow velocity

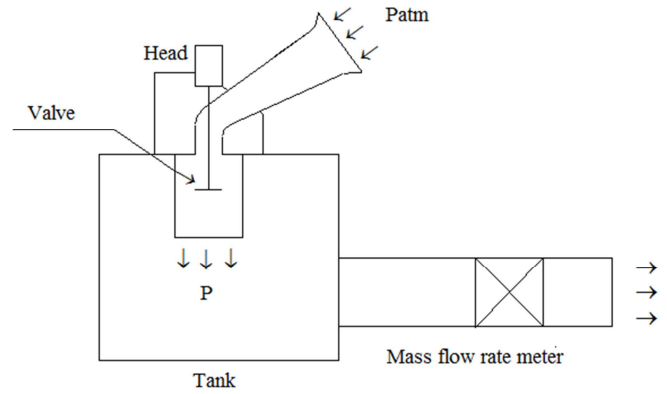


Figure 1 - Equipment for steady state flow measurement

Typically it is imposed a constant value for the expansion ratio (β), defined as the ratio between the atmospheric pressure and the pressure inside the tank. This coefficient is kept constant for all elevations of the valve by adjusting the pressure in the equalization tank.

When the pressure differences between the environment and the equalization tank is 5 to 10 kPa, generally the air flow compressibility can be neglected, and equation (2) can be expressed by [12]:

$$\dot{m}_{real} = C_D A_{ref} \sqrt{2\rho(P_{ATM} - P)} \quad (3)$$

FUNDAMENTAL CONCEPTS

The mass flow through the valve is usually described by the compressible flow equation through a restriction. This equation is derived from dimensional analysis of an isentropic flow, using experimental data of the mass flow rate to obtain the discharge coefficient. The relationship between the actual and isentropic mass flow is given by [9]:

$$C_D = \frac{\dot{m}_{real}}{\dot{m}_{isen}} \quad (1)$$

The flow through the valve is given by [10]:

$$\dot{m}_{real} = \frac{C_D A_{ref} P_o}{(RT_o)^{1/2}} \left(\frac{P_T}{P_o}\right)^{1/\kappa} \left\{ \frac{2\kappa}{\kappa-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{(\kappa-1)/\kappa} \right] \right\}^{1/2} \quad (2)$$

The value of C_D and the reference area are related; their product $C_D A_{ref}$ is the effective flow area of the gas. The choice of the reference area is arbitrary [11]. The literature recognizes the following parameters: the area of the valve, the minor area of the valve seat, and the area of the curtain, defined as the product of the valve lift L_v and the diameter D_v .

The discharge coefficient is generally obtained experimentally by coupling the head to a tank of large volume (Fig. 1). The test essentially consists in forcing the air through a flow bench that reproduces the values of valve lifts, the geometry of the conduits in the cylinder and the surrounding of the valve.

In steady state flow only viscous forces affect the determination of C_D . In a real situation, the periodic opening of valves requires a restart of the movement of the gas inside the intake pipe. Therefore, inertia forces caused by the gas pulsation must be considered.

It is observed that inertia forces tends to decrease the discharge coefficient (corresponding to an increase in the total resistance), and so reducing the mass flow [3,13].

When these forces exist, a dynamic discharge coefficient is usually defined, different from the stationary one. The dynamic coefficient is defined exactly as the stationary (see Equation 1). The difference is that the real mass flow rate varies with time. Several authors [14, 15] have shown that the dynamic discharge coefficient is smaller than the stationary coefficient.

The average dynamic coefficient is obtained considering the mean value over a complete cycle of the crank angle [16]:

$$\bar{C}_D = \frac{1}{2\pi} \int_{-\pi}^{\pi} C_D(\theta) d\theta \quad (4)$$

Under normal working conditions of the engine, both the valve lift and the pressure difference are affected by engine speed. In a first analysis, it is desirable to "separate" the effects of variation of rotation from the effects of variation of pressure. This can be accomplished by keeping the pressure difference constant while maintain the valve movement, for a given engine speed.

RELATION BETWEEN THE DYNAMIC AND THE STATIONARY DISCHARGE COEFFICIENT

In this work a dimensional analysis was performed in order to minimize the number of variables to be analysed.

In a steady state flow bench usually the discharge coefficient and the head losses are determined. In a transient flow bench, the speed of the engine and the valve displacement are also considered.

In the case of stationary flow, dimensional analysis leads to the following relationship between the flow and the pressure difference [16]:

$$\frac{\dot{m}\sqrt{T}}{P} = f\left(\frac{\Delta P}{P}\right) \quad (6)$$

The discharge coefficient is independent of the expansion ratio, β , defined as [9,16]:

$$\beta = 1/[1 - (\Delta P / P)] \quad (7)$$

Under transient conditions, dimensional analysis leads to a relation as [16]:

$$\frac{\dot{m}\sqrt{T}}{P} = f\left(\frac{n}{\sqrt{T}}, \frac{\Delta P}{P}\right) \quad (8)$$

The term in the left in the equation 8 is related with the Mach number (Ma):

$$Ma = \frac{V}{\sqrt{kRT}} = \frac{\dot{m}}{\sqrt{kRT}} = \frac{\dot{m}.R.T}{P.L^2} = \frac{\dot{m}.\sqrt{R}.\sqrt{T}}{\sqrt{k}.P.L^2} \propto \frac{\dot{m}\sqrt{T}}{P} \quad (9)$$

The term $\frac{n}{\sqrt{T}}$ that appears in eq. (8) is the ratio of the exciting frequency of the source (in this case the frequency of the valve) and the frequency of the duct, given by [17]:

$$f = \frac{c}{4L} = \frac{\sqrt{kRT}}{4L} \quad (10)$$

The division of n by eq. (10) leads to:

$$\left[\frac{4nL}{\sqrt{kRT}}\right] \propto \left[\frac{nL}{\sqrt{kRT}}\right] \propto \left[\frac{n}{\sqrt{T}}\right] \quad (11)$$

The pressure forces are related to:

$$[F_P] = [\Delta P.L^2] \quad (12)$$

The inertia forces are:

$$F_i = m.a \quad (13)$$

Dimensionally "a" is related to:

$$[a] \propto [L.n^2] \quad (14)$$

The relationship between the mass and the volume is:

$$m = \rho.L^3 \quad (15)$$

This leads to the dimensional relationship:

$$[m] = [\rho.L^3] = \left[\frac{P}{RT}.L^3\right] \quad (16)$$

The inertial force is given by the dimensional relationship:

$$[F_i] = \left[\frac{P}{RT}.L^3.L.n^2\right] \quad (17)$$

The ratio between pressure forces and inertia forces is given by the dimensional relationship:

$$\left[\frac{F_P}{F_i}\right] = \left[\frac{\Delta P.L^2}{\frac{P}{RT}.L^3.L.n^2}\right] = \left[\frac{\Delta P.R.T}{P.n^2.L^2}\right] \propto \left[\frac{\Delta P.T}{P.n^2}\right] = X \quad (18)$$

The square of the ratio between the inertial forces and pressure forces is:

$$\sqrt{\frac{F_i}{F_P}} \propto \left[\frac{n.\sqrt{P}}{\sqrt{\Delta P.T}}\right] \quad (19)$$

The discharge coefficient is proportional to:

$$[C_D] = \frac{\dot{m}}{A_R.V_{is}.\rho} \propto \dot{m}\sqrt{\frac{T}{P.\Delta P}} = Y \quad (20)$$

where the isentropic velocity is :

$$V_{is} = \sqrt{\frac{2.\Delta P}{\rho}} \quad (21)$$

The resistive force, F_R , is related to:

$$[F_R] \propto \left[\frac{\rho.V^2}{2}.L^2\right] \propto \left[\frac{\dot{m}^2.R.T}{P.L^2}\right] \quad (22)$$

The ratio between the forces of resistance and inertia forces is given by the dimensional relationship:

$$\left[\frac{F_R}{F_i} \right] = \left(\frac{\dot{m}.T}{P.n} \right)^2 \cdot \left(\frac{R}{L^3} \right)^2 \propto \left(\frac{\dot{m}.T}{P.n} \right)^2 \quad (23)$$

When the pulsating phenomena are not considered,

$$F_P = F_i + F_R \quad (24)$$

or:

$$\frac{F_R}{F_i} = \frac{F_P}{F_i} - 1 \quad (25)$$

Substituting eq. (25) and (20) in eq. (27) the equation can be written as:

$$\left[\frac{\dot{m}.T}{P.n} \right]^2 = K_1 \cdot \left[\frac{\Delta P.T}{P.n^2} \right] - K_2 \quad (26)$$

Equation 26 relates to the ratio between resistance and inertia forces to the ratio between pressure and inertia forces. In this situation, the phenomena of wave (pressure swing) are not considered. At low rotational speeds and short inlet conduits, the pulsating phenomenon is small in relation to the other forces. Thus, it is expected that the relationship between parameters obtained from eq. (26) is linear. When the pulsating phenomenon significantly affects the flow, the relationship between the parameters deviates from a straight line.

EXPERIMENTAL SETUP

The experimental data were obtained in a flow bench, designed specially to perform non stationary tests (Fig. 2). The apparatus allows flow rate measurements through the intake or exhaust pipe, with the valve in movement, at a constant pressure drop throughout the system [13]. The major advantage of the flow test bench is that it enables the tuning frequencies of manifold designs and the rotation speed of the engine.

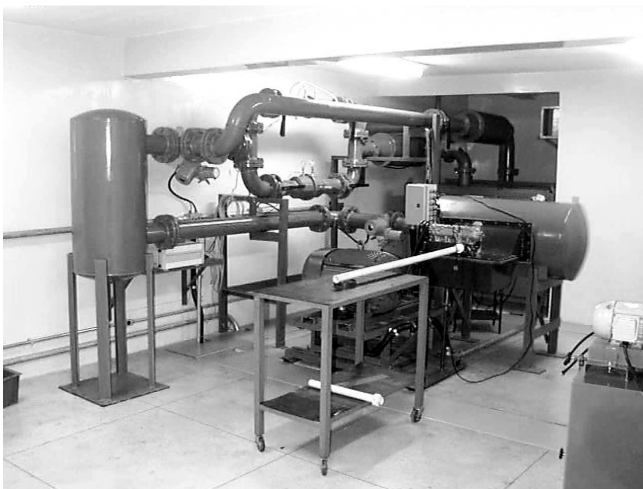


Figure 2 - Experimental rig

The constant pressure drop is obtained connecting a big reservoir to the intake system. Thereby, the pressure is equalized at that point. At the other side, the reservoir is connected to a blower that works at constant rotational speed. The reservoir has a volume of around 350 litres, 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.

One laminar flow meter is used to determine the mass flow rate. In this sensor, the fluid is forced through small passages that keep the flow laminar. Thus, the mass flow rate is proportional to the pressure difference through the meter. The instantaneous pressures were measured by piezoresistive pressure transducers, with a working range from 0 to 1 bar. Platinum RTD sensors were used to measure the temperature, with a working range between 0 and 60 °C. A four-cylinder, 1.0 liter engine head was fixed to the dumping tank. Only the intake valve of the first cylinder was operating, while the other valves remained closed. Initial tests were performed with three intake pipes with the same diameter (35 mm) and different lengths: 15 mm (named tube 0), 1000 mm (tube 1), and 2000 mm (tube 2). The tubes were used under the same test conditions, however alternately. (see figure 3).

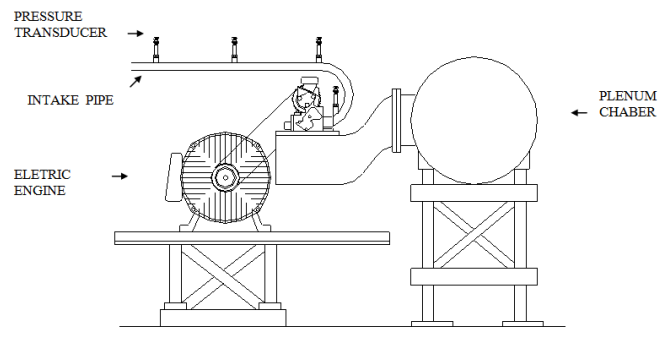


Figure 3 - Schematics of flow bench and the intake pipe

Camshaft rotational speed was varied from 600 to 2400 rev/min, in steps of 100 rev/min. The maximum rotation of 2400 rpm was chosen to not compromise the integrity of the test head. The pressure transducer was installed closer to the intake port (153 mm from the valve port) to measure pressure fluctuations. Pressure data acquisition was 10 kHz obtained for two engine cycles. Pressure difference between the chamber and the ambient was kept constant at 230 mmHg (0.306 bar).

RESULTS

The effects of inertia and resistance forces can be seen in Fig.4 (900 rpm) and Fig. 5 (2200 rpm) for the tube 1. The inertia forces are related with the pulses of pressure. The intake valve opens at 159° and closes at 301°. No pulse is observed in this range; the pressure decreases during the opening period and increases during the closing period of the valve. When the valve is closed, the pressure waves indicate a pulsating flow.

Moreover the resistance forces work reducing the pressure amplitude. The number of pulses is related with the tube length.

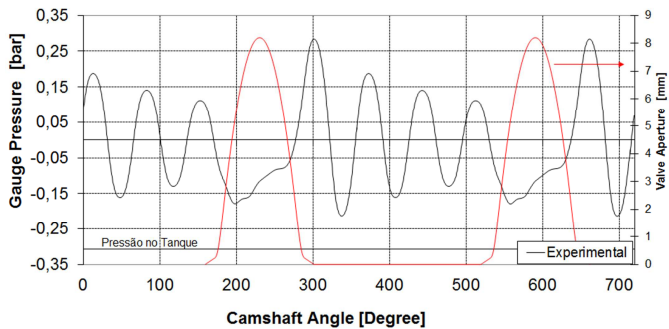


Figure 4 – Relative pressure vs. camshaft angle

When comparing Fig.4 and Fig.5 it can be noticed that the camshaft rotational speed influences the flow behaviour. The number of pulses decreases and the amplitude increases with the speed.

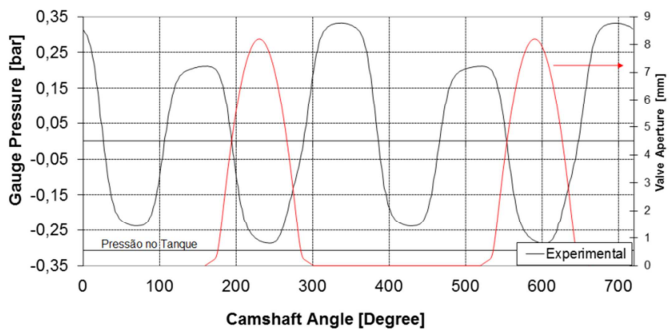


Figure 5 – Relative pressure vs. camshaft angle for 2200 rpm

Figure 6 shows the pulsating pressure for the tube 0 for 900 rpm. Basically there is no friction loss in the tube, and no pressure wave is observed during the intake valve closing period. In this case, the difference between the transient and the tank pressures is the largest among all tube configurations for the opening valve period. This difference is mainly related with the mass flow rate; the greater the pressure difference, the greater the mass flow rate (Fig. 7).

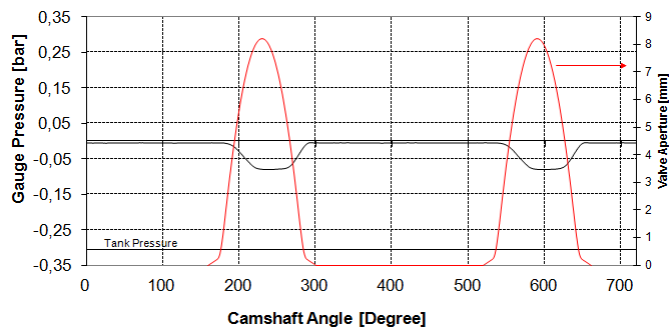


Figure 6 – Relative pressure vs. camshaft angle for 900 rpm

Figure 7 shows the influence of the camshaft rotational speed on the mass flow rate, from 400 to 2400 rpm for the tubes 0, 1, and 2. The general behaviour of the mass flow rate can be associated with the pulsating flow and friction losses. The greater mass flow rate is observed for the lower values of speed rotation, due to the lower friction losses [13,16]. One interest aspect regards to the tube 0, with the smallest intake pipe. Since the length is very small, there is almost no friction loss. The curve is practically a straight line. As aforementioned, the peaks and valleys of the curves are related with the pressure wave inside the pipe.

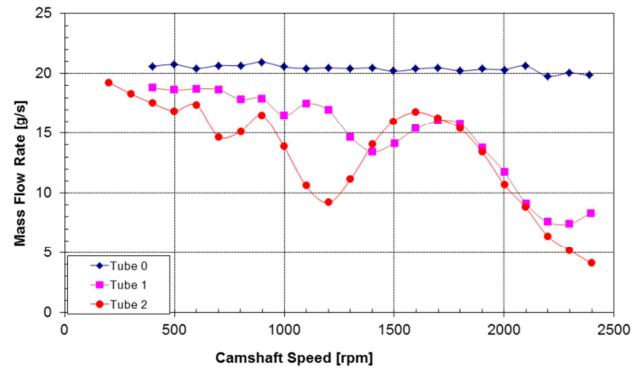


Figure 7 - Mass flow rate vs. camshaft speed,

Figure 8 shows the variation of the average dynamic discharge coefficient of tube 1 as a function of the camshaft rotational speed. Since there is a direct relationship between the dynamic discharge coefficient and the mass flow rate, the behaviour of the discharge coefficient presents the same trends observed for the mass flow rate (Figure 7).

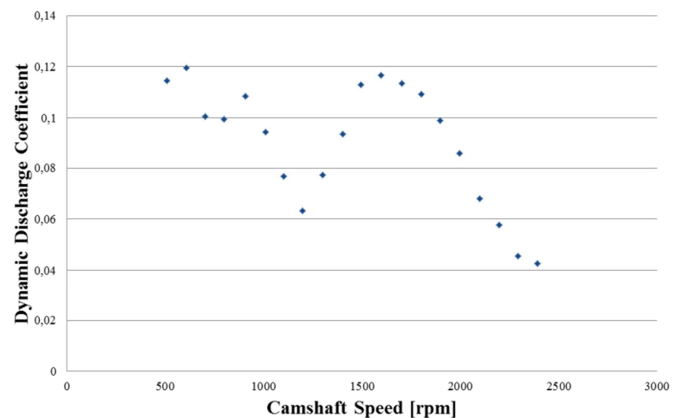


Figure 8 - Variation of the dynamic discharge coefficient as a function of camshaft rotation

The ratio between the pressure and inertia forces is inversely proportional to the square of the camshaft speed (Eq.18). The stationary discharge coefficient only takes into account the resistance forces. Considering the system operating with zero rotation, the dynamic discharge coefficient would tend to infinity. Thus, a comparison between stationary and

dynamic discharge coefficients can be made with a low speed camshaft condition.

Figure 9 shows the behaviour of a direct relation with the dynamic discharge coefficient (Y, defined in Eq. 20) and a direct relation with the ratio between pressure and inertia forces (X, defined in Eq.18). The results were obtained in an experiment conducted in the flow bench using the lowest possible camshaft rotation, equal to 180 rpm. It is important to notice that the abscissa is shown on a logarithmic scale. It can be seen that the dynamic coefficient is always lower than the stationary coefficient. The difference is associated to the pressure oscillations representing the pulsating phenomena.

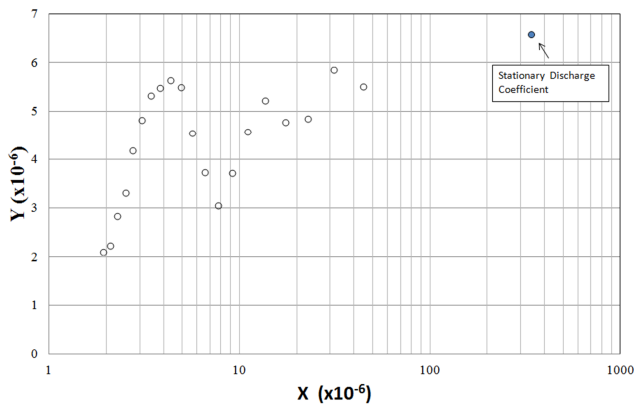


Figure 9 - Relationship between the stationary and dynamic discharge coefficient with the inertial and pressure forces

Figure 10 presents the behaviour of the ratio between resistance and inertia forces ($K * F_p / F_i$) and the ratio between pressure and inertia forces ($C * F_r / F_i$), for the tubes 0, 1 and 2. It can be seen that for low rotational speeds and short tube lengths, the relationship between the forces ratios is linear, as predicted by Eq. 26.

The influence of the pulsating phenomenon is stronger when there is a greater linearity between the forces ratios previously described. In this case, the tube 0 has the greater linearity, indicating smaller effects of pressure waves.

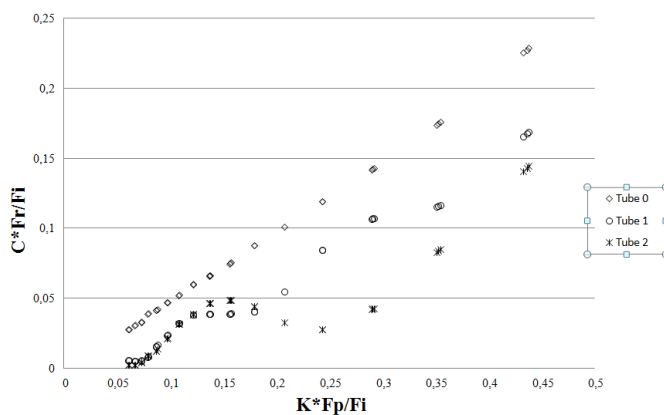


Figure 10 - Relation between forces ratios

CONCLUSIONS

In this paper the influence of transient phenomena in the discharge coefficient through the intake valve in an internal combustion engine was experimentally evaluated. The main conclusions are stated:

- From the data obtained, it was possible to identify the occurrence of pressure waves inside the intake pipes with different lengths;
- A pulsating flow was observed when the intake valve is closed. Resistance forces work reducing the pressure amplitude during the valve closing period;
- The camshaft speed influences the pressure waves and the mass flow rate. The number of pulses decreases and the amplitude increases with the speed. Also, greater mass flow rates are observed for the lower values of speed rotation, due to the lower friction losses;
- The discharge coefficient presents the same trends observed for the mass flow rate;
- For short tube lengths, the influence of the pulsating phenomenon is lower;
- There is a greater linearity between the ratio between resistance and inertia forces and the ratio between pressure and inertia forces for small camshaft speeds.

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