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Analytical and Experimental Study of Discharge Flow Behavior Provided by Electronically Controlled Valves in Hermetic Compressors.

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ABSTRACT

Trends in refrigeration market are forcing hermetic compressor manufacturers to increase the technical performance of their products as never before. New technologies are showing up as alternatives to face this challenge and help refrigeration compressor industry cope with this scenario. Among many possibilities electrically commanded valves is one that might be carefully studied and evaluated as a trend on refrigeration segment. This work studied analytical and experimentally the gas flow behavior along the discharge system of a serial hermetic compressor provided with an electrical commanded discharge valve. An incompressible unsteady viscous flow model was developed and applied to estimate gas pressures along discharge components in time domain. Experimental and calculated data were compared and presented good agreement in terms of absolute values and profiles, in spite of the relatively simple model employed.

Keywords: analytical model, electrical valve, discharge system, dynamic, pulsating flow.

1. INTRODUCTION

The investigation of gas flow behavior along the discharge system of a household hermetic compressor, under the action of an electrically commanded discharge valve, is the aim of this work. The effect caused by the valve actuation on chambers and ducts positioned after the discharge valve was studied theoretical and experimentally. The fast movements of opening and closing are remarkable features of this sort of valve; so strong disturbances are expected on the flow pressures behavior as they are already observed with ordinary reed valves.

Reed valves are commonly applied to refrigeration compressors in general. Studies are presented in the literature considering the gas flow on suction and discharge lines of compressors provided with reed valves. Pelagaly *et al.* (2000) simulated the gas flow from suction to discharge considering valves, chambers and ducts along the way, simulated and measured data were compared showing good agreement. Deschamps *et al.* (2002) developed a similar work on the suction muffler comparing Computer Fluid Dynamic (CFD) and Acoustic models data with experimental results. A lack of information about electrically commanded valves in refrigeration compressors was observed especially in the hermetic reciprocating segment. This fact is not a surprise as cost and technical challenges like sensors, logics, materials, dynamic response and others are strong barriers to the evolution of this technology in refrigeration compressors; what is common on early stages of new technologies.

The balance between analytical model complexity and accuracy of results was another point of interest in this work. CFD has been proving to be a fantastic tool to solve fluid dynamic problems and still have a great unexplored potential in the positive displacement compressors segment, as stated by Shiva (2004). The CFD solution implemented for the oil pump system of a reciprocating compressor, developed by Lückmann *et al.* (2009), is an example of this potential where the climbing oil flow was evaluated at starting condition. Gas flow analysis of reciprocating compressors employing three dimension (3D) techniques has been published and they generally rely on one dimension (1D) model to represent reed valve dynamics, as done by Bonnefoi *et al.* (2008). Pereira *et al.*

(2008) developed 1D, 2D and 3D models for household compressor performance prediction; close results were verified among them for different working conditions and component modifications. Lang (2008) simulated the suction muffler of a hermetic reciprocating compressor coupling different dimension models, allocating them according to the interests of the analysis. Park *et al.* (2008) developed a 1D model for compression process, valves and discharge system of a refrigeration compressor in order to investigate the cause of noise at around 250Hz due to discharge pressure pulsation.

Less complex models can be significantly improved if some experimental information were used to determine component and/or system parameters. This procedure usually upgrades model accuracy but restricts its validation to the components or systems experimentally evaluated. Nagy *et al.* (2008) developed a 1D model to check dynamic valves behavior in a serial hermetic refrigeration compressor and verified the benefits of the experimental parameter adjustment. Optimization procedures and sensibility analysis became more practical and/or feasible when working with less time expensive models because many conditions have to be evaluated for concluding remarks, as developed by Cavalcante *et al.* (2008).

In the present work a 1D model of the discharge system of a serial hermetic refrigeration compressor was developed to study the effect of an electrically commanded discharge valve. The discharge line was divided in sub components and their hydraulic characteristics were experimentally determined in steady state condition. This procedure led to an accurate determination of their friction factor coefficients, which could not be obtained from the classical literature due to non-standard component shapes. This characterization was employed in the dynamic analysis where mass balance and momentum equation were used to set an equation system solved on time domain.

2. ANALYTICAL MODEL

The first part of the Analytical Model development was the characterization of each sub component of the whole discharge line. These sub components are depicted in Figure 1 where the volume occupied by the gas in the discharge system can be seen in its actual shape in a perspective view, on the right side the schematic 1D model drawing is presented.

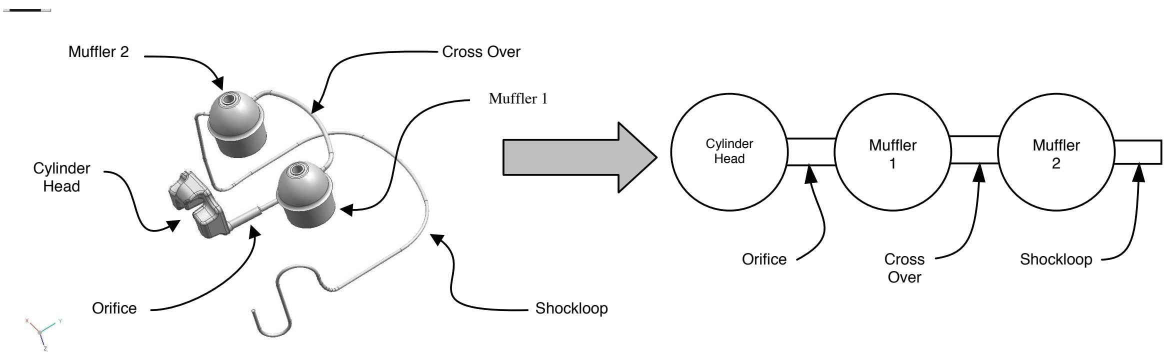

Figure 1: The internal volume of discharge system and its schematic view.

2.1 Component Characterization

Friction effects prevail in components where gas velocities assume high values like orifice, cross over and shockloop. Regular equations were assumed in the estimation of pressure drop on distributed (Equation 2) and localized (Equation 1) component parts in steady state. Ordinary components have their friction factor and friction loss coefficients available in the classical literature. The coefficients of non-standard components were experimentally determined by means of equation fitting methods based on least square techniques to minimize the error between measured and calculated data. Components like cylinder head and mufflers are reservoirs and the velocities inside them were assumed to be homogeneous and low enough to have the friction effects neglected.

$$\Delta P = -k\rho \frac{V^2}{2} \tag{1}$$

$$\Delta P = -f\left(\frac{L}{D}\right)\rho\frac{V^2}{2} \tag{2}$$

2.2 Transient Approach

The transient analytical model was developed based on a 1D approach applying the Mass Balance (Equation 3) and Momentum Equation (Equation 4) in their integral form. Control volumes evolving each of the components were defined and the mass balance and momentum equations were conveniently applied. Orifice, cross over and shockloop are components where the gas assumes high velocities gradients and the inertia effects became important, so momentum equation was applied. Cylinder head and mufflers are components were the velocities are relatively low and inertia effects are less important, so just mass balance equation was applied. The shear stress at wall surface was calculated according to Equation (5) applicable to fully developed turbulent flow. Energy equation was not applied as the tests were conducted with very low temperature gradients, gas and ambient were at close temperatures and heat transfer effects could be feasibly neglected.

$$\frac{\partial}{\partial t}\int_{CV}\rho.dv + \int_{CS}\rho.\vec{V}.d\vec{A} = 0 \tag{3}$$

$$\sum\vec{F} = \vec{P}_1 A_S + \vec{P}_2 A_S + \vec{\tau} A_W = \frac{\partial}{\partial t}\int_{CV}\vec{V}.\rho.dv + \int_{CS}\vec{V}.\rho.\vec{V}.d\vec{A} \tag{4}$$

$$\tau = \frac{D}{4}\left(-\frac{dP}{dx}\right) \tag{5}$$

The following assumptions were assumed in the model development:
1– Constant velocity profile and physical properties at control volumes entrance and exit,
2 - Refrigerant properties were considered constant in ducts and tubes and were evaluated at average component pressure,
3 - Refrigerant properties are function of time,
4 - Mass variation inside the orifice, cross over and shockloop are negligible (dM/dt = 0),
5 – The flow is incompressible and viscous,
6 – Gravitational forces are neglected,
7 – Refrigerant properties are considered spatially homogeneous in chamber volumes and just depend on time,
8 – Friction factor coefficients obtained in stationary condition were applied on transient approach,
9 – Flows are fully developed,

Two boundary conditions were established, one at the inlet and other at the exit of discharge system. To describe the pulsating mass flow rate delivered by the discharge valve to cylinder head a function was created based on the average flow rate and valve open period. Mass flow is assumed to describe a square profile along time according to valve fast movement, as can be seen on Figure 2.

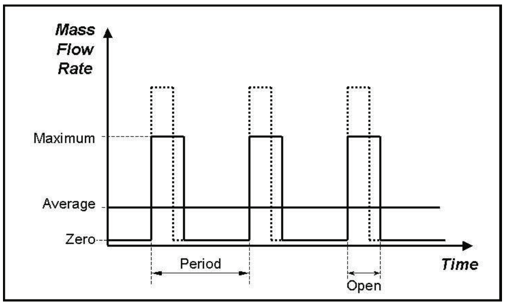

Figure 2: Mass flow rate delivered by the discharge valve to cylinder head.

The "Period" indicated in the time axis represents the complete cycle time where the valve opens once. The interval the valve stays opened is indicated as "Open", where the maximum mass flow rate is reached. Assuming a square profile the maximum mass flow rate depends on valve open time duration; if it is reduced the maximum mass flow must be increased in the same amount (described in dashed line) in order to give the same average flow rate along the cycle. This function describing mass flow rate was the boundary condition applied at discharge system entrance. The other boundary condition assumed was constant pressure on system exit.

3. NUMERICAL SOLUTION

The system of 6 differential equations defined by the application of mass balance and momentum equations on the components was programmed and solved in the Engineering Equation Solver (EES) software. The solution method employed was the 4^{th} order Runge Kutta algorithm implemented in software library. The solution was developed on time domain within the interval from 0 to 1s with a step size of integration of 0,0001s. The method took around 150s to find the solution with the following computer configuration: Intel Pentium, CPU 2.8GHz and 785.416KB RAM. EES provides transport and thermodynamic properties of refrigerants from a built in library based on Refprop routine developed by the National Institute of Standard and Technology (NIST). The advanced method provides very accurate property data for pure refrigerants and refrigerants mixtures.

4. EXPERIMENTAL PROCEDURE

4.1 Equipment

The schematic view of the experimental apparatus built to perform the tests can be seen on Figure 3. It is a closed circuit designed to run steady state and dynamic tests with R134a. Pressure sensors were installed in the points indicated P0, P1, P2 and P3, these measurements made possible the pressure drop evaluation of orifice, cross over and shockloop. The compressor provides gas circulation along the circuit and a heat exchanger was included for temperature stabilization purposes. To eliminate any possible pulse generated by the compressor on the tested system a plenum was included after the heat exchanger. A flow rate meter measured mass flow in the circuit and also evaluated gas temperature. Just before the tested discharge system (surrounded by a dashed line) the electrical commanded valve was positioned. Steady state tests were done without the electrical valve and waveform generator; they were applied just for dynamic experiments.

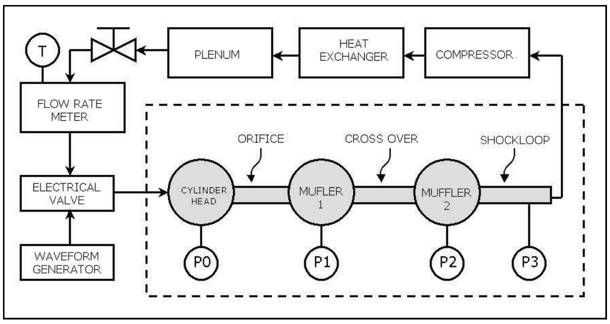

Figure 3: Schematic view of experimental test apparatus.

The technical specifications of the equipment are related below,

<u>Pressure sensors:</u>

1) ET-3DC-312-1400mbar – Differential pressure transducer destined to measure very low-pressure differences where high accuracy is needed. Range 0 to 1400mbar, infinitesimal resolution with a total error band of ±0.25% FSO, inclusive of all errors over a wide temperature range of -40ºC to +120ºC and bandwidth 0 to 2500Hz.

2) ETL-76M-190-300 – Small and amplified pressure transducers for absolute pressure. Range 0 to 300psia, Infinitesimal resolution with a total error band of ±0.25% FSO, inclusive of all errors over a wide temperature range of -40ºC to +120ºC and bandwidth 0 to 2500Hz.

<u>Flow Meter:</u>

Sensor model RHM03GNT, flow range 0,05 to 1 kg/min with temperature range of -20ºC to 120°C and maximum pressure 150bar. The flow meter also gives the fluid temperature by the analogical output 0-20mA

Electrical Valve:
MHE2-MS1H-3/2G Festo Solenoid valve with nominal flow rate 100 l/min, maximum frequency of 330Hz and operating pressure of 0.9 to 8bar
Waveform generator:
Agilent 33220A arbitrary waveform generator used to command the solenoid valve.
Data acquisition system:
Computer, data acquisition board model PCI6052E National instruments, SCXI-1000 chassis with SCXI-1125 conditioning module and terminal block SCXI-1313. Software for acquisition and management data developed in Labview™ language.

4.1 Test Procedure

Steady state experiments were performed without the electrical valve, which was taken out of the circuit and replaced by a regular tube, and the procedure was done according to the following description. The gas was driven through the circuit as the compressor was turned on, after a few minutes the stabilization was reached and the parameters (pressures, temperature and mass flow rate) were recorded. Other mass flow rate values were obtained adjusting the register, positioned after the plenum, shown in Figure 3. After a few minutes to the whole circuit stabilized at the new flow rate value and the parameters were recorded. This procedure was repeated for 43 values of mass flow rate within the range of approximately 0,4 to 5 g/s.

The dynamic test was done with the electrical controlled discharge valve installed in the test circuit and its working conditions like opening time, frequency and waveform were settled in the Waveform Generator. Valve and compressor were turned on and after stabilization the measured data were recorded during a period of 1s with an acquisition rate of 20kHz.

5. RESULTS AND DISCUSSION

The results obtained in steady state regime for pressure drop of discharge system components are presented in Figures 4 to 7. The experimental information is displayed in dots and the continuous line represents the fitted model from these measured results. The agreement is quite good showing that the model developed on theoretical and experimental basis described the phenomena very properly. It is clearly seen that shockloop is responsible for almost all pressure drop of complete discharge system, what can be observed comparing Figures 6 and 7. The orifice performed the smallest pressure drop level (Figure 4) and cross over shown the intermediate level (Figure 6), both together represent less than 10% of the whole discharge line. In spite of the good agreement of data and model it must be reminded that generality was reduced as the adjusted model is valid specifically to the system experimentally evaluated here, so if system configuration changes the procedure must be redone.

Steady state curves presented a different shape from the expected quadratic format, commonly observed for pressure drop. This apparent incoherence can be explained by a connection between density and mass flow rate forced by experimental apparatus and procedure. The register placed after the plenum and before flow rate meter (Figure 3) was used to adjust mass flow rate. Opening the register mass flow rate could be increased, and vice versa. Higher values of mass flow rate reached by register opening also elevated the pressures on system tested, increasing gas density. Rewriting Equations (1) and (2) combined with Equation (6) lead to Equations (7) and (8):

$$\dot{m} = \rho \overline{V} A_S \tag{6}$$

$$\Delta P = -\left(\frac{k}{2A_S^2}\right)\frac{\dot{m}^2}{\rho} \tag{7}$$

$$\Delta P = -\frac{f}{2A_S^2}\left(\frac{L}{D}\right)\frac{\dot{m}^2}{\rho} \tag{8}$$

Equations (7) and (8) show that keeping density constant the pressure drop becomes a quadratic function of mass flow, except by small changes of friction factor and friction loss coefficient. But as density increases with mass flow rate the quadratic trend is reduced and approximates to a straight line observed in the following figures.

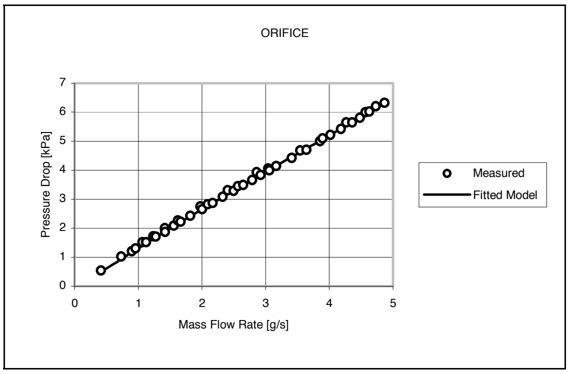

Figure 4: Orifice characterization on steady state.

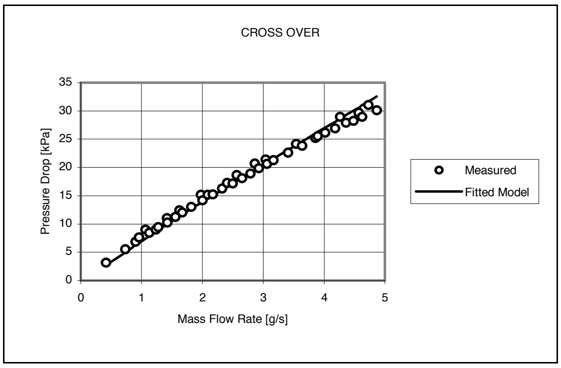

Figure 5: Cross Over characterization on steady state.

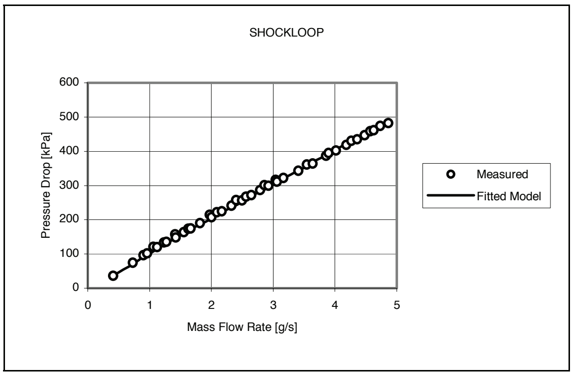

Figure 6: Shockloop characterization on steady state**.**

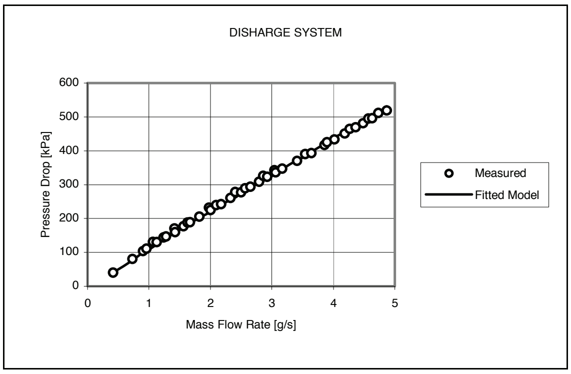

Figure 7: Discharge System characterization on steady state.

Transient test was developed under the following conditions:

Average mass flow rate = 0,58g/s
Gas temperature = 24,5°C
Ambient Temperature = 24°C
Electrical valve open time = 0,002s (10% of period)
Valve repeatable period = 0,02s (equivalent to 50Hz)
Waveform = square
Exit Pressure = 101,6kPa

The information obtained in transient regime can be seen on Figures 8 to 13 for pressures, mass flow rate and velocities. Theoretical mass flow rate through discharge valve was also plotted to reveal its effect concerning gradients and synchronism. Pressures plotted in Figures 8 to 11 are the difference between absolute values obtained for each component and constant exit pressure. The comparison of measured and simulated pressures in cylinder head, muffler 1 and muffler 2 are depicted in Figures 8, 9 and 10, respectively. Pressures in cylinder head (Figure 8) presented very good agreement during valve opening period, but after shut down some discrepancy was observed especially in the floating frequency of measured data, which seems to be higher than calculated. It is clear that discharge valve opening and closing movements assign strong gradients to cylinder head pressure. Experimental and calculated pressures in Muffler 1, observed in Figure 9, presented conformity and a smaller oscillatory range when compared with cylinder head. Muffler 2 presented the smallest oscillatory range in spite of the peaks in cylinder head, good agreement between experiment and theory was found. Although pressure fluctuations provided by discharge valve in cylinder head are very intense they were quite reduced along discharge line. Pressure pulsation is a source of noise according to Park *et al*. (2008), so attention should be paid concerning the application of this kind of valve.

Pressure pulsation might lead to pulsating flows in orifice. Observing Figure 11 it is possible to see some periods when muffler 1 pressure overcomes cylinder head promoting a flow velocity reduction in the orifice. This reduction is strong enough to provide a downward flow indicated by negative values of mass flow rate observed in Figure 12. The phenomena can also be seen on Figure 13 in terms of velocities. In spite of no experimental confirmation of results from Figures 11 to 13, they seem to be an evidence of noise problems on the application of electrically commanded valves.

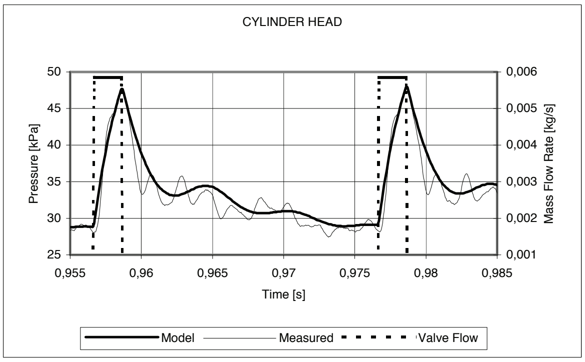

Figure 8: Pressure inside cylinder Head.

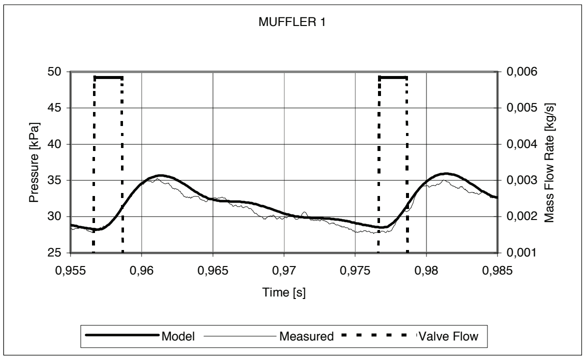

Figure 9: Pressure inside Muffler 1

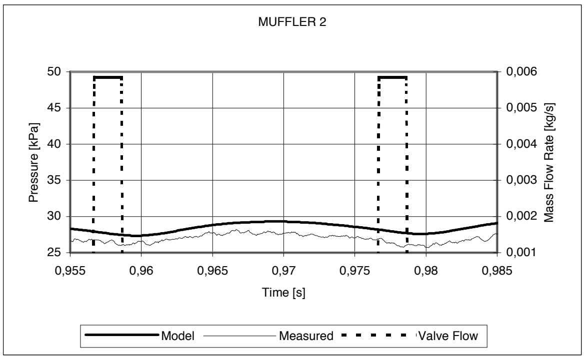

Figure 10: Pressure inside Muffler 2

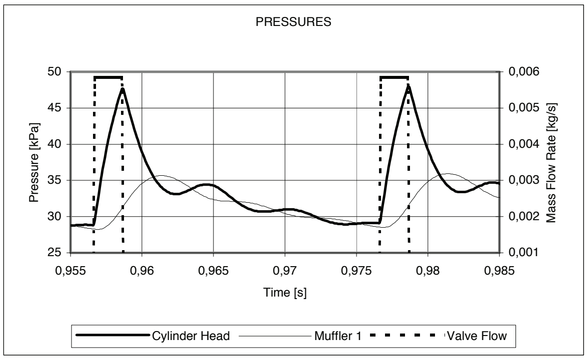

Figure 11: Orifice and Muffler 1 simulated pressures

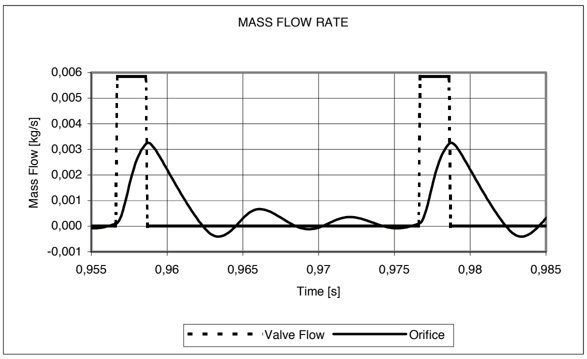

Figure 12: Orifice and valve mass flow rate

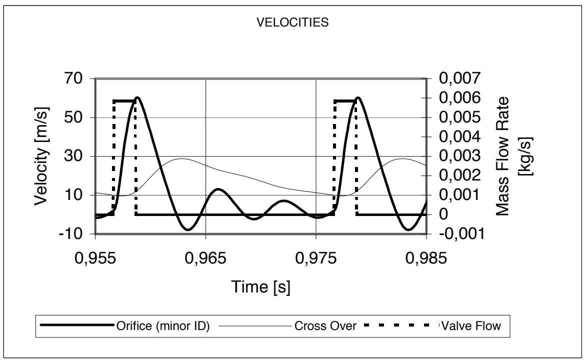

Figure 13: Orifice, cross over and shockloop velocities.

6. CONCLUSIONS

The gas flow behavior along the discharge system of a regular compressor provided with an electrically commanded discharge valve was evaluated. A 1D dynamic model was developed to analyze the phenomena and the results were compared with measured data. The discharge system was separated in components, which were experimentally characterized in steady state regime and these results were applied to the transient model. Good agreement of theoretical and experimental was observed on pressures and their gradients along time except a small discrepancy was found in cylinder head pressure after valve shutdown in terms of floating frequency. Under a qualitative point of view the agreement between model and measurement was very interesting as the model applied was relatively simple. Potential problems of noise due to pressure pulsation were found and deserve special attention.

NOMENCLATURE

ΔP	pressure drop	(Pa)
ρ	density	(kg/m^3)
V	velocity	(m/s)
L	tube length	(m)
D	internal tube diameter	(m)
F	force	(N)
k	friction loss coefficient	
f	friction factor	
A	area	(m^2)
P	pressure	(Pa)
τ	shear stress	(Pa)
x	length along flow direction	(m)
$\mathcal{V}$	volume	(m^3)

Subscripts

CS	control surface
W	tube or orifice wall
S	cross section
1	upstream
2	downstream
CV	control volume

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