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Experimental Investigation of Damping Coefficient for Compressor Reed Valves

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ABSTRACT

Reed valve dynamics has a major role on the performance and reliability of compressors and has been the subject of many studies over the years. In terms of mathematical modeling, most of these studies describe valve dynamics through a mass-spring-damper system in which a damping coefficient has to be empirically adjusted. This is a consequence of little knowledge about the effect of different parameters on valve damping. For instance, in spite of great effort to understand viscous effects on the pressure distribution acting on the valve surface, very few works have addressed such effects on valve damping, especially for valve geometries found in compressors. The present study aims to experimentally quantify valve damping under controlled conditions considering the effects of clamping geometry, clamping force, gasket thickness and the presence of fluid around the valve. Finally, experimental data of damping coefficients are used to predict valve dynamics and compressor efficiency. We found that the damping coefficients of typical valves adopted in small reciprocating compressors have negligible effect on compressor efficiency and valve bending stress, but can change by up to 17% the valve impact velocity against the seat.

1. INTRODUCTION

The dynamics of reed valves can be described in a simplified way with a single degree-of-freedom model:

$$m_{eq}\ddot{x} + c\dot{x} + kx = F_{valv} - F_{ot} \tag{1}$$

where m_{eq}, c and k are the reed equivalent mass, damping coefficient and stiffness, respectively. On the other hand, F_{valv} is the force induced by the pressure load on the reed surface and F_{ot} can represent any other force, such as reed pre-tension or stiction force due to the presence of a lubricating oil film between the reed and the valve seat (Khalifa and Liu, 1998). Finally, x, $\dot{x}$ and $\ddot{x}$ are the instantaneous reed lift, velocity and acceleration, respectively.

Valve dynamics is affected by dissipative (damping) forces, leading to smaller valve lifts and valve velocities. Smaller valve lifts increase viscous loss in the suction and discharge processes, reducing the compressor efficiency. On the other hand, small lifts reduce both the valve bending stress and the valve impact velocity, increasing valve reliability. Valve damping can be attributed to three different dissipative sources: i) material damping due to complex interactions in the crystalline structure and internal imperfections; ii) clamping damping due to the friction force of micro-slips between the surfaces that are used to fasten the valve; iii) viscous damping due to the presence of fluid surrounding the valve during its displacement. Prater and Hnat (2003) carried out experiments to characterize reed valve damping in rotary compressors. Material, clamping and viscous damping were obtained by measuring the valve free vibration and its oscillation decay with an optical sensor to avoid any interference in the valve dynamics.

The present paper reports the results of an experiment investigation similar to that of Prater and Hnat (2003) with the aim of determining the damping coefficient of a reed type valve of a small reciprocating compressor. The influence of clamping geometry, fastening force and density of the fluid surrounding the valve are investigated. Additionally, simulations are carried out to estimate the effect of valve damping on the compressor efficiency and valve impact velocity.

2. EXPERIMENTAL SETUP

The method applied to measure the valve damping follows the procedure described by Granick and Stern (1965), but we consider actual valve geometries instead of a simplified beam geometry. The experiment takes place inside a small vessel (Figure 1) with absolute pressure below 2 mbar and temperature equal to 25 °C. At such condition the air density was approximately 0.002 kg/m³, hence minimizing viscous damping.

In order to measure the valve damping the reed valve was allowed to oscillate freely at its natural frequency after being excited by an automatic device mounted inside the vessel. Valve oscillation was measured using two coils of a fast linear displacement transducer (FLDT), as shown in Figure 2. The use of more sophisticated devices, such as a laser displacement meter, was not possible due to space limitation inside the vessel. The FLDT coils were design to accurately measure small displacements and to compensate errors caused by temperature change in the coils. The test bench also had a signal processor and a data acquisition system with sampling rate of 100 kHz.

The inductive coils to measure valve displacement were calibrated for each valve assembly. The calibration procedure consisted in moving the valve from its rest position to its maximum displacement, with the displacement being simultaneously measured by a high-accuracy laser sensor and the coils connected to the signal sensor. A calibration curve was then made available via a transfer function between the voltage signal from the two coils and the valve displacement measured with the laser system. Figure 3 presents a diagram of the calibration apparatus.

The damping coefficient was measured for a typical suction valve of a small reciprocating compressor used in household refrigeration systems. Measurements were repeated four times for each configuration of interest, including valve reassembly, following a procedure of six steps:

i. Assembling the reed valve in the in the measuring device formed by the two coils (Figure2);
ii. Calibrating the valve displacement transducer (FLDT coils);
iii. Mounting the measuring device inside the vessel (Figure1);
iv. Establishing vacuum in the vessel with a vacuum pump;
v. Moving the reed valve to its maximum displacement (x_0) with a mechanical device (Figure2);
vi. Releasing the reed valve and measuring its oscillating motion.

Figure 1: Vessel with rarefied atmosphere.

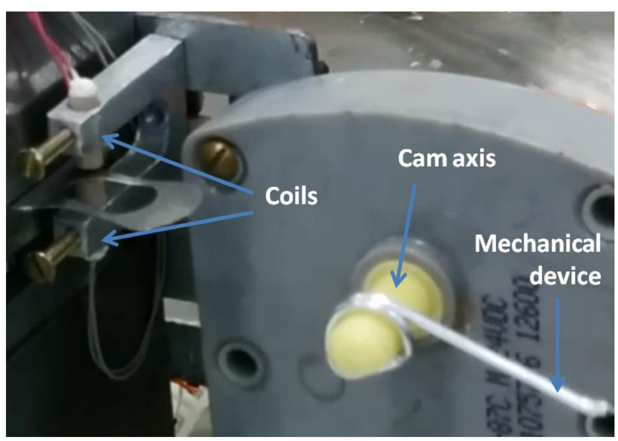

Figure 2: Test section.

The valve damping coefficient can be expressed as:

$$c = 2\gamma m_{eq} \tag{2}$$

The decay parameter γ in Equation 2 is estimated by adjusting its value in Equation (3) so as to obtain the actual decay of the valve oscillating motion observed experimentally (green line), as shown in Figure 4.

$$x_{valv} = x_0 e^{(-\gamma t)} \tag{3}$$

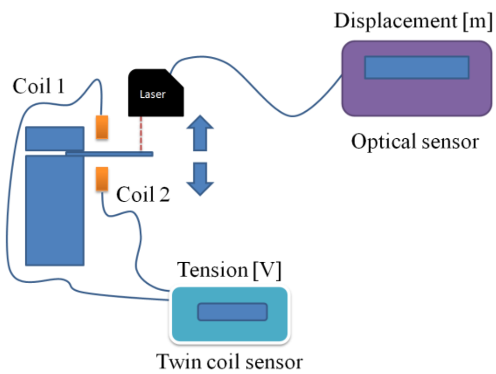

Figure 3: Calibration apparatus.

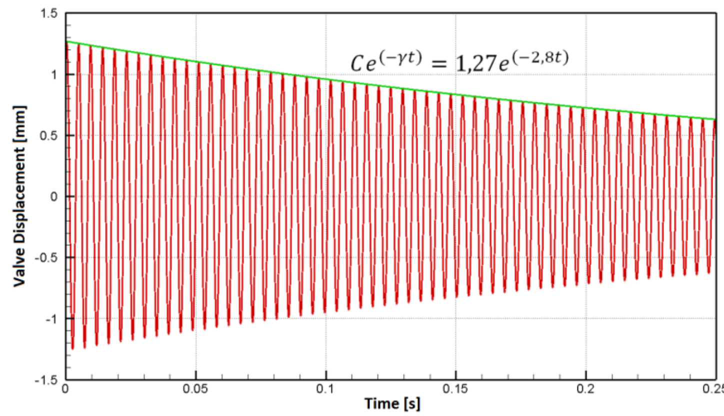

Figure 4: Determination of γ coefficient.

3. EXPERIMENTAL RESULTS

Valve damping was initially analyzed in relation to valve clamping with and without gasket. When the valve fastening system adopts a gasket the maximum torque is applied to the bolts, which represents the situation in the early life of the compressor. In the absence of a gasket, the valve is fastened in place against the other metallic surface by using the same torque. This configuration is expected to result in smaller dissipation force at the clamp region. Figure 5 shows the average values and dispersions of measurements for the damping ratio with and without gasket. As can be noted, the dispersion of both results are similar and small enough to allow the conclusion that the damping ratio for the clamping with gasket is approximately 60% greater than that without gasket. In fact, the damping ratio for the clamping without gasket is quite small and can be attributed solely to the material damping.

When the compressor is in the field, i.e., operating in a refrigeration system, the clamping torque is much lower than the original torque applied in the compressor assembly due to torque relaxation in extreme temperatures. Hence, we decided to analyze also the damping coefficient as a function of two different torques (T1 = 9 Nm; T2 = 4.5 Nm) and two gasket thicknesses (t1 = 0.25 mm; t2 = 0.48 mm), resulting in four test configurations (T1t1, T2t1, T1t2, T2t2). Figure 6 shows a clear increase in the valve damping when the torque is decreased, probably because the reduction of the fastening force makes the gasket less rigid. Additionally, the valve damping seems to increase with the gasket thickness for the same reason.

Nashif *et al.* (1985) analyzed the effect of the beam natural frequency f_n on the valve damping. We explored this aspect by comparing the dynamics of a second valve with $f_n = 280$ Hz with the dynamics of the original valve whose natural frequency was $f_n = 200$ Hz. Both natural frequencies are typical of valves used in compressors designed for domestic refrigeration. Figure 7 presents prediction and measurement for the decay of the valve oscillating motion with $f_n = 280$ Hz. It should be mentioned that the material damping specified in the simulation model was equal to the value measured for the valve with $f_n = 200$ Hz. Both results are quite similar and show that the natural frequency is of secondary importance in the valve material damping.

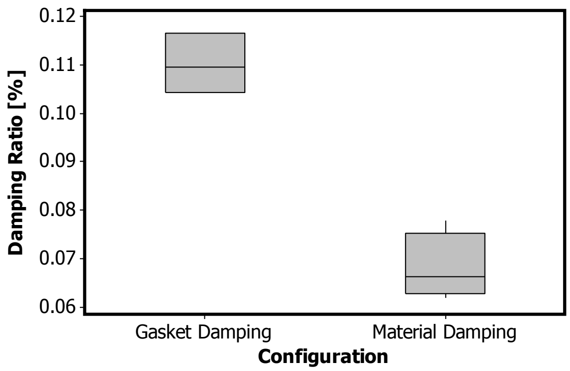

Figure 5: Damping ratios for valves with and without gasket.

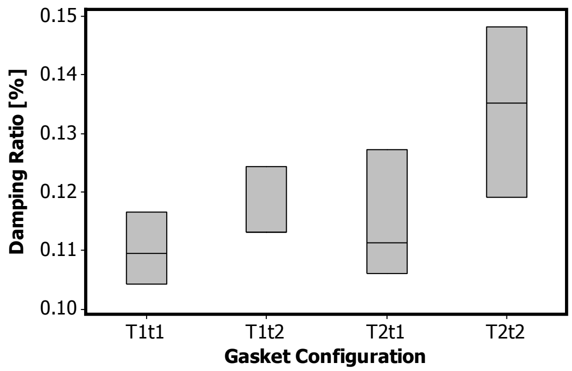

Figure 6: Damping ratios for different torques and gasket thicknesses.

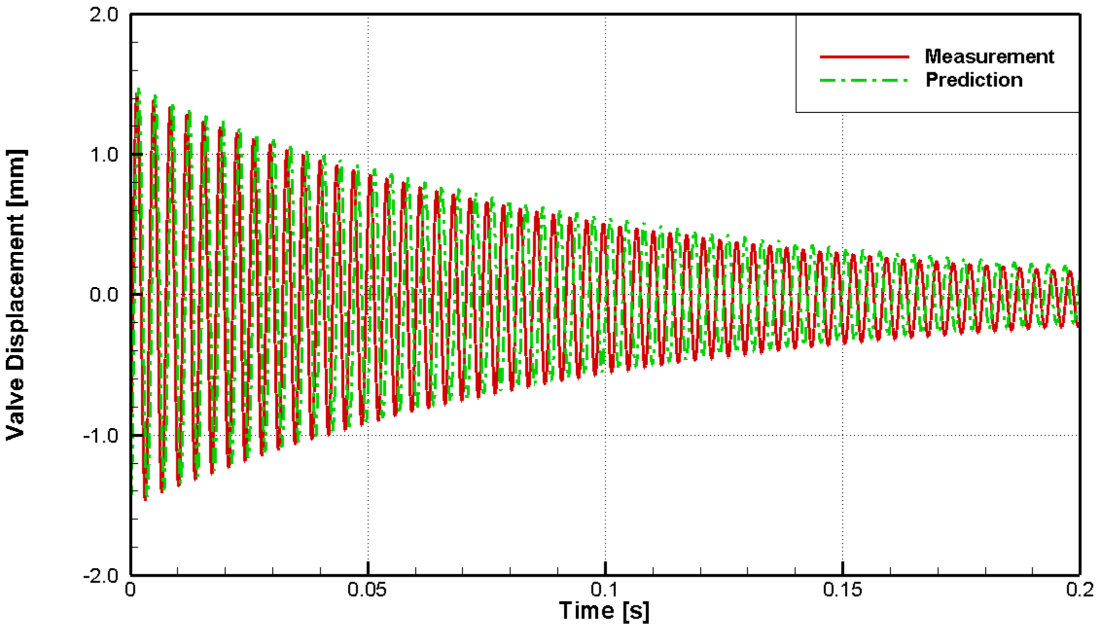

Figure 7: Comparison between numerical and experimental results for valve displacement (without gasket).

4. NUMERICAL ANALYSIS

The experimental setup only allows tests with air at pressures lower than or equal to the atmospheric pressure, hence not being able to cover the entire range of pressure levels experienced by the compressor during its compression cycle. In order to investigate the viscous damping for different gases and pressures we decided to adopt two auxiliary simulation models, one based on the finite volume method (FVM) for the fluid flow through the valve and the other following the finite element method (FEM) to solve the valve displacement.

As shown in Figure 8, the computational domain for the fluid flow is formed by a large volume of fluid around the valve to minimize the effect of the adopted boundary condition (total non-reflective pressure) on the solution. Figure 9 shows the computational grid of the solution domain for the valve dynamics via FEM. In this fluid-structure interaction analysis, the valve dynamics and the time dependent flow field are coupled and have to be solved simultaneously. For instance, the fluid flow pressure field causes the reed to deform, which in turn affects the fluid flow solution. Further details of this fluid-structure model can be obtained in Shiomi *et al* (2009).

The simulation procedure consists in applying an initial impact on the valve and then registering the decay of its oscillating motion similarly as carried out in the experimental setup. We tested different fluid densities in the simulations, since Prater and Hnat (2003) reported this parameter strongly affects the viscous damping. The results allowed us to obtain an expression for the damping ratio, $\xi\ (= c/c_c)$, as a function of the fluid density (Equation 4). In accordance to the operating conditions of refrigeration compressors, the density was varied from 1kg/m³ to 70kg/m³, leading to changes up to 18% in the viscous damping ratio.

$$\xi = 4.785 \times 10^{-6}\rho^2 + 2.256 \times 10^{-3}\rho - 9.377 \times 10^{-4} \tag{4}$$

For validation purpose, the damping ratio, ξ, given by Equation 4, was adopted in Equation 1 to predict the valve oscillation decay when left to oscillate freely on air at atmospheric pressure. The material damping previously measured was adopted in the model. As shown in Figure 10, close agreement was found between measurement and prediction, validating Equation 4.

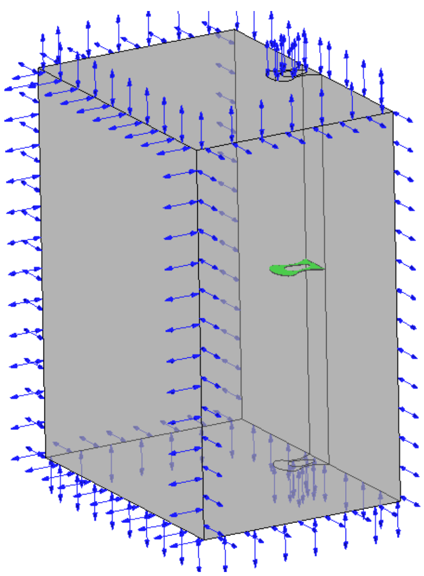

Figure 8: Fluid flow solution domain.

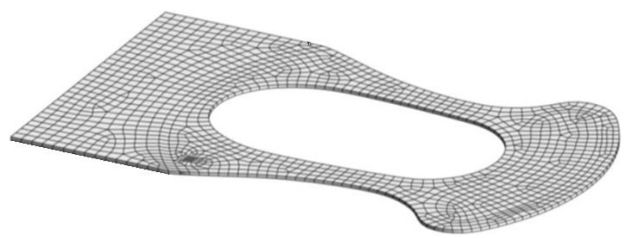

Figure 9: Computational grid of the structural domain.

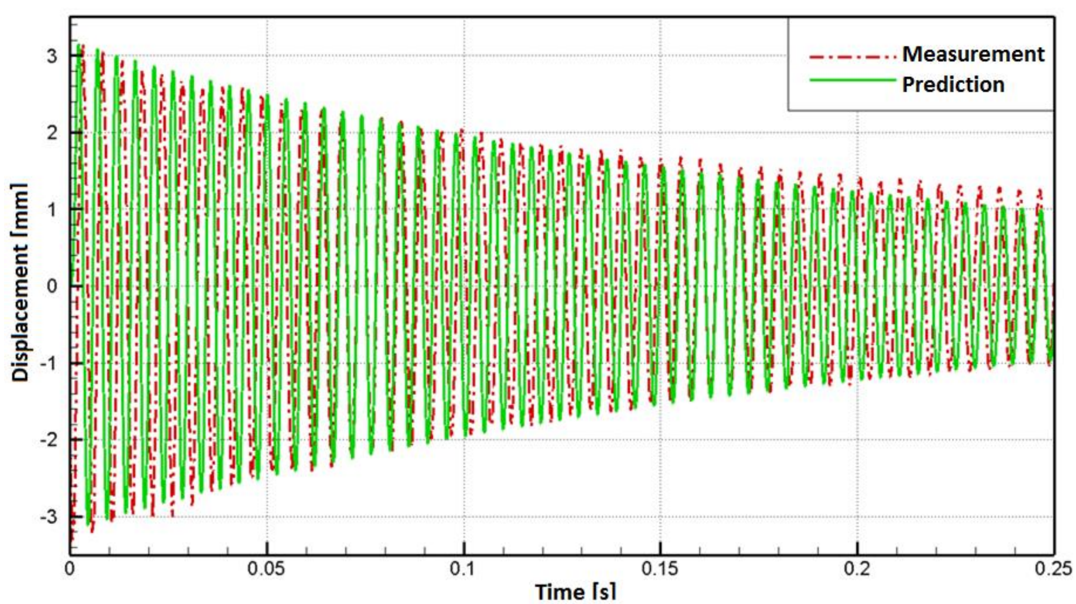

Figure 10: Comparison between numerical and experimental results for valve displacement considering air as the working fluid.

5. EFFECT OF DAMPING COEFFICIENT ON COMPRESSOR PERFORMANCE

Figure 11 shows the contributions of material, gasket, fluid and the total damping ratio, ξ, for the suction valve of a compressor operating with R-600a. As can be noted, the total damping coefficient is not high, being just 0.4% of the critical damping, c_c. The contribution of the fluid surrounding the valve on the damping is similar to the other two contributions (material and gasket). The total damping associated with the discharge valve of the same compressor is 2.6% of the critical damping (Figure 12) and it is essentially due the fluid. Naturally, the damping coefficient is also a function of the valve geometry and Equation 4 should be used with caution in the simulation of considerably distinct valve designs.

5.1. Compressor Efficiency

The effect of the damping coefficient on the efficiency of a small reciprocating compressor was analyzed taking into account the dynamics of suction and discharge valves. A common value attributed to the damping ratio of suction valves is $\xi \cong 0.1$. Figure 13 shows the increase of energy loss in the suction process due to valve damping in comparison to a baseline situation without damping. As can be seen, a damping ratio of 0.1 would increase the suction loss by 5.8%, whereas the actual damping ratio of 0.004 (0.4%) is responsible by an increase of just 0.2% in the suction loss. Therefore, the assumption of no damping for the suction valve would not bring any significant error in predictions of the compressor efficiency.

As seen before, the damping ratio of the discharge valve is greater than that of the suction valve. Moreover, the discharge process takes place in a much shorter period than the suction process. Figure 14 shows that the valve damping leads to a reduction of just 2.5% in the discharge loss. Therefore, the damping coefficient could also be neglected in the modeling of the discharge valve dynamics. It is a common practice to model the dynamics of the discharge valve as a critically damped system ($\xi = 1$). The results in Figure 14 show that this assumption leads to an error of almost 26% in the discharge loss.

Overall, high damping coefficients for both valves ($\xi = 0.1$ and 1) can lead to reductions up to 0.6% in the COP of typical household refrigeration systems. On the other hand, the damping coefficients obtained from our measurements would decrease COP by less than 0.1%, which is again evidence that valve damping is not important when the compressor efficiency is the main interest of the analysis.

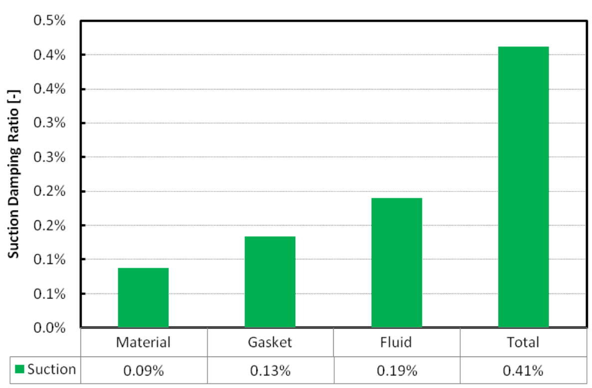

Figure 11: Damping ratio for the suction valve of a compressor operating with R-600a.

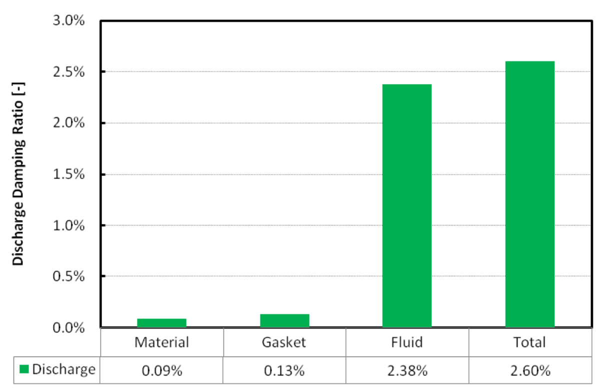

Figure 12: Damping ratio for the discharge valve of a compressor operating with R-600a.

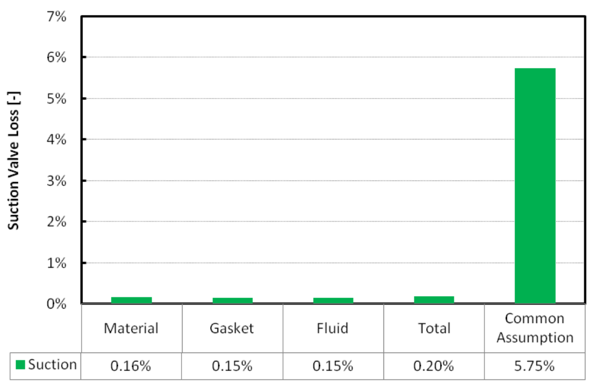

Figure 13: Suction loss due to suction valve damping.

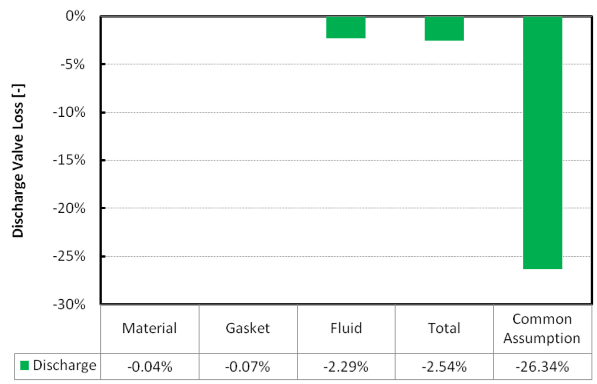

Figure 14: Discharge loss change due to discharge valve damping.

5.2. Valve Impact Velocity

In this section we explore the effect of the damping coefficient on the valve maximum opening, which defines the bending stress and, to some extent, the valve impact velocity. The analysis considers the compressor under an extreme operating condition that may occur during its duty cycles. Figure 15 shows that the effect of the damping coefficient on the suction valve opening is small, but it can also be observed that the common assumption of $\xi \cong 0.1$ leads to considerable error in the result. Figure 16 considers the influence of the damping on the suction valve impact velocity. As can be seen, despite its negligible effect on the valve opening, the actual damping affects the valve impact velocity by 17%, which is quite significant in terms of compressor reliability.

The discharge valve usually adopts a stopper device to limit its maximum opening. In the present analysis, the damping force increases the discharge valve impact velocity by 1.1%. On the other hand, the assumption of a critically damped system ($\xi = 1$) for the valve dynamics would overestimate the valve impact velocity by almost 28%. This occurred in our simulation because the high damping coefficient delayed the valve closing. As a consequence, the piston had already started its motion towards the bottom dead center position just before the valve was closed, suddenly increasing the pressure load on the valve and its final velocity.

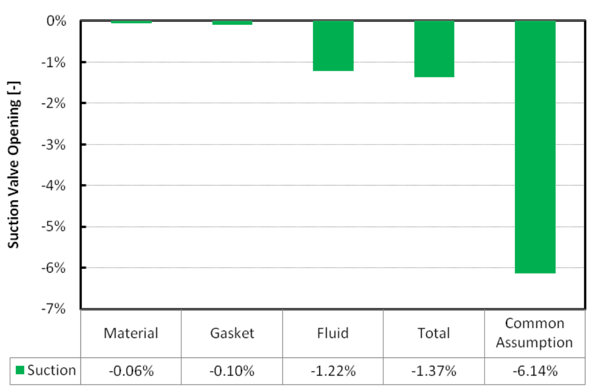

Figure 15: Effect of the damping coefficient on the suction valve opening.

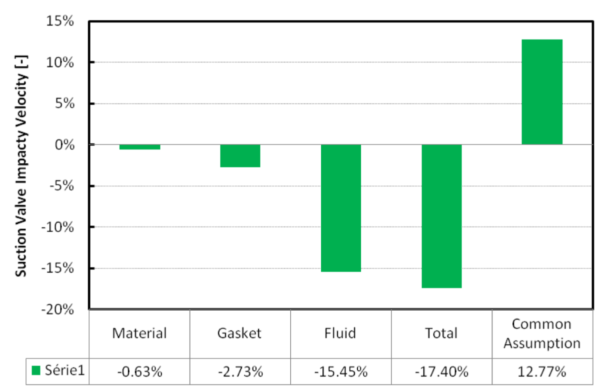

Figure 16: Effect of the damping coefficient on the suction valve impact velocity.

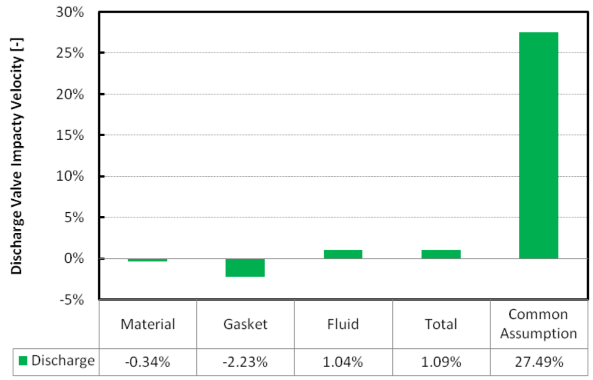

Figure 17: Effect of the damping coefficient on the discharge valve impact velocity.

6. CONCLUSIONS

The present investigation experimentally quantified valve damping under controlled conditions considering the effects of clamping geometry, clamping force, gasket thickness and the presence of fluid around the valve. The results showed that the damping due to the material and clamping is much less important than that associated with the fluid. It was observed that valve damping can be neglected without incurring significant errors in predictions of compressor efficiency. We found that the damping coefficients of valves of small reciprocating compressors have negligible effect on the valve bending stress. However, the damping force can change the valve impact velocity by 17%, which is important in the assessment of compressor reliability.

NOMENCLATURE

x	valve displacement	(m)	**Subscript**	
$\dot{x}$	valve velocity	(m/s)	*Valv*	valve
$\ddot{x}$	valve acceleration	(m/s²)	0	initial condition
F	force over the valve	(N)	ot	other
m	valve mass	(kg)	eq	equivalent
k	valve stiffness	(N/m)	*n*	natural
f	valve frequency	(Hz)		
γ	half damping and mass ratio	(1/s)		
t	time	(s)		
c	damping coefficient	(kg/s)		
ξ	damping ratio	(-)		
ρ	specific mass	(kg/m³)		

REFERENCES

Granick, N., Stern, J. (1965). Material Damping of Aluminum by a Resonant-Dwell Technique, NASA Technical Note, D-2893, Washington, D.C., USA.

Khalifa, H. E., Liu, X. (1998). Analysis of Stiction Effect on the Dynamics of Compressor Suction Valve, In: *Proc. Int. Compressor Eng. Conference at Purdue, West Lafayette, USA* (87-92).

Nashif, A. D., Jones, I. G., Henderson, J. P. (1985). Vibration Damping, Wiley, New York.

Prater Jr., G., Hnat, W.P. (2003), Optical Measurement of Discharge Valve Modal Parameters for a Rolling Piston Refrigeration Compressor, Measurement, Vol. 33, p. 75-84.

Shiomi, F. H., Takemori, C. K., & Deschamps, C. J. (2009). Comparative Analysis of Different Modeling Approaches for Valve Dynamics of Small Reciprocating Compressors. In: *Proc. 2009 IIR/IIF Int. Congress on Compressors, Papiernicka, Slovakia* (8 p.).