Prediction of Reed Valve Velocity Impact

Igor L L P SILVA[1], Artur D FAVERA[1], Claudio J SANTOS[2], Cesar J DESCHAMPS[1]

[1]POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics
Federal University of Santa Catarina
Florianopolis, SC, Brazil

[2]NIDEC Global Appliance, R&D
Joinville, SC, Brazil

* Corresponding Author: deschamps@polo.ufsc.br

ABSTRACT

The reliability of compressor valves is a significant challenge in extending the operational speed range of compressors due to the risk of impact fatigue failure. Valve raw material suppliers have adopted the velocity of valve impact against the seat as a criterion to assess the reliability of impact fatigue. This article analyzes the capabilities of a low-cost numerical model developed with GT-SUITE in predicting the suction valve impact velocity, while analyzing the influence of each model parameter and operational condition on the prediction of velocity impact. The numerical investigation was validated through measurements of valve velocity carried out with a laser doppler vibrometer. The study shows that valve contact modeling significantly affects the prediction of valve dynamics due to the rebound effect following valve impact. Furthermore, oil stiction has substantial influence on the prediction of valve velocity impact under high-speed operations. Overall, the model was able to predict the impact velocity in good agreement with the measurements. Compressor speed was found to significantly increase the valve impact velocity up to a certain point, after which it shows little variation between the two highest speeds tested. On the other hand, the pressure ratio showed an insignificant influence on valve impact velocity. The findings of this study improve the valve dynamics predictions, especially under high-speed operations.

1. INTRODUCTION

Reciprocating compressors have extended their operational speed range to allow the same cooling capacity with smaller displacements. The valve system represents a significant challenge in ensuring performance and reliability under high-speed operations. Valve reliability is associated with bending fatigue, resulting from the valve's oscillating motion, and impact fatigue, due to the valve's repetitive impacts against the seat. Impact fatigue is paticularly challenging for valve reliability analysis as it involves a more complex valve failure phenomenon.

Given the importance of impact fatigue, numerical analyses have been conducted to examine the stress during valve impact. Lajús Junior et al. (2013) investigated the main parameters influencing valve failure using an explicit impact numerical method, revealing that maximum impact stress is significantly affected by minor geometric variations. Tofique (2021) analyzed the stress impact using a numerical-experimental study using air pulses to control valve opening. The fluid-structure model developed by the author was validated through experimental data of valve motion, with the predicted stress fields correlating well with experimenatly observed failures, particularly the maximum stress at the valve tip.

Despite the need to understand stress fields at valve impact, measuring and predicting these parameters is very difficult due to the rapid and complex nature of loading generated during impacts. Therefore, valve raw material suppliers have adopted the valve impact velocity against the seat as a primary criterion to ensure impact fatigue reliability. Following this approach, this article explores the capabilities of a low-cost model to predict suction valve impact velocity, supported by measurements of valve velocity.

2. NUMERICAL MODEL

2.1 Valve Dynamics

A low-cost model was implemented using the multiphysics software GT-SUITE to predict the impact velocity of the suction valve of a reciprocating compressor. The valve dynamics was modeled using a one-degree-of-freedom (1-DOF) mass-spring-damper system, with the reed motion parallel to the seat, as depected in Figure 1. Accordingly, the valve dynamics was resolved via the following equation:

$$m_{eq}\ddot{s}_v + c_{eq}\dot{s}_v + k_{eq}s_v = F_p + F_{sc} + F_c, \tag{1}$$

where $\ddot{s}_v$, $\dot{s}_v$ are s_v represent the valve acceleration, valve velocity and valve lift, respectively. In the equivalent system, m_{eq}, c_{eq} and k_{eq} correspond to valve's equivalent mass, equivalent damping and valve stiffness, respectively. The equivalent mass of the valve accounts for its inertia during opening and closing motions. The equivalent damping is associated with the energy dissipation attributed to internal material damping and viscous damping due to the motion of the valve surrounded by fluid (Lohn et al., 2016).

The external loads F_p, F_{sc} and F_c on the right side of Eq.(1) are associated with the pressure load, oil stiction force and contact force, respectively. The force due to pressure load is estimated using the concept of effective force area A_{ef}, as follows:

$$F_p = A_{ef}(s_v) \cdot \Delta P, \tag{2}$$

where ΔP is the pressure difference between upstream and downstream of the flow through the valve. The oil sticion force due to oil film dilating between reed and seat can be accounted via the model of Khalifa and Liu (1998):

$$F_{st} = \frac{3\pi\mu R_o^2}{2(s_v + h_0)^3}\dot{s}_v\left(1 - X^4 + \frac{1 - 2X^2 + X^4}{\ln X}\right), \tag{3}$$

where μ corresponds to the oil viscosity, h_0 the initial oil film thickness when the valve is closed and X corresponds to the ratio bewteen valve radius R_v and orifice radius R_o. Finally, the contact force was estimated with the following equivalent spring-damper model:

$$F_c = \begin{cases} k_c\delta + c_c\dot{s}_{v,c}, & \delta > 0, \\ 0, & \delta \leq 0 \end{cases} \tag{3}$$

where k_c and c_c corresponds to the contact stiffness and contact damping, respectively. The term δ is the penetration between the valve reed and the seat, and $\dot{s}_{v,c}$ corresponds to the valve velocity during contact penetration.

The valve impact velocity is calculated at the last timestep prior to the contact between valve and seat. Hence, the timestep is restricted during the valve closing motion near the seat to better predict the impact instant. Usually, the suction valve impacts against the seat multiple times within the same suction process. In this study, the impact velocity will be considered as the maximum impact velocity throughout the entire suction process. The simulaton was assumed to reach fully periodic regime when the valve impact velocity varied less than 2% of its maximum value for five consecutive compression cycles.

2.2 Compression Cycle

The compression cycle model was simulated with the GT-SUITE software. Figure 2 illustrates volumes and tubes adopted to represent the compression chamber and discharge muffler. For the compression chamber, a lumped model was adopted, solving the conservation equations for mass and energy in their integral formulation. The mass flow through the valves was estimated using effective flow areas with reference to isentropic flow in a convergent nozzle. Leakage through the piston-cylinder gap and in valve system were not taken into account. The compression chamber volume was analytically calculated as a function of the crank angle and mechanism parameters.

The gas density in the compression chamber was evaluated from mass balance, while the gas temperature was calculated from energy balance. The resulting pressure in the compression chamber was then obtained from the density and temperature using the Refprop database (LEMMON, 2018). For the flow field in the discharge filter, a one-dimensional formulation for mass, momentum, and energy conservation equations was adopted (Deschamps et al., 2002). The simulation model was developed to represent the simplified geometry of suction system in the experimental setup, where the suction chamber was open to the ambient environment.

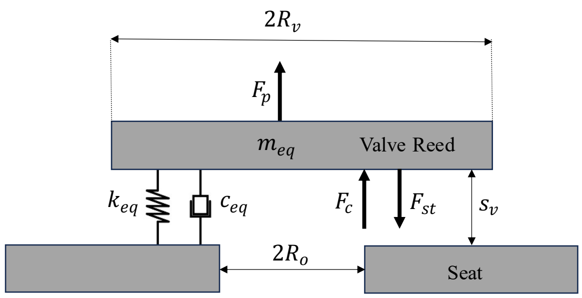


Figure 1: Valve 1-DOF equivalent system**.**

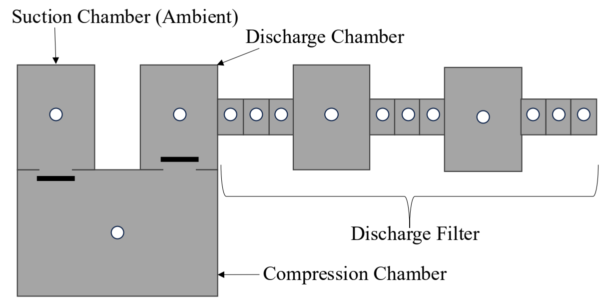


Figure 2: Flow domain discretization.

3. EXPERIMENTAL STUDY

The bench structure consists of a metallic box with an acrylic window on the front and slots on the back to allow optical access for valve velocity measurements. For these measurements, it was necessary to open the compressor housing and remove the compressor's mechanical assembly. Since the discharge valve controls the discharge pressure, the cylinder head forming the discharge chamber was retained, which made optical access to this valve difficult. Conversely, the suction muffler was removed, exposing the suction valves to the ambient environment. Consequently, velocity measurements were only performed on the suction valves. Inside the housing, the mechanical assembly is supported by 4 springs designed to attenuate the vibration levels transmitted from the compressor to the housing during operation. However, since the compressor must remain fixed and stable on the bench for accurate valve motion measurement, spring supports are not suitable. Thus, the springs were replaced by fixed supports that prevent movement of the mechanical assembly along the valve motion axis.

The bench was designed for the compressor to operate in an open system, unlike the closed system typical of a refrigerator. The working fluid is air at atmospheric pressure with an ambient temperature of 22°C at suction. To establish a pressure difference between the suction and discharge, the discharge line was pressurized using regulator valves, which were monitored by a pressure sensor, as shown in Figure 3.

For measuring valve velocity, a PSV-500 scanning vibrometer eqiped with a Laser Doppler Vibrometer (LDV) was employed. The effectiveness of this method has been valiated by Fenkl et al. (2016), Collings and Lenz (2010), Nagy et al. (2008), and Buligan et al. (2002). The equipment has a response frequency of 125 kHz and can measure velocities up to a full scale of 10 m/s. The laser head was mounted on a tripod to ensure stability, and it was pointed to the rear of the bench, where slots were positioned to enable the laser beam to reach the valve surface.

Measurements were conducted on the suction valve of a compressor tested at four operating speeds. The minimum speed served as the reference, with the subsequent speeds being medium speed (1.4 times the minimum), high speed (2.1 times the minimum) and maximum speed (2.9 times the minimum). The objective was to assess the effect of operating speed on the valve's impact velocity. Additionally, at the medium speed, the tests explored the influence of varying discharge pressures (Pdes = 2, 3, and 4 bar). The pressure range tested on the bench was limited to 4 bar to prevent excessively high temperature at the end of the compression process of air, which is significantly higher than those of refrigerants used in small capacity refrigeration systems. After the completation of measurements for each test condition, the motion speed of the compressor's mechanical assembly was also measured to verify its potential influence on the valve velocity measurement.

The velocity data collected required processing to calculate the impact velocity. To this end, a Python algorithm was developed. This algorithm is partulalry useful for analysing valve impacts, when decelerations are significantly higher than the accelerations during the rest of the valve motion, resulting in pronounced peaks in velocity derivatives at the moment of impact, as can be seen in Figure 4. The algorithm calculates the derivatives of the measured velocities at the highest closing velocities of the valve. It scan the values in this region, and when the derivative deviates beyound a range defined by the standard deviations of these derivatives, it indicates a sharp deceleration indicative of an impact, thus determing the impact velocity.

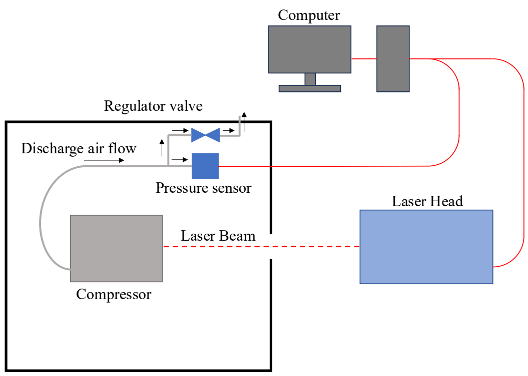


Figure 3: Valve velocity measurement**.**

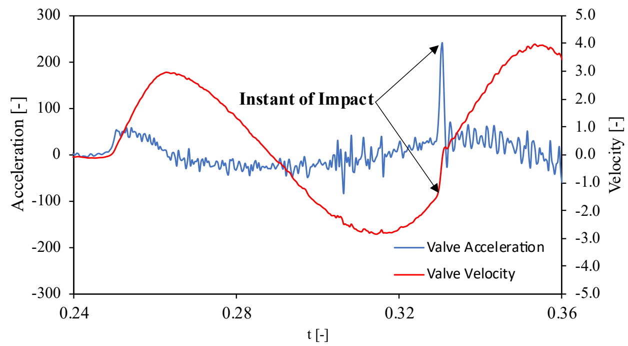


Figure 4: Valve acceleration at impact.

4. RESULTS

The primary objective of this paper is to analyze the capabilities of a low-cost numerical model for predicting the impact velocity of a suction valve. This analysis includes studying the influence of each model parameter and operating conditions on the prediction of impact velocity. To identify opportunities for enhancing the valve dynamic model, an experimental investigation was also carried out. The data for valve postion, velocity, and impact velocity, both numerical and experimental, will be presented herein in dimensionless form, normalized to the maximum valve velocity and displacement measured in the experimental setup. Additionally, the time scale will also be presented in dimensionless form, normalized to the compressor's operating frequency at the minimum speed tested.

4.1 Contact Force

The contact force is an essential parameter for predicting a physically consistent valve dynamics. Figure 5 shows measurements of suction valve position and velocity curves at the maximum tested speed (2.9) and 2 bar pressure in the discharge line. After each valve impact against the seat, the valve velocity undergoes an abrupt change in magnitude and direction, which is depicted by the gray bars. This steep change in valve velocity following impact results from extreme acceleration, directly modeled by the contact force. Therefore, the contact parameters were adjusted for simulate the valve's rebound after the impact.

Numerical results that accout for valve rebound after impact are more accurate, as can be observed in Figure 6. Without considering the valve rebound, the model shows the valve velocity dropping to zero after impact, with the valve subsequently beginning to reopen due to the pressure load. In this scenario, the valve's kinetic energy is completely dissipated following the impact. However, when the numerical model includes rebound, only a portion of the kinetic energy is dissipated at impact, allowing the valve to resume its opening motion immediately with a non-zero velocity.

When the rebound effect is not accounted into the model, the valve tends to delay its motion after each valve impact, as shown in Figure 6 (a). Furthermore, the final valve oscillations are not predicted by the model that does not include the rebound effect. Figure 6 (c) shows the pressure in the compressor chamber, with the dashed line representing the suction ambient pressure. The model that omits rebound effect fail to predict valve opening when the pressure load tends to close the valve. This motion is accurately characterized only with a physically consistent contact model, as can be seen in Figure 6 (b).

An incorrect contact modeling significantly influences the prediction of impact velocity during valve motions, particularly when there are multiple impacts against the seat during the suction process. If the contact is not accurately modeled, the pressure in the compression chamber decreases further due to a delay in valve opening, leading to an increase in the pressure load F_p on the valve. This results in a larger subsequent valve opening, causing the valve to close more quickly under the influence of the spring force, as illustrated in Figure 6 (b). Table 1 and Table 2 present the predicted dimensionless valve impact velocities, with and without the rebound effect. Notably, the rebound effect tend to reduce de valve impact velocity predictions in all cases analyzed.

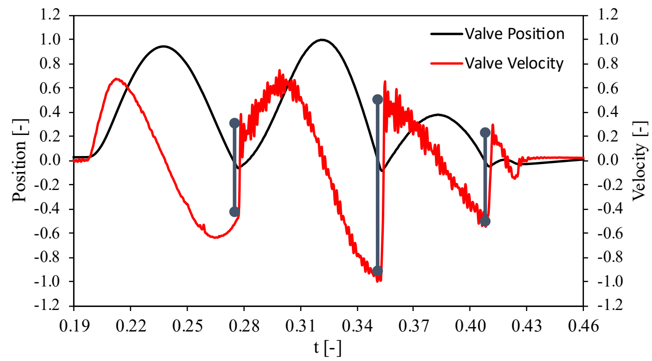
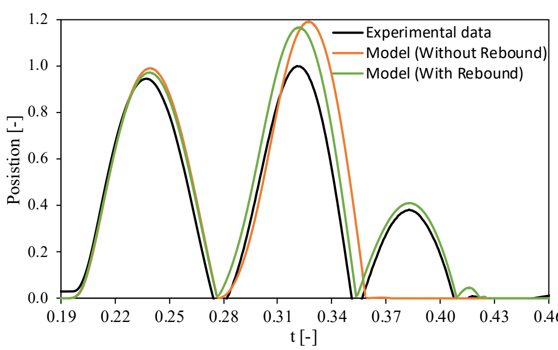
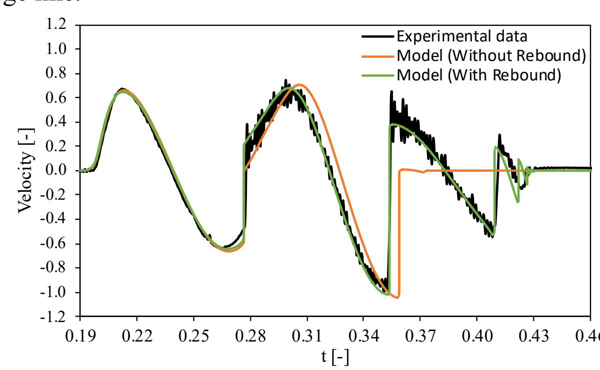


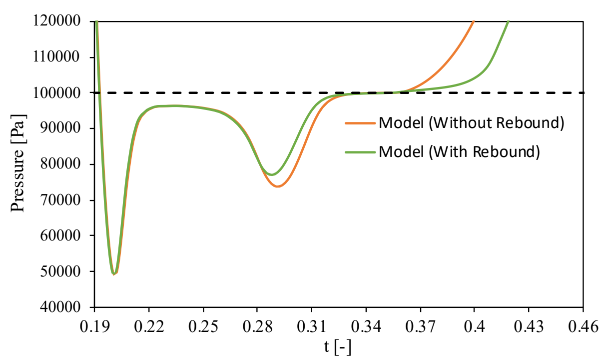
Figure 5: Experimental data for valve position and valve velocity; maximum speed and 2 bar pressure in the discharge line.



a) Valve position



b) Valve velocity



c) Cylinder pressure

Figure 6: Rebound effect at maximum speed with 2 bar pressure in the discharge line.

Table 1: Dimensionless velocity impact map (With Rebound)

Discharge Pressure [bar]		2	3	4
Compressor Speed [-]	1.0	0.48	-	-
	1.4	0.70	0.67	0.66
	2.1	1.00	-	-
	2.9	1.02	-	-

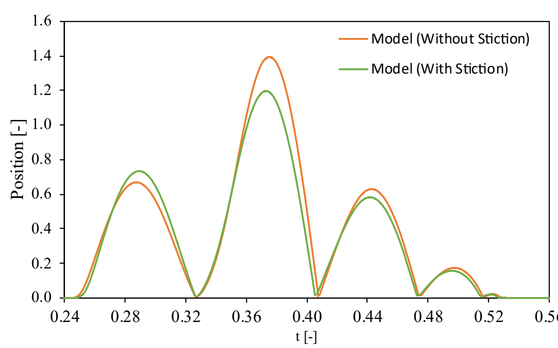
Table 2: Dimensionless velocity impact map (Without Rebound)

Discharge Pressure [bar]		2	3	4
Compressor Speed [-]	1.0	0.52	-	-
	1.4	0.74	0.72	0.72
	2.1	1.04	-	-
	2.9	1.05	-	-

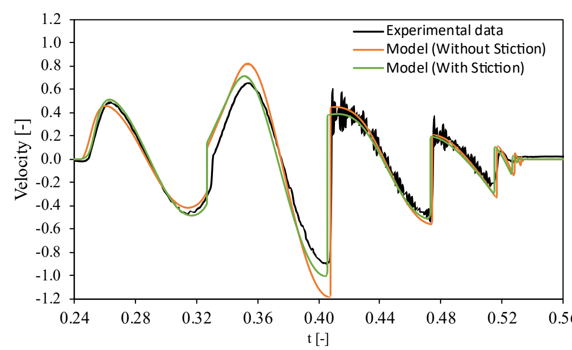
4.2 Oil Stiction

Oil stiction is a significant challenge for valve dynamic models. Although Eq. (3) provides a simple method to account the influence of oil stiction force, estimating the initial oil thickness h_0 remains a primary difficulty. Typically, this parameter is calibrated using experimental valve motion curves, since oil stiction tends to delay the valve opening. However, because the present measurements were not synchronized with the crank angle, the initial oil thickness was adjusted based on the experimental peak velocity observed at the first valve opening.

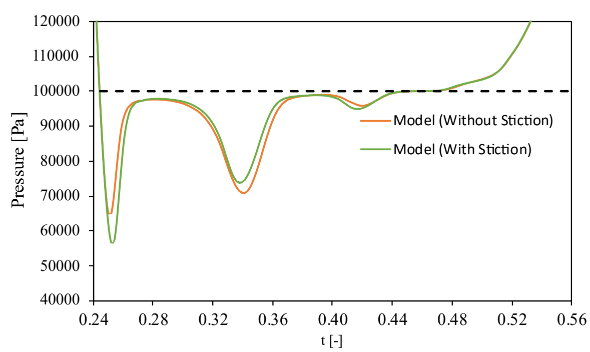
According to Eq. (3), as the initial oil thickness decreases, the stiction force increases, resulting in a significant delay in valve motion. Consequently, the pressure inside the cylinder drops further, leading to a larger opening during the first valve oscillation, as illustrated in Figure 7. With a larger initial opening, the valve experiences a higher impact velocity during the first oscillation. Hence, during the second oscillation, the valve starts its opening motion at a higher velocity due to the rebound effect. On the other hand, a model without the stiction effect shows a lower velocity at the start of the second oscillation, since the first impact velocity was lower. Similar to the rebound effect, the pressure in the cylinder decreases rapidly under this condition, resulting in a larger second valve opening and a higher impact velocity.



a) Valve position



b) Valve velocity



c) Cylinder pressure

Figure 7: Stiction effect at high speed with 2 bar pressure in the discharge line.

Essentially, in the current case, the increase in the first opening due to oil stiction results in a smaller second valve opening because of the combined effects of rebound and in-cylinder pressure drop following the first impact. Consequently, the model that does not account stiction force, overestimates the highest impact velocity at high compressor speed that occurs on the second valve oscilation. For the high compressor speed (2.1) and 2 bar pressure in the discharge line, this condition shows the greatest difference in impact prediction due to the oil stiction effect. Tables 3 and 4 present the dimensionless valve impact velocity maps predicted with and without the oil stiction, respectively. The oil stiction tends to reduce de valve impact prediction at high compressor speeds (1.4 to 2.9). At the minimum speed analyzed, the oil stiction did not influence the valve impact prediction. This is justified because it is the only condition analyzed where the highest impact velocity is predicted during the third valve oscillation, as shown in Figure 8.

Table 3: Dimensionless velocity impact map (With Stiction)

Discharge Pressure [bar]		2	3	4
Compressor Speed [-]	1.0	0.48	-	-
	1.4	0.70	0.67	0.66
	2.1	1.00	-	-
	2.9	1.02	-	-

Table 4: Dimensionless velocity impact map (Without Stiction)

Discharge Pressure [bar]		2	3	4
Compressor Speed [-]	1.0	0.48	-	-
	1.4	0.81	0.73	0.71
	2.1	1.18	-	-
	2.9	1.07	-	-

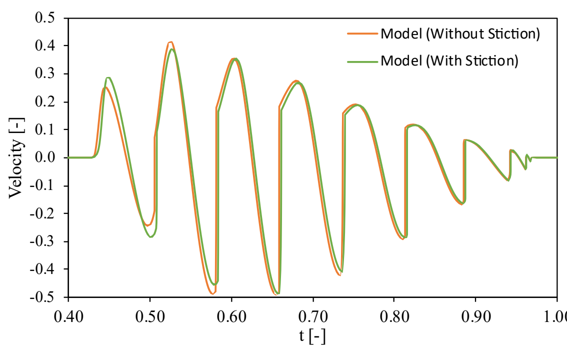


Figure 8: Stiction effect at minimum speed with 2 bar pressure in the discharge line.

4.3 Compressor Speed and Pressure Ratio

The compressor speed significantly influences valve dynamics. The model was assessed at four rotational speed while maintaining the same pressure ratio. As compressor speed increases, the duration of the suction process decreases. However, the characteristic frequency of valve oscillation appears to be independent of compressor speed. Consequently, the number of valve oscillations diminishes as the compressor speed increases, a trend shown in Figure 9. Except at the lowest compressor speed analyzed, the highest impact velocity was observed during the second valve impact, as supported by experimental data.

As the compressor speed increases, the rate of variation in the volume of the compression chamber also increases. This leads to a more rapid decrease in the pressure in the cylinder, thereby increasing the pressure load on the valve. Consequently, the suction valve shows larger displacements and higher impact velocities.

Table 5 and Figure 10 compare the variation in valve impact with compressor speed at a constant pressure ratio. The model predicts a trend similar to the experimental data, i.e., the impact velocity increases significantly from the minimum to the high speed, then stabilizes as the compressor is set to operate at maximum speed. However, at the lowest compressor speed, the model underestimates the impact velocity, while in all other conditions, it overestimates the velocitycompared to the measurements.

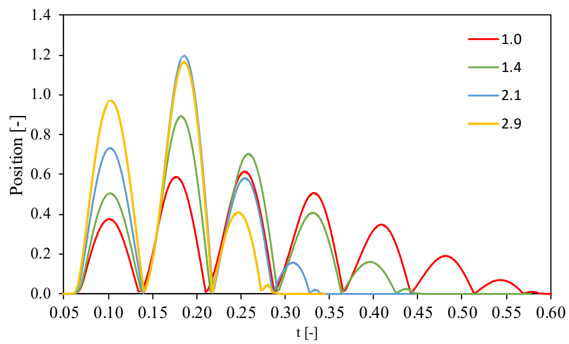
Similar to the effect of compressor speed, as the pressure ratio increases, the suction period decreases due to the clearance volume effect, as shown in Figure 11. Hence,the opening of the suction valve opening begins at different piston velocity depending on the pressure ratio, which could lead to an increase in valve opening. However, the model predicts an insignificant influence of the pressure ratio on valve impact velocity compared to the influence of compressor speed. This trend was validated by the experimental data shown in Table 6. It is important to note that in this study the pressure ratio was varied by changing the discharge pressure while keeping a constant suction pressure.

Table 5: Dimensionless velocity impact map (Compressor speed influence)

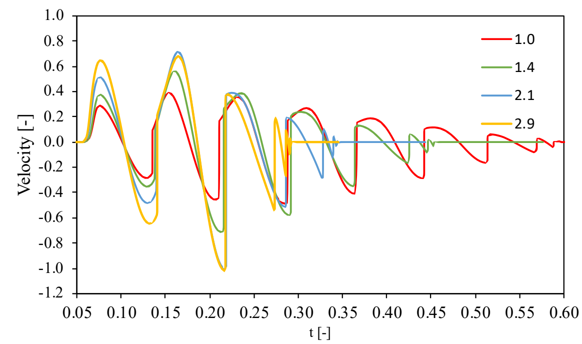
-		Experimental	Model
Compressor Speed [-]	**1.0**	0.52	0.48
	1.4	0.63	0.71
	2.1	0.88	1.00
	2.9	0.92	1.02

Table 6: Dimensionless velocity impact map (Pressure ratio influence)

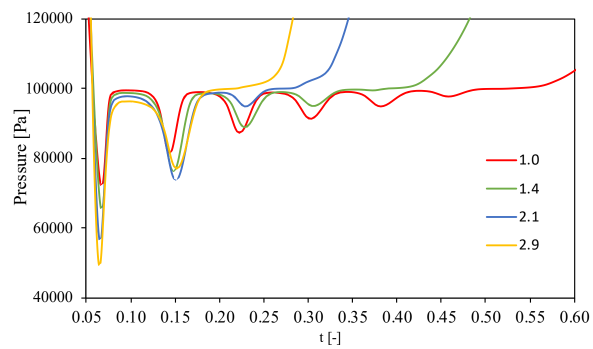
-		Experimental	Model
Pressure Ratio [-]	**2**	0.63	0.71
	3	0.61	0.68
	4	0.59	0.66



a) Valve position



b) Valve velocity



c) Cylinder pressure

Figure 9: Compressor speed effect with 2 bar pressure in the discharge line.

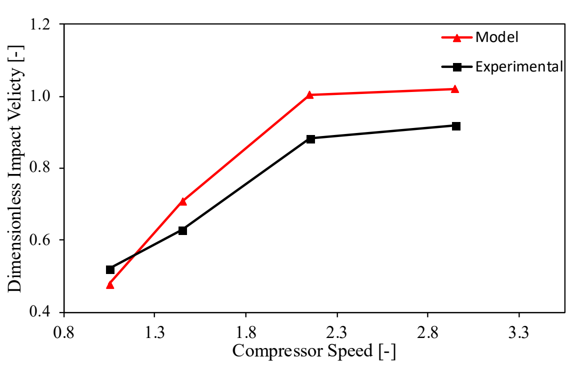


Figure 10: Dimensionless velocity impact curve.

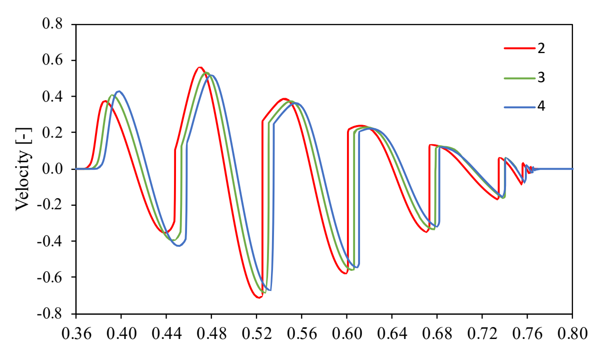


Figure 11: Pressure ratio effect at 4000 rpm.

5. CONCLUSIONS

The main objectives of this study were to explore the capabilities of a low-cost model in predicting suction valve impact velocity and to assess the influence of different parametera on this prediction, with the support of measurements of valve velocity. The study revealed that the rebound effect significantly influences the accuracy of valve dynamics predictions. Inaccurate contact modeling resulted in valve timing delays, leading to overestimated predcitions of impact velocity. Furthermore, the model without the rebound effect was unable to predict the final valve oscillation during high-speed operations, potentially leading to incorrect assessment of compressor performance. Similarly, oil stiction significantly affected the predicitons of impact velocity during high-speed operations, with the maximum impact velocity observed during the second valve impact. The model showed good agreement with experimental data regarding the effects of compressor speed and pressure ratio. A rapid increase in impact velocity was predicted when changing from low to high-speed operations, but almost no difference when changing from high to maximum speed. The pressure ratio had a negligible effect on the suction valve impact velocity, when maintaining constant the suction pressure. The findings from this study enhance the modeling accuracy in predicting valve dynamics under high-speed operations, contributing to improved performance and reliability analysis.

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