MODELING STRATEGIES FOR THE DYNAMICS OF REED TYPE VALVES

Fabian H. Shiomi, fabian@polo.ufsc.br
Cesar J. Deschamps, deschamps@polo.ufsc.br
POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics
Department of Mechanical Engineering
Federal University of Santa Catarina
88040-900, Florianópolis, SC, Brazil

Celso K. Takemori, Celso_K_Takemori@embraco.com.br
Research & Development Group
EMBRACO
89219-901, Joinville, SC, Brazil

Abstract. *Reed type valves are commonly adopted in small compressors due to their simplicity and low cost. As the compressor efficiency and reliability are greatly influenced by such valves, the correct prediction of their dynamics is essential for the compressor design. In this paper, a comparative analysis of two modeling approaches for valve dynamics is carried out. The simpler model considers the reed as a rigid body placed parallel to the valve seat and its displacement is restricted to a single-degree-of-freedom (SDOF) motion. In the second approach the reed is considered as an elastic body modeled by a multi-degree-of-freedom (MDOF) system and solved using a commercial solver based on the finite elements method. The turbulent and compressible flow through the valves is solved with the Shear Stress Transport (SST) turbulence model. Predictions of both models are in good agreement with experimental data, concerning phase and amplitude of the valve motion. The MDOF model exhibits a better accordance especially regarding the pressure pulsation in the suction chamber and p-V diagram, although being twice more computational expensive than the SDOF model.*

Keywords*: fluid-structure interaction, valve dynamics, reed valves*

1. INTRODUCTION

Reciprocating compressors usually adopt automatic reed-type valves for their simplicity and low cost. The compressor efficiency and reliability are greatly influenced by the discharge and suction valves and therefore the correct prediction of their dynamics is essential for design purpose. The simplest models are based on integral formulations for the compression process inside the cylinder and on semi-empirical parameters to evaluate the valves dynamics. Some of these analyses also employ one-dimensional models for the reed dynamics based on the finite elements method (Courtois et al., 2002).

As a result of the increase of computational resources, more complex analyses have become viable. For instance, Matos *et al.* (2002) developed a two-dimensional numerical model to simulate the operating cycle of a reciprocating compressor, including compressibility and turbulence effects on the fluid flow inside the cylinder and through the discharge valve whose dynamics was simultaneously solved by using a mass-spring-damper model.

The development of commercial CFD codes and the extensive application of computer-aided design (CAD) have made it possible to design three-dimensional numerical models for the complete reciprocating compressor geometry (Pereira *et al.*, 2007, Pereira *et al.*, 2008). Dynamics of both suction and discharge valves are modeled as a mass-spring-damper system with a single-degree-of-freedom.

By using commercial codes based on the finite elements methodology, fluid-structure interaction (FSI) models allowed the analysis of structural dynamics of compressor valves coupled the flow field. Kim *et al.* (2006) adopted a two-dimensional model to predict the dynamics of a suction valve of a reciprocating compressor. Three-dimensional models have also been proposed to investigate the dynamics of the suction valve of a linear compressor (Suh *et al.*, 2006) and the discharge valve of a reciprocating compressor (Kim *et al.*, 2008).

In the present paper, a comparative analysis of two different modeling approaches is carried out to investigate the dynamics of the suction valve of a small reciprocating compressor. A three-dimensional computational model for the fluid domain was constructed by using a commercial CFD package. The reed motion was coupled to the fluid flow field and computed through two different modeling approaches. The simpler model considers the reed as a rigid body placed parallel to the valve seat and its displacement is restricted to a single-degree-of-freedom (SDOF) movement. In the second approach the reed is considered as an elastic body modeled by a multi-degree-of-freedom (MDOF) system with a commercial multiphysics solver based on the finite element method (FEM).

2. MATHEMATICAL MODEL

2.1. Fluid Flow

The flow is solved through the Reynolds-averaged Navier-Stokes equations (RANS), in which the value of a computed variable represents an ensemble average over many engine cycles at a specified spatial location. The turbulent stresses are expressed through the concept of eddy viscosity μ_t, which is evaluated with the Shear Stress Transport (SST) model. To complete the system of equations required to solve the compressible flow, the Redlich Kwong equation of state was adopted for simulating the behavior of real gas.

2.2. Section titles and subtitles

In this study, the dynamics of the reed valve is represented by single-degree-of-freedom (SDOF) and multi-degree-of-freedom (MDOF) models. The former considers the valve to be parallel to the valve seat and its dynamics is given as follow:

$$m\ddot{u} + c\dot{u} + ku = F_p + F_o \tag{1}$$

where m, c and k are the reed mass, damping coefficient and stiffness, respectively and their values are determined experimentally. F_p is the force resulting from the flow pressure distribution on the reed surface and F_o represents any other force, such as reed pre-tension and stiction force caused by the presence of a lubricating oil film between the reed and valve seat. At last, the quantities u, $\dot{u}$ and $\ddot{u}$ stands for the instantaneous reed lift, velocity and acceleration, respectively.

In general, the dynamic response of a structure can be better represented by a MDOF model. In this formulation, the motion of the structure is assumed to be defined by the displacements of a set of discrete points. The equation of motion for a structural MDOF model can be written similarly as done in the SDOF model:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F_a\} \tag{2}$$

where $[M]$, $[C]$ and $[K]$ are the global structural mass, damping and stiffness matrices, whose components depend on the material properties and constitutive relations. $\{F_a\}$ is the applied load vector, which represents the force due pressure distribution on the reed surface, as well as other forces aforementioned. The displacement, velocity and acceleration components of each point of the structure is represented by the vectors $\{u\}$, $\{\dot{u}\}$ and $\{\ddot{u}\}$, respectively.

3. NUMERICAL METHODOLOGY

The computational model for the fluid domain was developed using a commercial CFD code (Ansys, 2006) based on the finite volume method. The fluid domain solution includes the suction muffler, the suction valve and the compression chamber formed by the cylinder and the piston, as illustrated in Fig. 1. The discharge system is neglected, since only the evaluation of the suction process is sufficient for the purpose of comparing the models for the valve dynamics.

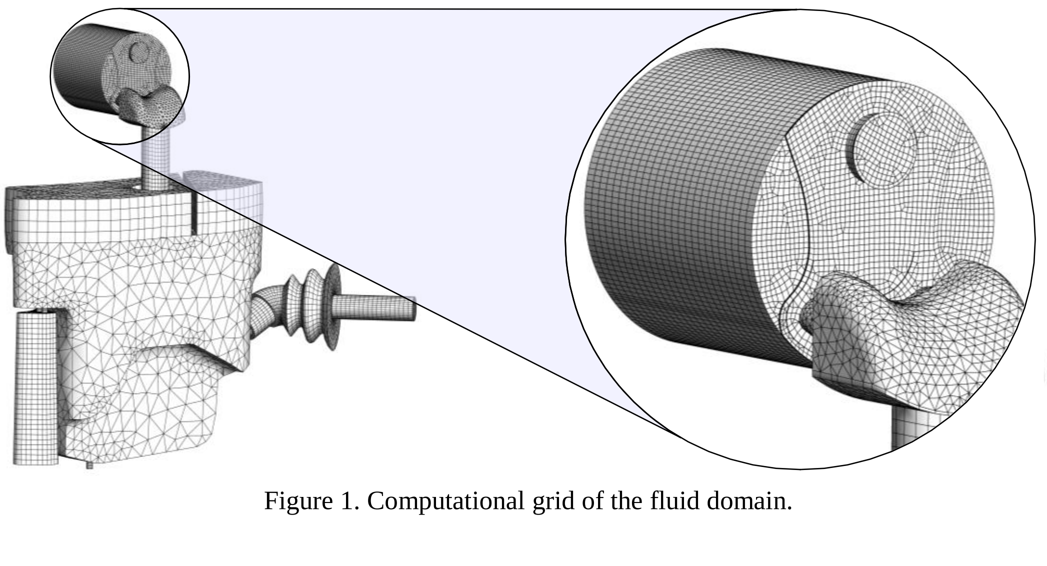


Figure 1. Computational grid of the fluid domain.

The presence of moving boundaries such as the piston and the suction valve requires a moving mesh strategy to simulate the compressor operation. The Second Order Backward Euler scheme was chosen for the discretization of the transient term. The interpolation of flow quantities at the control volume faces needed to compute the advection term was carried out with a high-order accuracy scheme.

The evaporation pressure was set as the boundary condition for the suction muffler and in order. The condensation pressure was imposed as the boundary condition for the discharge port so that at the top dead center of each cycle both models are subjected to the same condition. A uniform temperature was prescribed for the cylinder walls, whereas all other solid walls were considered to be adiabatic since no data was available to characterize the heat transfer in the suction muffler. To circumvent this difficulty, the temperature in the suction chamber was specified in accordance with available experimental data, so as to guarantee that the temperature of the gas entering the compression chamber had the correct value. The piston movement was defined by the crankshaft mechanism and the suction valve motion was described by the two models under analysis.

When SDOF model is adopted to solve the valve dynamics, no structural deformation is calculated since the reed is considered as a rigid body. In this case no structural model is created and the equation of motion (1) is solved simultaneously with the flow using an explicit Euler method. A stiction force of 0.2N due the presence of lubricating oil is considered to act on the suction valve up to a valve lift of 10 μm.

In order to compute the structural deformations, a MDOF model was developed with a commercial code (Ansys, 2007) based on the finite element method. The structural domain is composed by the reed and the gasket used in the valve assembly, as shown in Fig. 2. A layer of 20-node solid elements defines the structure of the reed while another layer specified for a different material defines the gasket.

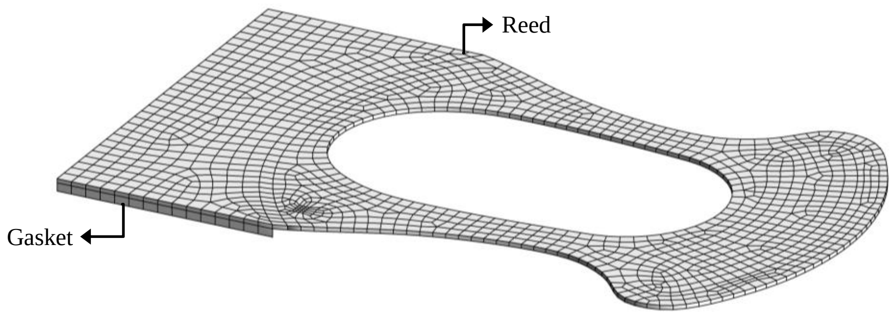


Figure 2. Computational grid of the structural domain.

Geometric nonlinearities are taken into account in the solution due the large deflections of the reed. As no valve stopper is adopted for suction valves only the contact between the reed top surface and the valve seat is considered, with a rigid-to-flexible contact model being defined by using surface-to-surface contact elements.

The fluid flow pressure distribution is considered on both surfaces of the reed, while all nodes from the gasket bottom surface have their displacement constrained. The same value of stiction force adopted in the previous model is considered for the MDOF model. The fully transient analysis is solved via the Newmark time integration method and the contact problem is solved with the Augmented Lagrange Multiplier method of solution.

In the present fluid-structure interaction (FSI) 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.

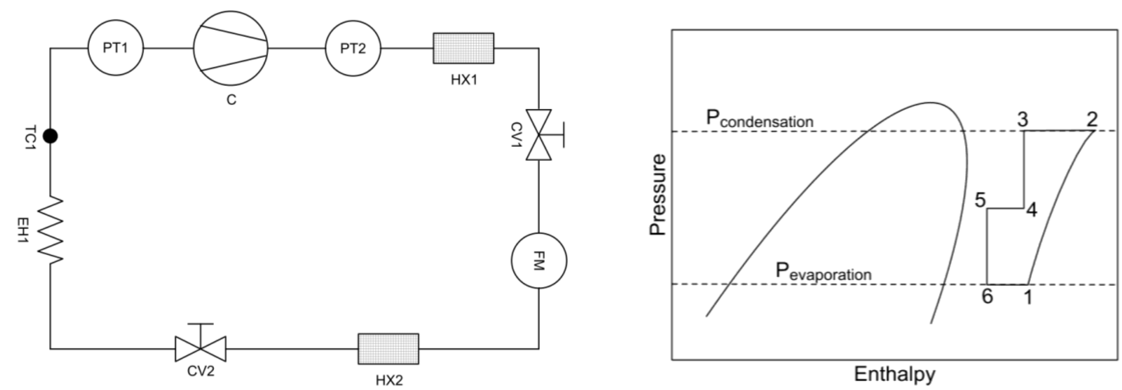
In the case of the SDOF model, the evaluation of the valve dynamics does not require the solution of a structural domain since there is no deformation. On the other hand, when multiple solvers (MDOF and CFD) are used to compute different domains their solutions must be externally coupled. In the present study, the fluid and structural domains were solved sequentially, with the data transfer between the solvers being managed by an enhanced multi-field coupling routine provided in software code (Ansys, 2007). Such a routine is responsible for mapping the fluid-solid grid interfaces and transferring the pressure load from the fluid flow domain to the solid and returning the displacement of each element of the reed.

The Profile Preserving Interpolation method was adopted to transfer the pressure load between the dissimilar meshes on the coupling interfaces. Results for pressure load and nodal displacement can be exchanged between both domains several times during a single time step until convergence is reached, but in the present analysis the exchange was limited to just one per time step. This restriction reduces the simulation time and does not affect significantly the solution accuracy since a small time step size was adopted. For both numerical models a time step size of 2×10^{-5}s was employed, corresponding to an increment of 0.432 degree of crankshaft angle.

4. RESULTS

In the present paper, numerical predictions from the two models are compared with experimental data for pressure in the suction and compression chambers and valve displacement. The measurements were carried out in a calorimeter facility for a 60 Hz compressor operating with R134a with a cooling capacity of 185 W. Measurements are obtained for energy consumption, mass flow rate, valve displacement, as well as fluid temperature and pressure in different parts of the compressor. All measurements represent an average of experimental data obtained in five independent tests for each condition analyzed. The uncertainty associated with measurements of compressor energy consumption is estimated to be ±4%.

As can be seen in the schematic representation of the calorimeter (Fig. 3a), the experimental setup is composed by pipelines, control valves (CV), a mass flow meter (FM), heat exchangers (HX), a thermocouple (TC) and pressure transducers (PT). The calorimeter is designed in a way that refrigerant fluid can flow through the high and low pressure lines in the superheated state, as indicated in the pressure-enthalpy diagram of Fig. 3(b). According to this arrangement, refrigerant enters the compressor (C) at point 1 and is compressed to point 2. Then, after being cooled by a heat exchanger (HX1) to point 3, the refrigerant is adiabatically throttled to an intermediate pressure level (point 4), by means of a control valve (CV1), in which its mass flow rate is obtained with a Coriolis flowmeter (FM). The refrigerant is cooled again with a heat exchanger (HX2) to point 5 and then throttled via a control valve (CV2) to the evaporation pressure (point 6). Finally, with the assistance of an electrical heater (EH1) and a thermocouple (TC1), the compressor suction line temperature is adjusted to the required superheated condition (point 1), completing the operating cycle. By adjusting the refrigerant charge in the system and the settings of control valves, heat exchangers and heaters, the thermodynamic conditions at points 1 and 2 can be adjusted to any required test condition. The main advantage of this experimental setup is that the compressor can be analyzed without any reference to a specific refrigeration system.



(a) (b)

Figure 3. Experimental setup: (a) layout schematic; (b) superheated cycle representation.

The first step in the experimental procedure is to submit the compressor and the pipeline to an adequate vacuum condition, in order to remove air, humidity and any other contaminant inside the system. Then, the system receives a charge of refrigerant and the flowmeter reading is set to zero. After the compressor is switched on, a period of approximately 6 hours is needed to establish a fully periodic operating condition because of compressor thermal inertia. During this process, the control valves in the high and low pressure lines have to be continuously adjusted to establish the specified suction and discharge pressure conditions and the required mass flow rate. The system is considered to have reached a steady operating condition when, in a period of 45 minutes, the temperatures in several locations of the compressor vary less than 1^oC and the mass flow rate and compressor energy consumption do not change more than 1%. When this condition is satisfied, data for energy consumption and mass flow rate are acquired during a period of 10 minutes. Finally, data for pressure in the suction and discharge chambers, pressure in the cylinder and displacements of suction and discharge valves are collected for 50 operation cycles of the compressor, allowing the evaluation of an average value for each quantity and, as a consequence, a reduction in the measurement uncertainty.

Piezoelectric pressure transducers are selected for the measurements in the suction and discharge chambers and inside the cylinder, due to its high response frequency, small size and reliability regarding the hostile conditions inside the compressor. It is not possible to flush mount the transducer at the cylinder wall and, therefore, a pressure tap hole is provided in the dead volume region. To correlate the pressure measurement with the crank angle, a sensing winding is assembled to the crankcase to collect the signal emitted by a magnet fixed to the crankshaft. The instantaneous crankshaft position is calculated taking into account the compressor mechanism characteristics. Small sensing windings are also assembled into the valve plate seat to give the valve lift according to the crankshaft position. More details about the typical instrumentation for reciprocating compressors can be found in Hjelmgren (2002).

Measurements for pressure and crank angle position, needed to construct the indicator diagram, are obtained with a sampling rate of 60 kHz. Such a high rate of sampling is adopted to satisfy the following requirements: i) correct characterization of pressure time variation in the compression chamber and in the suction and discharge chambers; ii) accuracy for the piston location along its motion; iii) adequate time resolution to allow the evaluation of power consumption from the indicator diagram.

Figure 4 depicts a comparison between numerical results and experimental data for valve lift, with an indication of the approximate crank angles at which the suction valve opens and closes. As can be seen, both models predict results in good agreement with the experimental data concerning both phase and amplitude. The amplitudes of the valve displacement obtained by the SDOF model are closer to the experimental data. However, the MDOF model returns better estimates for valve opening and crank angle at which the valve closes, including the amplitude of the third valve opening motion. It can also be seen that the MODF model predicts that the reed hits the valve seat twice before it closes, a feature that is neither observed in the SDOF model nor in the measurements.

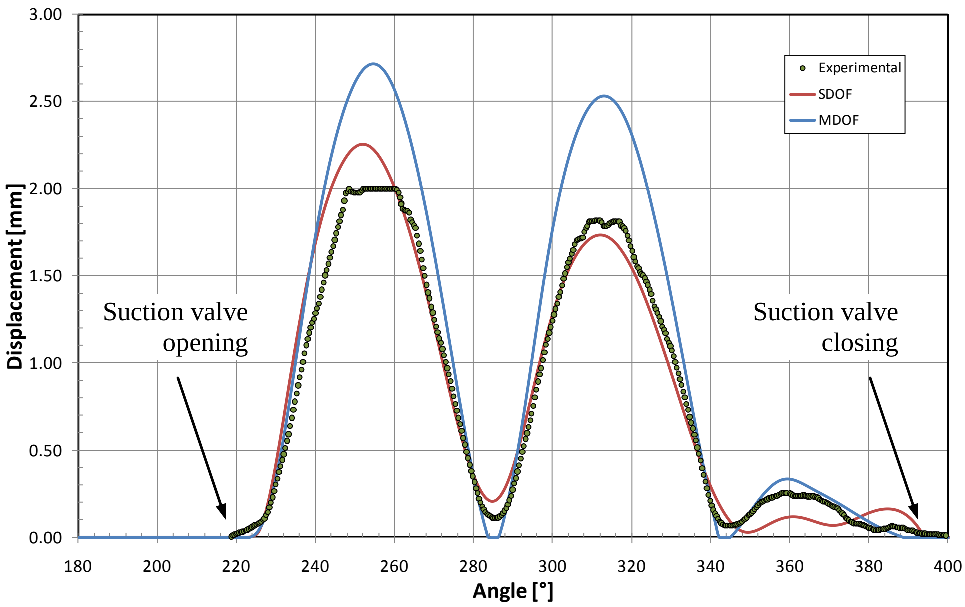


Figure 4. Suction valve displacement.

The influence of the valve opening and closing motions can be observed in the results for gas pressure inside the suction chamber and the cylinder during the suction process. Figure 5 illustrates the results for pressure inside the suction chamber along the cycle, referred to the pressure in suction line located at the compressor shell. When the suction valve opens at approximately 220° of crankshaft angle, indicated by the dashed line, gas start flowing from the suction chamber to the cylinder, which causes the pressure drop observed up to the angle of 240°. As the valve gets closer to its seat for the first time, a peak of pressure in the suction chamber occurs, right after the angle of 280°. The second valve opening causes another pressure drop and as it gets closer to the valve seat again one more peak is observed, this time at the angle of 340°. The suction valve closes after the top dead center (~32^o of crankshaft angle) and the pressure inside the suction chamber keeps pulsating until the valve opens again in the next cycle. Both numerical models are capable of predicting phase and amplitude of pressure pulsations in close agreement with to the experimental data, although the SDOF model under predicts the amplitude of the second and third peaks during the valve opening motion.

Figure 6 shows a comparison between numerical and experimental results for the suction process represented in a p-V diagram. The pressure values were normalized by the pressure condition in the suction line and the instantaneous volume of compression chamber was made dimensionless by the total volume swept by the piston. Both models are in good agreement with the experimental data during the period in which the valve is closed. On the other hand, when the valve is open the MDOF provides a slightly better prediction for the pressure variation inside the cylinder.

The compressor mass flow rate $\dot{m}$ is greatly affected by the valve dynamics and, therefore, the differences already observed between both models have an impact on $\dot{m}$. As be seen in Fig. 7, the MDOF model predicts higher values of mass flow rate during the suction process, whereas the magnitude of the backflow (negative mass flow rate) returned by the SDOF model is smaller. As a consequence of these two aspects, both models give similar results for the average mass flow rate along the cycle (approximately 3.85 kg/h).

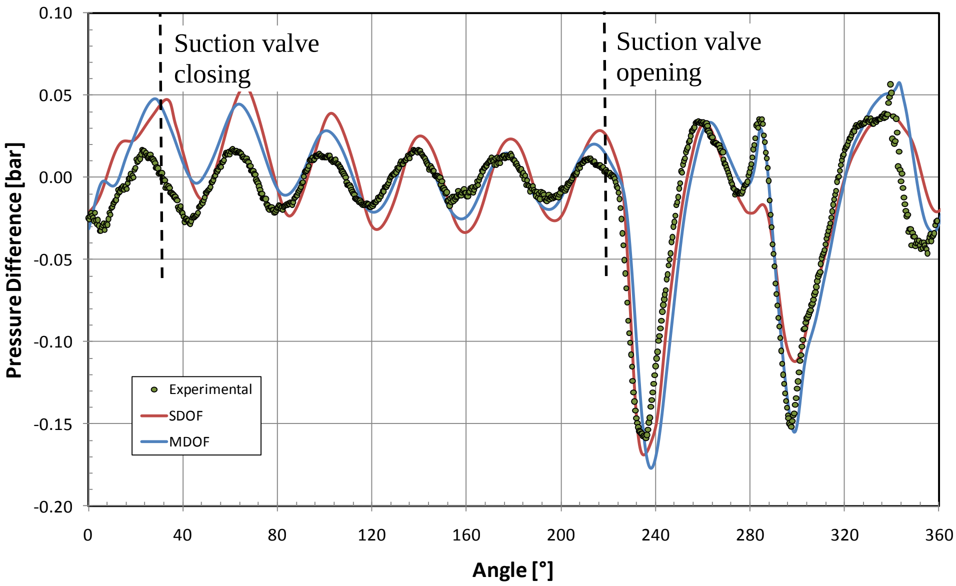


Figure 5. Pressure pulsation in the suction chamber.

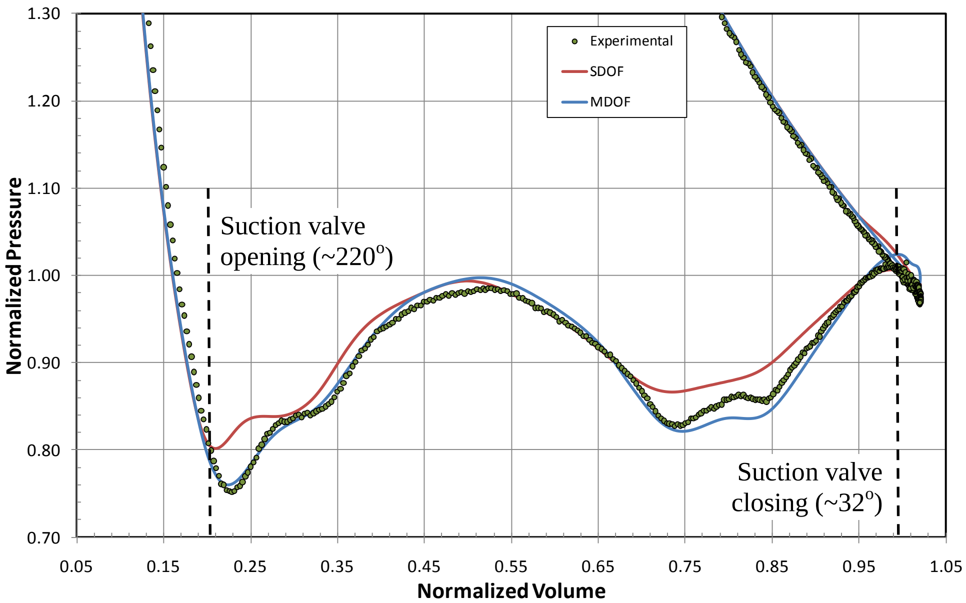


Figure 6 – Suction process represented in the indicator diagram.

Table 1 presents results of irreversibility in the suction valve and compressor efficiency. The power consumption of the suction process is mainly due the energy loss associated with flow restriction originated by the reed dynamics. In fact, flow restriction experienced by the gas entering the cylinder is smaller when the reed is modeled as a rigid body and this is the reason for the power consumption evaluated with the SDOF model being smaller. However, when the reed bending is included in the case of the MDOF model, the passage area that results for the flow is more representative of the actual compressor situation. Hence, the power consumption associated with the suction process (first column) obtained with the MDOF model is nearly the same value indicated by the measurement, whereas the SDOF model under predicts this quantity by 16%.

The volumetric efficiency is directly proportional to the compressor mass flow rate $\dot{m}$. As previously verified, both models provide similar values for $\dot{m}$ and, therefore, the same occurring for volumetric efficiency, as seen on the second column of Tab. 1. Nevertheless, it should be noticed that predictions for volumetric efficiency are 8% greater than the corresponding experimental data. Such a difference between numerical and experimental results can be associated with several aspects, such as uncertainty in the mass flow rate measurement (~1%), possibility of greater backflow in the prototype, cylinder dead volume not well specified in the models, errors in the predictions for valve dynamics and pulsating flow through the suction and discharge systems.

Compressor isentropic efficiency is related to the compression power consumption, which includes losses in the suction and discharge valves. As no discharge valve is included in both numerical models, only the energy loss associated with discharge orifice can be accounted. For this reason, the comparison between numerical and experimental results for isentropic efficiency does not consider the energy loss due the discharge process. Table 1 shows again that both models give similar results for the isentropic efficiency, being 6% greater than the experimental data.

FSI simulations have taken approximately twice longer the time spent by the simpler analysis (SDOF model). In order to simulate one complete compression cycle the simplified model takes approximately 14h while the MDOF model requires 28h to perform the same analysis. Such a difference is partially due to the solution of a structural domain and the coupling with the fluid flow solution in the MDOF model. Another factor that contributes for the MODF being more expensive is the necessity of evaluating the solid contact between reed and valve seat three times in each cycle. The numerical solution of the contact phenomenon is usually a lengthy iterative process due the high nonlinearities involved in the problem.

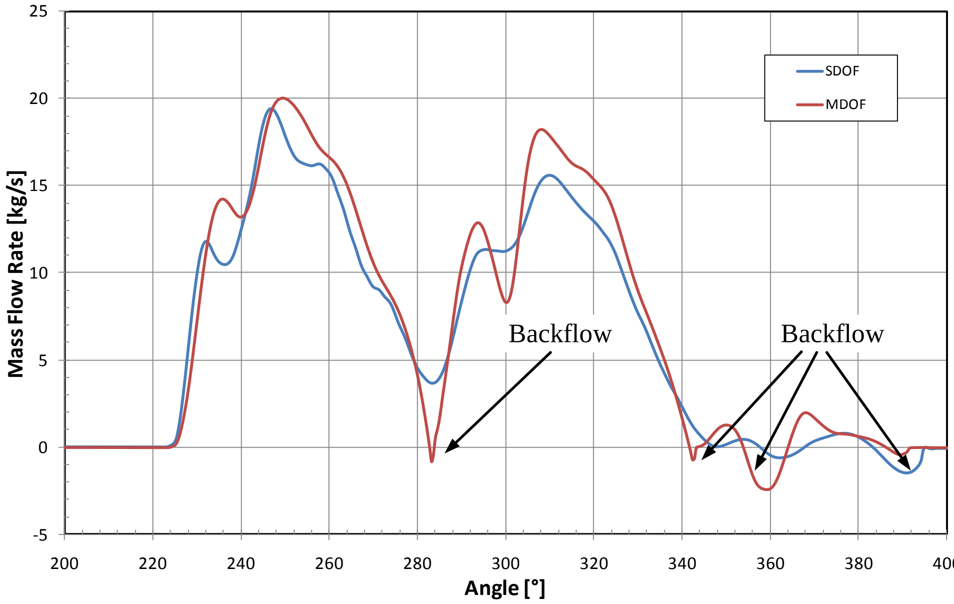


Figure 7. Instantaneous mass flow rate during the suction process.

Table 1. Experimental and numerical results for compressor global parameters.

	Power Consumption at Suction [W]	Volumetric Efficiency	Isentropic Efficiency
Experimental Data	2.73	0.76	0.88
SDOF Model	2.30	0.82	0.94
MDOF Model	2.77	0.82	0.93

5. CONSCLUSIONS

This paper has presented two different approaches for modeling the valve dynamics of small reciprocating compressors. The first method describes the valve dynamics through a single degree of freedom (SDOF) model, in which the reed is a rigid body moving parallel to the valve seat. The second model (MDOF) takes into account the reed bending, with deformations being solved via the finite element method coupled to the fluid flow field. Comparisons between numerical and experimental results were carried out for valve displacement, pressure pulsation in the suction chamber, p-V diagram, volumetric efficiency and isentropic efficiency. The MDOF model was seen to return a better agreement with experimental data, especially regarding pressure pulsation in the suction chamber and p-V diagram. However, it requires approximately 28h of computational time to simulate a single compression cycle, which is twice greater than the simulation time of the SDOF model. When the reed deformation is not the main interest of the analysis, the SDOF can offer reasonable results with a much smaller computational cost.

6. ACKNOWLEDGEMENTS

This study is part of a technical-scientific program between the Federal University of Santa Catarina and EMBRACO, partially funded by CNPq (Brazilian Research Council) through Grant No. 573581/2008-8 (National Institute of Science and Technology in Refrigeration and Thermophysics). Support from FINEP (Federal Agency of Research and Projects Financing) and CAPES (Coordination for the Improvement of High Level Personnel) are also acknowledged.

7. REFERENCES

Ansys Inc., 2006, ANSYS CFX 11.0 User’s Manual, Canonsburg/PA, USA.

Ansys Inc., 2007, ANSYS 11.0 User's Guide, Canonsburg/PA, USA.

Courtois S., Arnoult E., Wagstaff P., Gavric, L., 2002, On Finite Element Modeling of Valve Dynamics: Impacts, Oil Stiction, Gas Flow, Proceedings of the International Compressor Engineering Conference at Purdue, Paper C13-2.

Hjelmgren, J., 2002, Dynamic Measurement of Pressure – A Literature Survey, SP Swedish National Testing and Research Institute, SP Measurement Technology, SP REPORT 2002:34.

Kim J., Wang S., Park S., Ryu S., La J., 2006, Valve Dynamic Analysis of a Hermetic Reciprocating Compressor, Proceedings of the International Compressor Engineering Conference at Purdue, Paper C107.

Kim H., Ahn J., Kim D., 2008, Fluid Structure Interaction and Impact Analysis of Reciprocating Compressor Discharge Valves, Proceedings of the International Compressor Engineering Conference at Purdue, Paper 1112.

Matos, F.F.S., Prata, A.T. and Deschamps, C.J., 2006, A Two-Dimensional Simulation Model for Reciprocating Compressors with Automatic Valves, Proceedings of the International Compressor Engineering Conference at Purdue, Paper C053.

Pereira E.L.L., Deschamps C.J., Ribas Jr F.A., 2007, Performance Analysis of Reciprocating Compressors Through CFD Simulation, Proceedings of the International Conference on Compressors and their Systems, City University, London, pp. 309-318.

Pereira E.L.L., Deschamps C.J., Ribas Jr F.A., 2008, A Comparative Analysis of Numerical Simulation Approaches for Reciprocating Compressors, Proceedings of the International Compressor Engineering Conference at Purdue, Paper 1303.

Suh K., Heo D.N., Kim, H.S., CAE/CFD Application for Linear Compressor, Compressor, Proceedings of the International Compressor Engineering Conference at Purdue, Paper C032.

8. RESPONSIBILITY NOTICE

The author(s) is (are) the only responsible for the printed material included in this paper.