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ABSTRACT

The present paper describes a simulation model developed to predict the performance of rolling-piston compressors with special attention to the suction and discharge processes. The relevant input data required by the model, such as clearances between moving parts, valve stiffness and natural frequency and electric motor efficiency, were obtained experimentally. Correlations for effective flow and force areas associated with the suction and discharge processes were derived from flow simulations. It was found that the position of the rolling piston in relation to the suction and discharge ports must be included to fully characterize the effective flow and force areas. Numerical predictions of the thermodynamic inefficiencies associated with a R22 rolling-piston compressor were compared with measurements and good agreement was found at different operating conditions.

1. INTRODUCTION

The rolling piston compressor is composed of a compression chamber and a suction chamber separated from each other by a rolling piston and a vane (Figure 1). The rolling piston is assembled on the shaft eccentric, allowing it to rotate and orbit simultaneously within the cylinder. A vane, present in the slot of the cylinder, is pushed against the roller by a spring. This type of compressor does not need a suction valve for two reasons: (i) the compression chamber and suction chambers are separate, making the flow through the suction port a continuous process and (ii) the pressure inside the suction chamber is, in the largest part of the cycle, lower than the pressure in the suction line, reducing the possibility of backflow.

The suction process starts when the piston passes over the suction orifice. During the suction part of the cycle, gas is drawn through a suction port into the suction chamber as its volume is increased. Simultaneously, the compression and discharge processes take place in the decreasing volume for the other chamber, on the opposite side of the piston and vane. The gas is compressed until the pressure in the compression chamber becomes higher than the pressure in the discharge line. Then the discharge process takes place until the piston passes over the discharge orifice.

Different simulation models are available to predict the performance of rolling-piston compressors. The most common approach in the industry adopts lumped formulations for the suction and compression chambers (Gyberg and Nissen, 1984; Krueger, 1988; Ooi and Wong, 1997). Simulation models based on three-dimensional formulations have been developed with commercial codes that allow the integration of CFD and CAD models (Geng *et al.*, 2004; Liang *et al.*, 2010). However, despite recent advances in numerical methods, the computational cost of a full three-dimensional simulation is still impracticable for extensive compressor optimization in which several design alternatives have to be analyzed. In fact, lumped models offer a cost-effective alternative for preliminary optimization, whereas three-dimensional models can help understand complex physical phenomena during the compression cycle and define the best compressor design.

In lumped models, the mass flow rate in the suction and discharge processes and the flow induced force acting on the discharge reed are obtained with reference to the effective flow area coefficient (C_{A_v}) and effective force area coefficient (C_{A_F}), respectively.

The effective flow area coefficient (C_{A_v}), also known as the discharge coefficient, is a parameter related to the flow restriction imposed by viscous friction along the channel geometry and defined as:

$$C_{A_\mathrm{v}} = \frac{\dot{m}}{\dot{m}_{th}} \ , \tag{1}$$

where $\dot{m}$ e $\dot{m}_{th}$ are the actual and theoretical mass flow rate, respectively. The theoretical mass flow rate $\dot{m}_{th}$ is that of an isentropic compressible flow through a convergent nozzle with outlet area A, i.e.:

$$\dot{m}_{th} = A p_u \sqrt{\frac{2\gamma}{RT_u(k-1)}} \sqrt{\left(\frac{p_d}{p_u}\right)^{\frac{2}{\gamma}} - \left(\frac{p_d}{p_u}\right)^{\frac{(\gamma+1)}{\gamma}}} \ , \tag{2}$$

where p_u and T_u denote pressure and temperature at the flow upstream, p_d is the pressure at the flow downstream, γ is the specific heat ratio, and R is the refrigerant gas constant. Naturally, the flow condition through the valve can be either critical or subcritical. The actual mass flow rate $\dot{m}$ can be obtained experimentally or from numerical simulation. It is preferable to obtain this parameter via simulation because it is usually cheaper and faster than measurements, since the experimental apparatus is expensive and requires an experienced operator.

The dynamics of the discharge valve is characterized by the effective force area coefficient,

$$C_{A_f} = \frac{1}{A}\frac{F}{\Delta p} \ , \tag{3}$$

This coefficient is a function of the valve opening and position of the rolling piston and represents how effectively the pressure difference (Δp) is used to produce the opening force (F). More details on effective flow and force areas can be found in Soedel (2007).

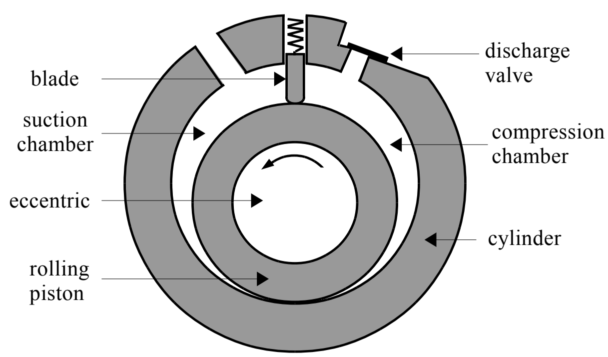

Figure 1: Compression mechanism of rolling piston compressor.

Despite many investigations on rolling piston compressors, it has not been found in the literature any study about the effect of the rolling piston on the effective flow and force areas. For instance, the effective flow area coefficient for circular ports, $C_{A_\mathrm{v}} = 0.6$ (Potter and Wiggert, 1991), is inappropriate to characterize the mass flow rate in the suction process when the rolling piston is near the suction port. Moreover, the mass flow rate in the discharge valve is affected by flow restrictions due to both the reed and the rolling piston.

Puff and Souza (1994) proposed $C_{A_\mathrm{v}} = 0.3$ to characterize the mass flow rate in the suction port and $C_{A_\mathrm{v}} = 0.7$ for the discharge valve, regardless of its opening. Although Puff and Souza (1994) found such values appropriate for the compressor model evaluated in their study, one should not expect the same finding in other compressor models. Kawai (1984) adopted a similar approach and experimentally characterized C_{A_v} for the different discharge ports. Ma and Bae (1996) provided a brief description of an experimental setup developed to measure effective force areas of the discharge valve of a rolling piston compressor considering the valve retainer.

This paper reports a method to obtain effective flow and force area coefficients of rolling piston compressors based on computation fluid dynamics (CFD). The method takes into account the influence of the rolling piston on the suction and discharge processes, as well as the presence of the reed in the case of the discharge port. The aim is to show the importance of correctly estimating the effective flow area and effective force area coefficients on the prediction of losses in the suction and discharge processes.

2. SIMULATION MODEL

The first step to elaborate the numerical model is to characterize the geometry of the suction and compression chambers. Then, these chambers are split into two solution domains, one for the suction process and the other for the discharge process. Figure 2 shows the suction and discharge domains when the rolling piston occupies the crank angle θ = 240°. Naturally, the suction and discharge domains vary according to the position of the rolling piston.

The simulation of the fluid flow in the suction domain is simpler than in the discharge domain, since only the position of the rolling piston has to be considered in relation to the suction port. On the other hand, the fluid flow in the discharge domain is affected by both the rolling piston and the reed valve.

2.1 Suction Port

The effective flow area coefficient of the suction port was obtained as a function of the rolling piston position, by solving the flow for twenty-five solution domains in crank angles varying from 0° to 360°. At each crank angle, a different grid was used to discretize the solution domain. The grids were sufficiently refined so as to solve the fluid flow properly, but to avoid excessive computational cost. The following steps were necessary to prepare the simulation model: (i) prescription of boundary conditions; (ii) specification of control parameters for the iterative solution procedure, such as convergence criteria; (iii) identification of parameters necessary to evaluate the effective area coefficients.

Figure 3 shows the three boundary conditions that were used to simulate the flow through the suction port: (i) mass flow rate at the inlet of the suction tube, (ii) pressure at the flow outlet; (iii) no-slip flow conditions for solid walls (cylinder and rolling piston). All simulations were carried out with the commercial code Ansys CFX for the condition of stationary flow. Since the motion of the rolling piston was neglected in the model, it was necessary to define an outflow boundary in the solution domain. Following this strategy, the flow was assumed to enter the solution domain through the suction tube and to exit through the top and bottom surfaces of the compression chamber, as depicted in Figure 3. The solution of the flow through the suction port allows estimation of the effective flow area coefficient ($C_{A_{v,s}}$) following Equation (1) that can be used in the simulation of rolling piston compressors.

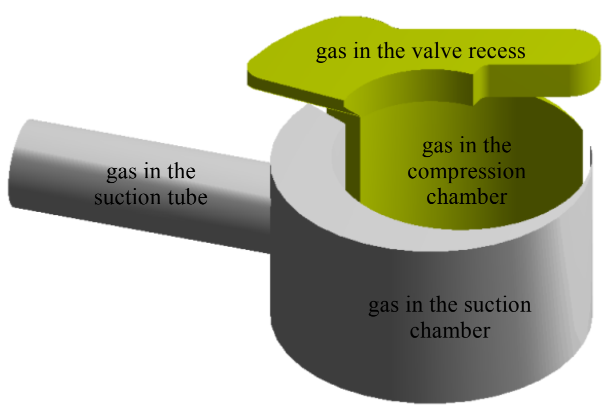

Figure 2: Suction and discharge domains.

2.2 Discharge Valve

The simulation of the discharge domain requires a more elaborate treatment and higher computational effort when compared to the suction domain. In fact, the presence of the discharge reed valve implies, in further modeling steps: (i) the calculation of effective force area coefficients to predict valve dynamics, (ii) more refined computational grids, since the valve opening can be as small as 0.1mm; (iii) greater number of simulations (at least seven valve openings for each crank angle analyzed). However, the number of crank angles could be reduced from twenty-five to ten because the valve remains closed during most of the compression cycle.

As illustrated in Figure 4, the boundary conditions used for the flow through the discharge valve are: (i) mass flow rate at the inlet section, (ii) pressure at the outflow surface and (iii) no-slip flow conditions for solid walls (cylinder, rolling piston and discharge valve). In the solution domain, that was adopted to simulate the discharge process, the fluid flow enters through the bottom surface of the compression chamber and exits through the valve gap.

The numerical procedure adopted in the calculation of the effective flow area coefficient ($C_{A_{v,d}}$) is similar to that made for the suction domain, but also takes into account the opening of the discharge valve. The numerical results for effective flow area coefficients were used to obtain a correlation applicable to the simulation of rolling piston compressors. Further details on the numerical models for the suction and discharge processes are available in Brancher (2013).

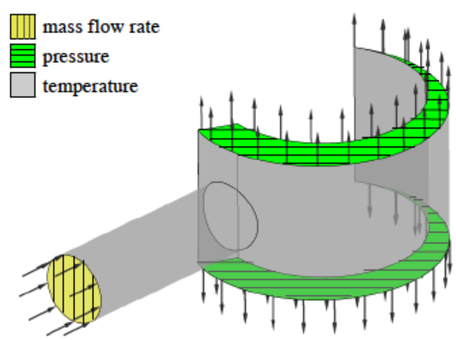

Figure 3: Boundary conditions for suction domain.

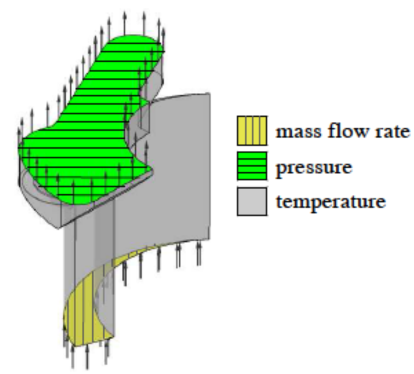

Figure 4: Boundary conditions for discharge domain.

3. RESULTS

3.1 Suction Port

Because there is no valve in the suction port, the effective flow area coefficient is the only parameter required to evaluate the mass flow rate entering the suction chamber. Figure 5 shows the variation of the effective flow area coefficient ($C_{A_{v,s}}$) as a function of the crank angle. As can be seen, the coefficient is zero in the beginning of the suction cycle and hence, the same occurs with the mass flow rate. The suction process only starts after the rolling piston reaches the suction port. It was found that during the period in which the piston moves over the suction port (approximately between the crank angles of 14° and 40°), the effective flow area coefficient increases gradually as a result of flow restriction due to the rolling piston.

As the rolling piston moves away from the suction port, its influence on the flow is gradually decreased. Consequently, the effective flow area coefficient is increased until it reaches a maximum value in the crank angle, which corresponds to the position of greatest distance between the rolling piston and the suction port. The crank angle of this maximum value corresponds to 207°, while the center of the suction port is located at the crank angle of 27°. After the crank angle of 207°, the rolling piston gets closer to the suction port again, increasing flow restriction there and decreasing the effective flow coefficient.

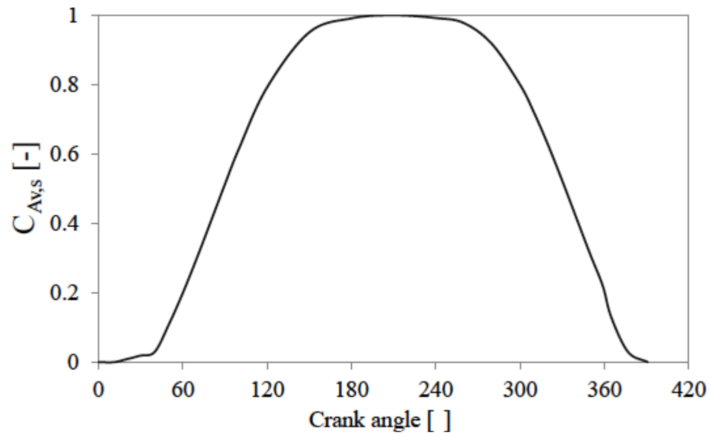

Figure 5: Effective flow area coefficient in the suction port as a function of crank angle.

3.2 Discharge Valve

As already explained, the simulation of the flow in the discharge process is more complex due to the presence of the reed valve rolling piston in the solution domain. Therefore, the effective flow area coefficient, $C_{A_{v,d}}$, is a function of both the valve opening and the proximity of the piston.

The smallest crank angle simulated for the discharge domain was 195°, since the valve never opens before the crank angle of 200° in different operating conditions. Figure 6 shows results for $C_{A_{v,d}}$ as a function of the crank angle and valve opening. As expected, the effective flow area coefficient increases with valve opening, since the higher the opening is, the higher the available area for fluid flow is. However, $C_{A_{v,d}}$ is seen to decrease in a range of crank angles for each of the valve openings due to the proximity of the piston. Since flow restriction decreases with valve opening, the restriction due to the rolling piston is more significant for large valve openings.

Figure 7 shows results for effective force area coefficient, C_{A_f}, as a function of valve opening and rolling piston position (crank angle). The effective force area coefficient suffers a considerable reduction for small valve openings because the flow reaches high velocity levels, reducing the pressure load on the valve and hence the flow induced force F. It is also interesting to note from Figure 7 that C_{A_f} is greater at smaller crank angles, as a consequence of the high mass flow rate that increases flow induced force F on the valve. As the crank angle increases, flow restriction due to the piston also increases, decreasing the mass flow rate and the force on the valve.

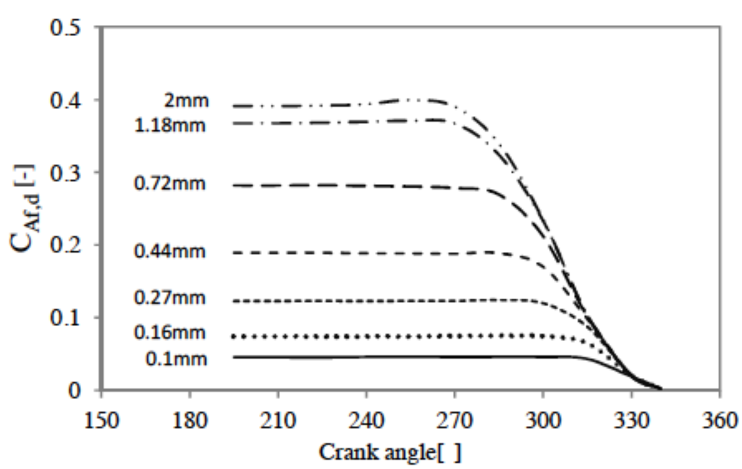

Figure 6: Effective flow area coefficient for the flow through the discharge valve.

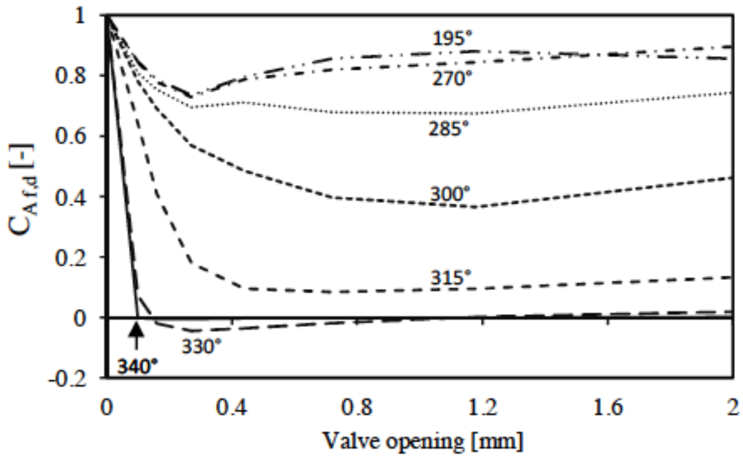

Figure 7: Effective force area coefficient for the flow through the discharge valve.

3.3 Compression Cycle

The rolling piston compressor studied in this paper was designed to operate in HBP (high back pressure) conditions, represented by the pair of evaporating and condensing temperatures [7.2°C/54.4°C]. The thermodynamic performance of the compressor was experimentally analyzed via p-V diagram. A calorimeter facility was employed to establish, and control, the operating conditions in which the compressor was tested.

The first step in the experimental procedure is to submit the compressor and the pipeline to an adequate vacuum condition, in order to remove the air, humidity, and any other contaminant inside the system. The system then receives a charge of refrigerant and the flow meter reading is set to zero. After the compressor is switched on, a period of approximately 4 hours is needed to establish fully periodic operating conditions 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 compressor is considered to have reached fully periodic operating conditions when the temperatures at several locations of the compressor vary less than 1^{o}C and both the mass flow rate and compressor energy consumption do not change more than 2% during one hour.

Experimental and numerical results for the compressor p-V diagram in Figure 8 show good agreement. The small difference seen during the initial stage of the suction process occurs because the model predicts a greater pressure drop in the suction port. However, the experimental data shows that the pressure in the suction chamber is above the suction line for a certain period of time. This could be caused by leakage at the top of the blade due to the instrumentation of a pressure transducer required for the p-V diagram.

The pressure drop in the suction chamber, demonstrated by the numerical results, is explained by gas expansion during the period in which the rolling piston blocks the suction port. Immediately after the piston passes over the suction port, gas is admitted into the suction chamber and the pressure begins to rise. With the advance of the rolling piston, the pressure in the suction chamber tends to reach equilibrium with the pressure in the suction line, showing only small fluctuations.

At the end of the suction process, both results show that the pressure in the suction chamber is slightly above the pressure in the suction line, which could cause a backflow through the suction port and reduce the volumetric efficiency. Despite some differences between numerical and experimental results in Figure 8, the agreement is satisfactory for the lumped formulation adopted in this study.

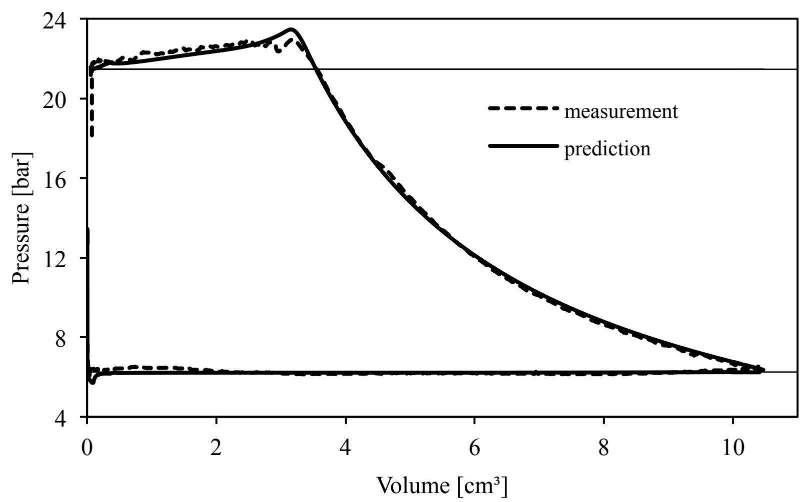

Figure 8: Experimental and numerical results for p-V diagram. Simulation model with effective area coefficients from CFD.

Figure 9 presents a comparison between experimental and numerical results of power losses in the suction and discharge processes. As can be seen, the simulation model with effective area coefficients C_{A_v} from CFD is capable of predicting such losses within 10% of the measured value. However, when the model adopts the values of 0.3 and 0.7, as suggested by Puff and Souza (1994) for C_{A_v} in the suction and discharge processes, respectively, predictions are quite different in comparison with measurements. As expected, this discrepancy is also reflected in the p-V diagram (Figure 10).

The results discussed in this section demonstrate that CFD is an efficient way to characterize the suction and discharge processes in rolling piston compressors, since it can take into account the influence of the rolling piston and discharge valve.

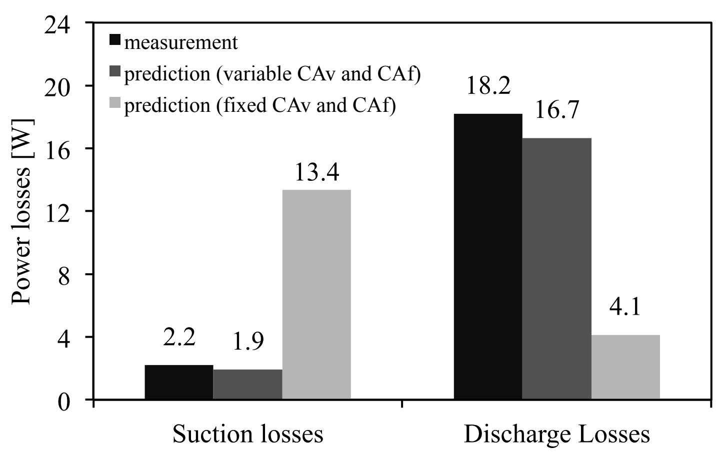

Figure 9: Predicted and measured power losses in the suction and discharge processes.

4. CONCLUSIONS

This paper presented a new method to determine effective flow area and effective force area coefficients for rolling piston compressors, considering the influence of both the piston and the discharge valve on the suction and discharge processes. Such coefficients were used as input data in the simulation model of the compressor. Predictions for the p-V diagram and power losses in the suction and discharge processes were found in agreement with measurements. On the other hand, the adoption of constant values for such coefficients in the simulation model was shown to be of limited use in compressors, other than that from which such values were calibrated.

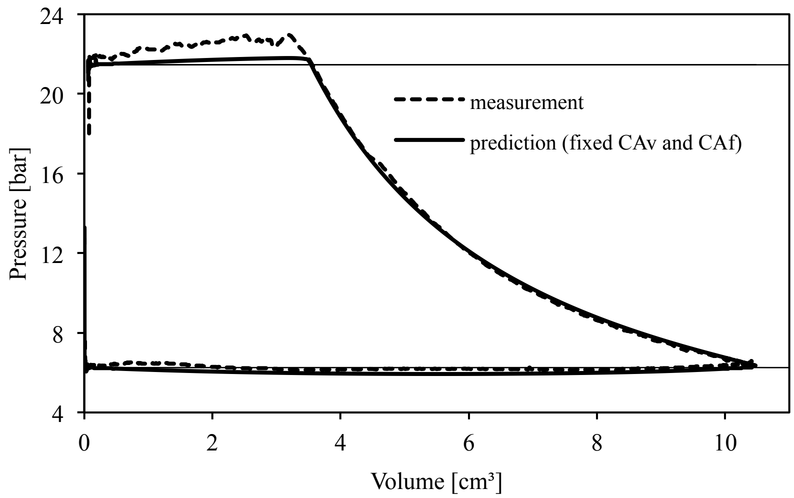

Figure 10: Experimental and numerical results for p-V diagram. Simulation model with fixed effective area coefficients proposed by Puff and Souza (1994).

REFERENCES

Brancher, R.D., 2013, *Development and Validation of a Model to Predict the Performance of Rolling Piston Compressors* (in Portuguese), M.Sc. Thesis, Federal University of Santa Catarina, Brazil.

Geng, W., Liu, C.H., Wang, Y.Z., 2004, The Performance Optimization of Rolling Piston Compressors based on CFD Simulation, *Proc. Int. Compressor Engineering Conference at Purdue*. paper 1621.

Gyberg, F., Nissen, H.S., 1984 A Simulation Model for Fixed Vane Rotary Compressor Using Real Gas Properties. . *Proc. Int. Compressor Engineering Conference at Purdue*, p. 33-39.

Kawai, H., 1984, Efficiency Improvement in Rolling Piston Type Rotary Compressors. *Proc. International Compressor Engineering Conference at Purdue*, paper 467.

Krueger, M., 1988, *Theoretical Simulation and Experimental Evaluation of a Hermetic Rolling Piston Rotary Compressor*, M.Sc. Thesis, Purdue University.

Liang, S., Xia, S., Kang, X., Zhou, P., Liu, Q., Hu, Y., 2010, Investigation of Refrigerant Flow Simulation and Experiment of Rolling Piston, *Proc. Int. Compressor Engineering Conference at Purdue*, paper 1955.

Ma, Y.C., Bae, J.Y., 1996, Determination of Effective Force Area and Valve Behavior on the Rolling Piston Type Compressor, *Proc. International Compressor Engineering Conference at Purdue*, paper 1134.

Ooi, K.T., Wong, T.N., 1997, A computer simulation of a rotary compressor for house-hold refrigerators, *Applied Thermal Engineering*, vol. 17, no. 1: p. 65-78.

Pereira ,E.L.L., 2006, *Analysis of Automatic Valve Systems of Reciprocating Compressors* (in Portuguese). M.Sc. Thesis, Federal University of Santa Catarina.

Potter, M.C., Wiggert, D.C., 1991, *Mechanics of Fluids*, USA: Prentice Hall, p. 290-291.

Puff, R., Souza, E.A., 1994, *Numerical Tool for Analysis of Performance of Rolling Piston Compressors* (in portuguese), Technical Report, EMBRACO.

Soedel, W., 2007, *Sound and Vibration of Positive Displacement Compressors*, CRC Press.

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