A NUMERICAL MODEL TO PREDICT THE THERMODYNAMIC PERFORMANCE OF VARIABLE CAPACITY COMPRESSORS

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Abstract. *The present paper considers the assessment of a numerical model to simulate the thermodynamic performance of variable capacity compressors (VCC), with especial attention to superheating. Each component is mathematically described, with an account for piston displacement as a function of crankshaft angle, thermodynamic processes inside the cylinder, valve dynamics and gas pulsation in mufflers. Several parameters are calculated for a compressor cycle, such as temperature, cooling capacity, isentropic and volumetric efficiencies. Thermodynamic properties for the refrigerant are evaluated through a program link to an external library. The compressor internal temperatures are supplied as input for the simulation program. This is accomplished by a routine that evaluates the temperature in eight control volumes through energy balances. The results show the model is limited to a range of compressor speed close to that adopted for its calibration condition.*

Keywords*: compressor, heat transfer, computational methodologies, speed variable*

1. INTRODUCTION

Ribas Jr. *et al.* (2008) presented the sources of thermodynamic irreversibilities in a high-efficiency compressor with a capacity of 900 BTU/h (ASHRAE / LBP) operating with R134a. They showed that the compressor performance is quite affected by its thermal profile due to gas superheating, which is responsible for almost 49% of the overall compressor thermodynamic loss. Superheating is associated with heat generated and released inside the compressor due to the compression process, friction in the bearings and irreversibilities of the electric motor. This useless superheating provokes an increase in compression work per unit mass and a reduction in volumetric efficiency, since the lower is the gas density in the suction chamber the worse will be the volumetric efficiency. Superheating losses are affected by many factors, which interact among themselves in a nontrivial manner. For instance, the gas heating in the suction process can take place at the muffler walls that are warmed by the gas inside the compressor shell. Yet, the temperature of the gas inside the compressor shell is a result of heat being dissipated by hot sources, such as the gas discharge tube, and through the shell wall to the external ambient.

From the thermodynamic viewpoint, a direct, well insulated connection, between the evaporator and the compression chamber would be an effective solution to this problem. However, as considered by Meyer and Thompson (1990), other project requirements such as noise attenuation, separation of oil mixed with refrigerant and facilitation of start-up demand more elaborated proposals with simultaneous solutions to the suction system. Several papers related to heat transfer in reciprocating compressors have been published over the last twenty years. The research in the area considers experimental investigations and the development of analytical and computational models to allow the compressor thermal management. Efforts in this area are justified by the great potential of increasing performance for designs that take into account the thermal dynamics.

Meyer and Thompson (1990) presented one of the first modeling proposals to predict heat transfer in compressors, in which the suction temperature is found from energy balance in some components of the compressor. Convective and radiative heat transfer contributions were taken into account, with convective heat transfer correlations being obtained part from the literature, part experimentally. On the other hand, conduction heat transfer was neglected. It was observed that predictions of temperature levels in the suction path followed the same trends of experimental data. However, the authors found some discrepancy between numerical and experimental results in some specific operating conditions. Furthermore, they observed that the overall heat transfer coefficients were not influenced by the type of refrigerant fluid used in the tests. Meyer *et al.* (1990) observed that each 6ºC increase in superheated fluid temperature implies degradation of 1% in the overall efficiency. Finally, they concluded that a new analysis was required so as to include leakages in the cylinder and flow pulsation.

Todescat *et al.* (1992) also proposed an integral numerical analysis based on overall heat transfer coefficients for the compressor components. The compression chamber was solved under the assumption of cyclic steady-state operating condition, including the flow through suction and discharge valves, leakage in the piston and cylinder clearance and heat transfer inside the cylinder. The authors evaluated the overall heat transfer coefficients with reference to experimental data for mass flow rate and temperature in each component. They found that such coefficients remain constant with variations in the compressor operating condition. Therefore, the model was adopted to predict superheating in different operating conditions, bay changing evaporation and condensation pressures. Numerical results were in reasonable agreement with experimental data, despite not being able to predict temperature distribution in solid regions.

With the purpose of describing more accurately the temperature levels in the compressor components, Ooi (2003) and Almbauer *et al.* (2006) proposed a new lumped thermal conductance approach. Ooi (2003) divided the compressor into 46 solid parts and, by simplifying complex geometries, predicted heat transfer via selected standard convective correlations from the literature. The author emphasized caution with the geometric simplifications adopted in the model, even though the results showed an agreement with experimental data within 20%. Almbauer *et al.* (2006) analyzed heat transfer in the cylinder, valve plate and cylinder cover, where the highest levels of temperature are observed. The boundary conditions for the solid surface were obtained by combining numerical results for the flow and experimental measurements of temperature. Heat transfer functions were proposed to estimate conduction in solid components, but such functions were not found to be valid for all operating conditions.

Ribas (2007) and Schreiner *et al.* (2009) proposed a hybrid method to calculate heat transfer in the compressor. In their methods, experimental data was used to solve heat transfer inherent to the fluid flow, as proposed by Todescat *et al.* (1992), serving as boundary condition for the solid components, which were then solved by a differential formulation. In addition, temperatures measured at some points in the solid regions also served as boundary conditions. Results for temperature fields in the components studied were obtained with an inverse method. This hybrid method is one of the most robust solution methods for thermal analysis of compressors.

The choice of a method for thermal analysis depends on the specified levels of computational cost and accuracy. Differential approaches, such as the hybrid method, can provide details about the temperature field in solid regions at the expense of a higher computational cost. Yet, lumped methods are able to provide predictions for temperature in some strategic points in the compressor with a much lower computational cost, being convenient for optimization of moderate changes in geometry. This paper reports an assessment of the method proposed by Todescat *et al.* (1992) applied to a variable speed compressor, aimed at verifying its adequacy to analyze changes due to variations in the compressor speed.

2. MATHEMATICAL MODELLING

Figure 1 presents a schematic view of the reciprocating compressor and some of its main components. In small reciprocating compressors, refrigerant at low pressure is directed from the suction pipe to the compression chamber through the suction muffler. The compression chamber and the indicator diagram for a typical cycle are depicted in Fig. 2. When the piston moves downwards, it reaches a position where low pressure vapor is drawn in through the suction valve, which is opened automatically by the pressure difference between the cylinder and the suction chamber. The vapor keeps flowing in during the suction stroke as the piston moves towards the bottom dead center (BDC), filling the cylinder volume with vapor at suction pressure, p_s. The suction process is represented by curve A-C in the indicator diagram of Fig. 2b. After reaching the BDC, the piston starts to move in the opposite direction, the suction valve is closed, the vapor is trapped, and its pressure rises as the cylinder volume decreases. Eventually, the pressure reaches the pressure in the discharge chamber, p_d, and the discharge valve is forced to open. After the opening of the discharge valve, the piston keeps moving towards the top dead center (TDC), represented by point A, with high pressure gas flowing through the discharge muffler towards the discharge pipe.

2.1. Compression Process

In the simulation model of the compression chamber, the thermodynamic state of the fluid is evaluated from the energy equation written for the integral formulation (Ussyk, 1984). Hence, non-spatial inhomogeneities of the gas temperature field inside the cylinder are neglected and single values for pressure, temperature and specific volume describe the instantaneous thermodynamic state of the fluid. Some of the thermodynamic properties such as pressure, internal energy and viscosity, are obtained from libraries provided by REFPROP (NIST, 2008). Convective heat transfer at the cylinder wall is calculated with the Nusselt correlation proposed by Annand (1963), whereas the temperature at the cylinder surface has to be prescribed. Gas leakage in the cylinder-piston gap is modeled as laminar flow between two infinite flat plates, moved by the pressure difference between the compression chamber and the internal environment of the compressor.

In the present study, a one-dimensional simulation method (Deschamps *et al.*, 2002) solves the transient compressible flow in the suction and discharge systems. Valve displacement is modeled by a one-degree of freedom mass-spring model:

$$m_{\mathrm{eq}} x + c\,\dot{x} + k\,\ddot{x} = F_{\mathrm{p}} + F_{\mathrm{o}} \tag{1}$$

where m_{eq}, c and k are the reed equivalent mass, damping coefficient and stiffness, respectively, which have to be specified. On the other hand, F_p is the flow induced force on the reed and F_o can accommodate any other force, such as reed pre-tension and also stiction that may occur due to the presence of a lubricating oil film between the reed and the valve seat. Finally, x, $\dot{x}$ and $\ddot{x}$ are the instantaneous reed lift, velocity and acceleration, respectively. The equivalent mass m_{eq} is determined from data for reed stiffness, k, and natural frequency, f_n. The flow induced force acting on the reed and the mass flow rate through the valve are obtained with reference to the effective force area A_{ef} and the effective flow area A_{ee}, respectively. From the pressure difference across the valve, Δp_v, A_{ef} is determined from $A_{ef} = F/\Delta p_v$. The effective force area can be understood as a parameter related to how efficiently the pressure difference Δp_v opens the valve. On the other hand, for the same pressure drop, A_{ee} expresses the ratio between the actual mass flow rate through the valve and that given by an isentropic flow condition.

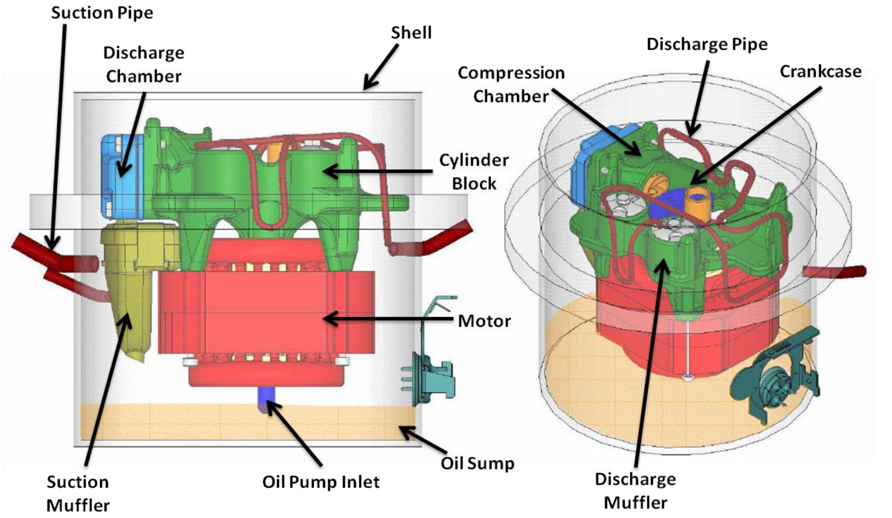

Figure 1. Three-dimensional view of a reciprocating compressor.

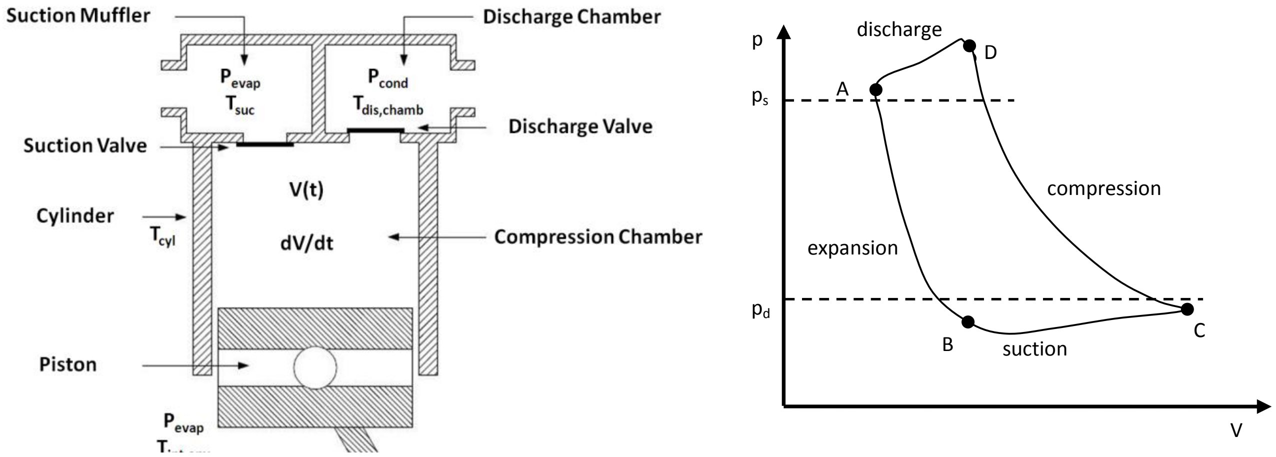

Figure 2. (a) Schematic of a reciprocating compressor and (b) its indicator diagram.

2.2. Heat Transfer

The evaporation and condensation pressures and superheating temperature at the evaporator outlet are a specified condition of the refrigeration system. However, temperatures at the cylinder surface, suction chamber, internal environment of the compressor shell and discharge chamber are a result of complex interactions between each component. Most simulation approaches apply an integral formulation for the energy conservation equation to some conveniently chosen components of the compressor, allowing them to interact with each other through a net of thermal connection. For instance, Todescat *et al.* (1992) developed a numerical model in which control volumes are linked through equivalent thermal conductances that are calibrated with reference to experimental data for the compressor thermal profile at a certain operating condition.

The use of thermocouples allows one to carry out energy balances in several of the compressor components, by evaluating the gas enthalpy variation between the entrance and exit sections, as depicted in Fig. 3. By doing this, it is possible to estimate the thermal energy, $\dot{Q}_w$, released or absorbed in each component of the compressor, as follows:

$$\dot{Q}_w = (\dot{m}h)_{in} - (\dot{m}h)_{ou} \quad (2)$$

Then, it is a straightforward task to characterize the heat transfer process at the component wall through the concept of an equivalent thermal conductance (*UA*):

$$\dot{Q}_w = UA \cdot (T_{gas} - T_{amb}) \quad (3)$$

Naturally, in Equation (2), the values of enthalpy at the entrance and exit sections of the component can be estimated from the temperature measurements at the same locations. Therefore, with temperature measurements also for the gas inside the component, T_{gas}, and for the gas in the surrounding ambient, T_{amb}, the equivalent conductance (*UA*) can be found from Eq. (3).

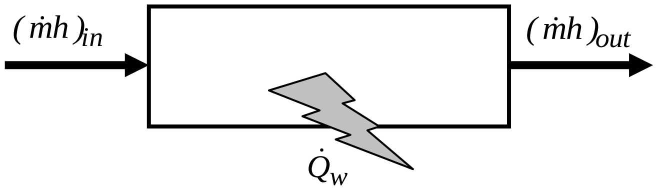

Figure 3: Schematic of thermal energy balance in a compressor component.

The enthalpies appearing in Eq. (2) are obtained from experimental measurements of temperature and pressure at the inlet and outlet of each control volume. With estimates of the equivalent thermal conductance for each control volume, it is possible to simulate the compressor, assuming that such conductances are intensive to the operating condition and working fluid, as observed experimentally by Dutra and Deschamps (2009) and Meyer and Thompson (1990).

The numerical model proposed by Todescat *et al.* (1992) is able to predict several complex phenomena inside the compressor. However, the model not flexible in the sense that it will not accurately predict the compressor thermal profile originated by variations in the compressor layout. In addition to that, corrections are also required for the conductances if the compressor operating condition is drastically different in comparison with the reference condition used to calibrate the model. In this respect, no study was found in the literature addressing the ability of the model in situations in which the compressor is changed, such as in the case of variable capacity compressors (VCC). The analysis of this aspect of the numerical model is the main purpose on the present investigation.

The compressor tested in this study was divided into seven components: shell, electric motor, suction muffler, mechanical assembly, discharge chamber, discharge muffler and discharge pipe. Figure 4 presents a schematic view of a hermetic refrigeration compressor with control volumes chosen for energy balances. The energy balance equations form a nonlinear system with one possible solution, which can be solved by an iterative procedure. For each control volume, there is an associated residual function given by Eq. (4), which is solved by the Newton-Raphson method for ordinary derivatives.

$$\emptyset_i(\overline{T}_i) = \left(\dot{W}_i + \sum \dot{m}_{in} h_{in}(\overline{T}_i) - \sum \dot{m}_{out} h_{out}(\overline{T}_i)\right) - UA_i(\overline{T}_i - \overline{T}_{ref}) \quad (4)$$

Once found the roots of the residual functions, the equations of energy conservation are satisfied and a new temperature field is obtained. This field serves as a boundary condition to simulate the compression process that in turn, updates values for heat transfer, mass flow rate, energy consumption and efficiency of the compressor. The loop formed by the coupling of these two programs is repeated until the module of the difference between the temperatures at each point obtained in the last two simulations is less than a convergence criterion, represented by Eq. (5).

$$\varepsilon_i > \left|\overline{T}_{i,n} - \overline{T}_{i,n-1}\right| \quad (5)$$

3. RESULTS AND DISCUSSIONS

The experimental data required to validate the simulation model was obtained in calorimeter facility. Measurements were provided for energy consumption, mass flow rate, valve displacement, as well as fluid temperature and pressure in different parts of the compressor. 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°C 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.

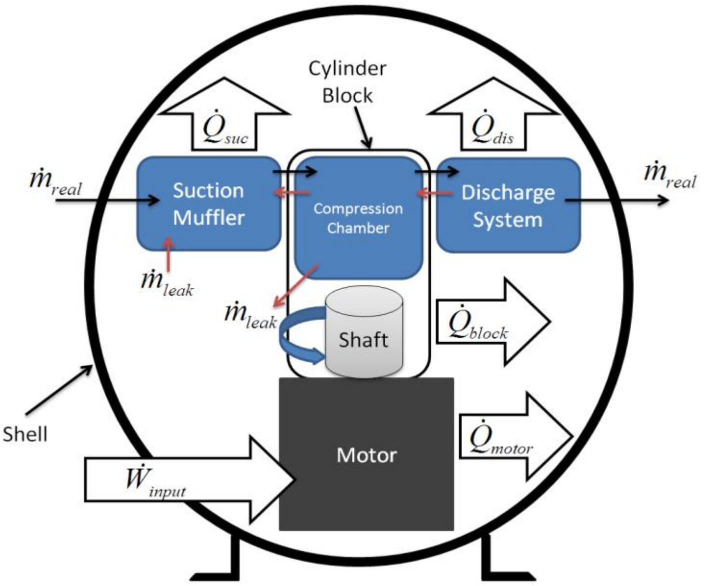

Figure 4. Schematic view of a hermetic refrigeration compressor with control volumes chosen for energy balances.

Experimental tests were conducted with an R600a variable speed reciprocating compressor with a displacement of 9.04cm^3. Two operating conditions were selected for simulations and are detailed in Table 1. For the first condition, the compressor was tested at 1200, 1400, 1600 and 2000rpm, while for the latter tests were conducted at 1400, 1600 and 3000rpm. The numerical model was calibrated at the operating condition 1 at 1400rpm.

Comparisons between numerical and experimental results depicted in Fig. 5 show good agreement for the speed range between 1200rpm and 2000rpm, with a maximum deviation of 1.1°C in the suction chamber temperature for both operating conditions, which is smaller than the experimental uncertainty. It is interesting to note that in this range of speed the temperature in the suction chamber does not vary significantly.

Table 1. Temperatures of the two operating conditions tested.

	Operating Cond. 1	**Operating Cond. 2**
Evaporation	-25°C	-23.3°C
Condensation	30°C	38°C
Superheating	25°C	32°C
Subcooling	25°C	32°C
External Environment	25°C	32°C

However, a discrepancy of 6°C in the suction chamber temperature was verified when the compressor operated at 3000rpm. In fact, whereas the experimental data indicates an increase in the suction temperature with the compressor speed the model shows the opposite trend. This deficiency is partly because the model adopts a constant thermal conductance for the suction system regardless the compressor speed. In fact, as one should expect, there is an enhancement of the heat transfer process in the suction system as the flow rate increases. In spite of this, as far as the cooling capacity is concerned, the maximum deviation between experimental and numerical results is around 5%. As can be seen in Fig. 6, the cooling capacity varies almost linearly with the compressor speed, indicating that within this speed range, the volumetric efficiency is practically constant.

The model presented herein is a good alternative for simulating the compressor performance under different operating conditions if the speed is kept constant. However, it has been shown that the hypothesis of constant thermal conductance fails if the compressor speed is far from that at which the model was calibrated. Although the error found for cooling capacity was much smaller than that related to the temperature in the suction chamber, it remains to be assessed the model capability for optimization of high efficiency compressors. In such analysis, the correct prediction of temperature at several locations inside the compressor is very important. Therefore, the model should be modified so as to be sensitive to variations in the operating speed, by expressing the thermal conductance as a function of mass flow rate.

Despite its shortcomings, the present model was applied to simulate the compressor performance under three speeds (1800, 3600 and 5400rpm). Results for efficiency losses due to superheating, suction and discharge systems are shown in Fig. 7. As expected, thermodynamic losses in the suction and discharge systems increase with the compressor speed, since the mass flow rate and, hence, viscous friction losses become more significant. On the other hand, a larger amount of heat is released when the compressor is set to operate at higher speeds, since more gas is compressed as a function of time, increasing the superheating in the suction system. In fact, it is expected that the temperatures of other components inside the compressor will also rise.

Finally, predictions for volumetric and isentropic efficiencies are shown in Fig. 8. It is interesting to note that isentropic efficiency increases when the speed is changed from 1800 to 3600rpm, but then decreases as the speed reaches 5400rpm. This is a consequence of much higher losses in the suction and discharge systems and, probably, also due to valve dynamics. When attention is directed to volumetric efficiency, a considerable decrease of performance is seen at the highest speed condition. Considering the losses of superheating and suction and discharge systems when the speed is changed from 3600 to 5400rpm, predictions suggest that this phenomenon is linked to losses in the suction and discharge systems.

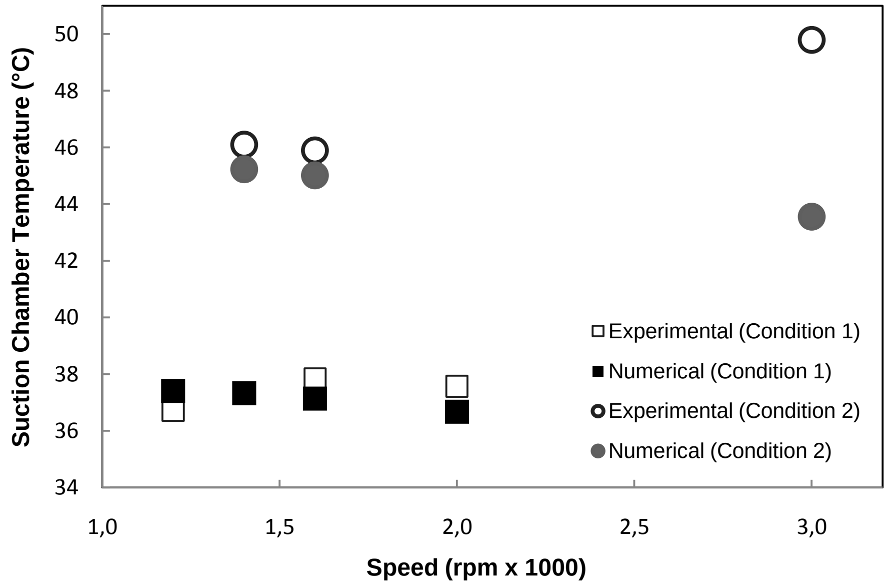

Figure 5. Numerical and experimental temperatures in the suction chamber.

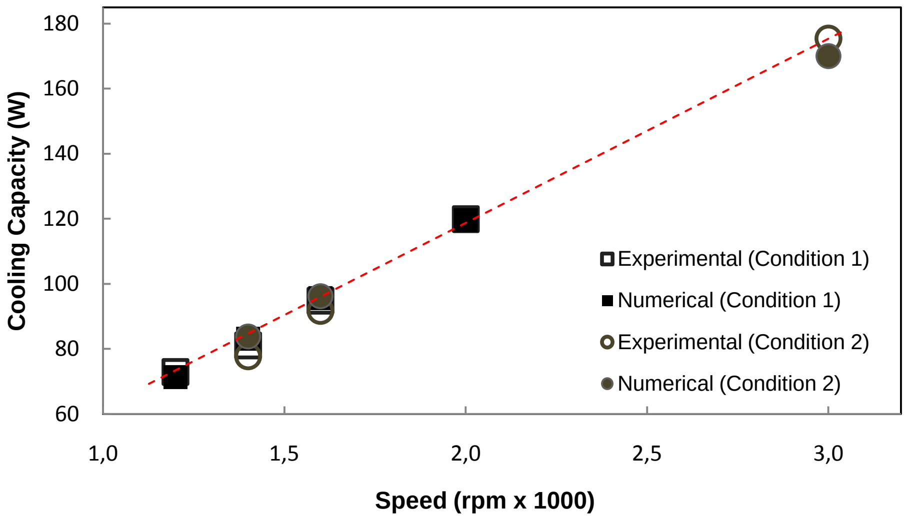

Figure 6. Numerical and experimental cooling capacity.

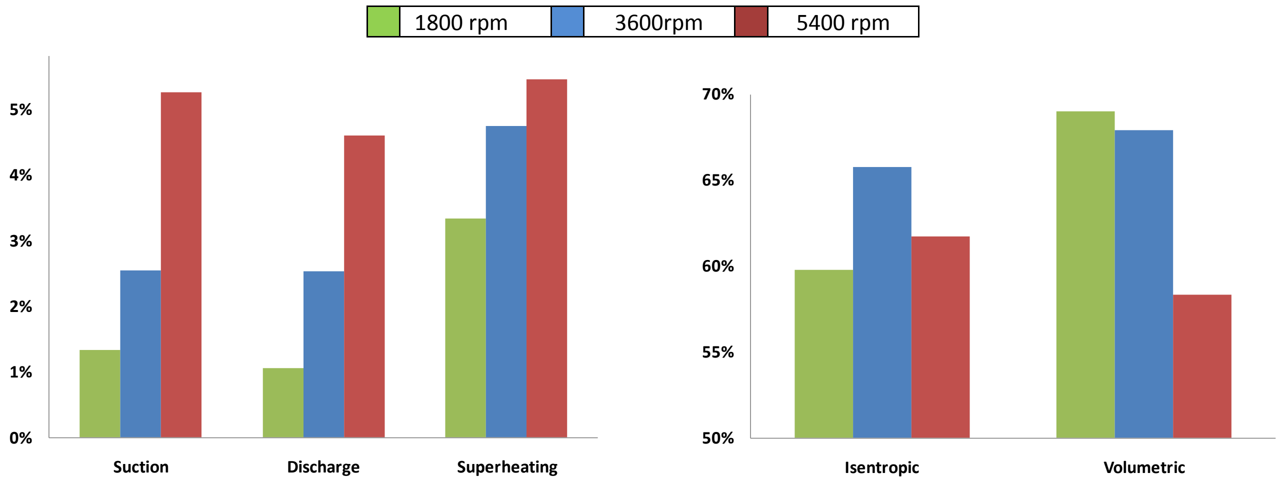

Figure 7. Efficiency losses associated with suction and discharge systems and superheating.

Figure 8. Isentropic and volumetric efficiencies as a function of compressor speed.

4. CONCLUSIONS

This paper considered the assessment of a numerical model to simulate the thermodynamic performance of variable capacity compressors (VCC), with especial attention to superheating. The compressor internal temperatures are supplied as input for the simulation program, with a routine that evaluates the temperature in eight control volumes through energy balances. The model was seen to provide results for temperature in the suction chamber and cooling capacity in good agreement with experimental data for different operating conditions. However, some discrepancy was observed when the compressor speed was considerably different from that in which the model was calibrated. A possible explanation for this failure is the fact the model does not take into account variations of thermal conductance originated by changes in the gas flow velocity as the compressor speed is increased.

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6. REFERENCES

Almbauer, R., Burgstaller, A., Abidin, Z., Nagy, D., 2006, "3-Dimensional Simulation for Obtaining the Heat Transfer Correlations of a Thermal Network Calculation for a Hermetic Reciprocating Compressor", Proceedings of the International Compressor Engineering Conference at Purdue, USA, pp. C079.

Annand, W.J.D., 1963, "Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines", Proceedings of the Instn. Mech. Engrs., Volume 177, Nr. 36, pp. 973-996.

Deschamps, C. J., Possamai, F. C., Pereira, E. L. L., 2002, "Numerical Simulation of Pulsating Flow in Suction Mufflers", Proceedings of the International Compressor Engineering Conference at Purdue, USA, pp. C11-4.

Dhar M and Soedel W, 1978. Influence of a Valve Stop and/or Suction Muffler on Suction Valve Noise of an Air Compressor, Proc. Int. Compressor Engng. Conf. at Purdue, pp. 36-44.

Dutra, T., Deschamps, C.J., 2009, "Combined Experimental and Numerical Characterization of Heat Transfer in Hermetic Reciprocating Compressors", Proceedings of the 7th World Conference on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, Krakow, Poland.

Meyer, W.A., Thompson, H.D., 1990, "An Analytical Model of Heat Transfer to the Suction Gas in a Low-Side Hermetic Refrigeration Compressor", Proceedings of the International Compressor Engineering Conference at Purdue, USA, pp. 898-907.

Ooi, K.T., 2003, "Heat Transfer Study of a Hermetic Refrigeration Compressor", Applied Thermal Engineering, Vol.23, pp. 1931-1945.

Ribas Jr., F.A., 2007, "Thermal Analysis of Reciprocating Compressors", Proceedings of the International Conference on Compressors and Their Systems, London, England, pp. 277-287.

Ribas Jr., F.A., Deschamps, C.J., Fagotti, F., Morriesen, A., Dutra, T., 2008, "Thermal Analysis of Reciprocating Compressors - A Critical Review", Proceedings of the International Compressor Engineering Conference at Purdue, USA, Paper 1306.

Schreiner, J.E., Ribas Jr., F.A., Deschamps C.J., Barbosa Jr., J.B., da Rosa, V.H.P., 2009, "Thermal Management of a Commercial Reciprocating Compressor Through Numerical Simulation", Proceedings of the International Conference on Compressors and Coolants Compressors", Rovinka, Slovak Republic.

Todescat, M.L., Fagotti, F., Prata, A.T., Ferreira, R.T.S., 1992, "Thermal Energy Analysis in Reciprocating Hermetic Compressors", Proceedings of the International Compressor Engineering Conference at Purdue, USA, pp. 1419-1428.

Ussik, M. S., 1984, "Numerical Simulation of Hermetic Reciprocating Compressors", Msc. Thesis (in Portuguese), Federal University of Santa Catarina.

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