NUMERICAL PREDICTION OF SUPERHEATING IN THE SUCTION MUFFLER OF A HERMETIC RECIPROCATING COMPRESSOR

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Abstract. *The efficiency of reciprocating compressors is quite affected by gas superheating in the suction system. This paper reports the numerical prediction of fluid flow through the suction muffler of a small hermetic reciprocating compressor by using a three-dimensional formulation. The integration of the conservation equations for the fluid refrigerant (continuity, momentum and energy) is carried out via the finite volume methodology. A hermetic reciprocating compressor was instrumented with several sensors to measure temperature and heat flux at different positions of the muffler wall. Moreover, a hotwire anemometer was installed in the muffler suction chamber for measurements of temperature and velocity transients. The simulation model validation was carried out through comparisons between numerical with experimental results for heat flux, temperature and velocity.*

Keywords*: Numerical heat transfer; Finite volume methodology; Compressor suction systems.*

1. INTRODUCTION

The demand for highly efficient refrigeration systems has been increasing quite steadily due to their impact on the energetic matrix and also because of environmental issues. In the United States, for instance, about 8% of the residential electrical energy consumption is due to refrigerators and freezers (Pereira *et al.*, 2008).

In a vapor-compression refrigeration system, the compressor plays a key role on the power consumption. The overall compressor performance is a result of electrical, mechanical and thermodynamic efficiencies. According to Ribas *et al.* (2008), the thermodynamic efficiency is much lower than the other two contributions, with values varying between 80 and 83% for household reciprocating compressors. The thermodynamic inefficiency can essentially be attributed to irreversibilities in the suction, discharge and compression processes. Ribas *et al.* (2008) also reported that gas superheating may account for up to half of the thermodynamic losses in a small reciprocating compressor. Nevertheless, while many efforts have been directed to improve the suction and discharge systems, mainly by decreasing flow restrictions, superheating losses have received much less attention along the years.

An excellent review of research on compressor thermal analysis up to 1998 has been presented by Shiva-Prasad (1998), including an account of main developments in theoretical, numerical and experimental methodologies. The review suggests that much effort is still required to properly understand heat transfer phenomena in compressors and to develop numerical models and experimental methodologies for compressor thermal design. It was also concluded that heat transfer in reciprocating compressors was perceived by many as a technological issue in which new materials are sought for improving the compressor reliability rather than its energy efficiency.

Superheating is a consequence of heat transfer between the refrigerant fluid and hot components of the compressor. This phenomenon affects not only the thermodynamic efficiency but also the compressor volumetric efficiency, since the gas density inside the cylinder is directly related to the gas temperature. The first source of superheating occurs in the crankcase, where the gas comes into contact with the electrical motor, the discharge system and the cylinder block. Although the heat transfer coefficient in the crankcase is relatively low, the temperature difference between the gas and the components is significant. Once inside the suction muffler, heat transfer takes place more intensively between the gas and the muffler walls because of significant velocity levels of the gas flow. Finally, during the suction process, refrigerant enters the compression chamber and is heated by the cylinder hot walls. Due to the quite high gas velocity at the exit of the valve and large temperature difference between the gas and the cylinder wall, heat transfer is greatly enhanced.

Some investigations have been devoted to the mapping of temperature and heat transfer in hermetic compressors. The use of thermocouples is the most traditional and consolidated experimental technique for compressor thermal analysis, but other experimental methodologies have been also applied, such as heat flux sensors (HFS's) for local heat flux and hotwire anemometry (HWA) for instantaneous velocity and temperature measurements.

HFS's have been increasingly employed in different applications, such as agrometeorology (Silberstein *et al.*, 2001; Borges *et al.*, 2008), material processing (Sabau and Wu, 2007), thermal comfort (Kurazumi *et. al*, 2008), building construction (Marinoski *et al.*, 2006), refrigeration (Silva, 1998; Seidel, 2001), and so on.

Recently, Dutra and Deschamps (2009a) reported an experimental heat transfer analysis of a small reciprocating compressor using HFS's. On the other hand, instantaneous temperature measurements were made by Shiva Prasad (1992) within a two-stage double acting 900 rpm compressor using a specially designed thermocouple. Bauer *et al.* (1998) carried out experiments in inlet manifolds of an IC engine running at 2750 rpm. In that work, they used thermocouples, heat flux sensors, as well as hot and cold wire sensors. The authors concluded that the period in which the flow is stagnated strongly contributes on the total heat transfer to the fluid in the inlet manifold.

Other relevant work concerning measurement of property transients was developed by Olczyk (2008). The paper regards to an experimental investigation of flow through inlet and discharge manifolds of diesel engines employing thermocouples and cold wire sensors. The author identified that the main difficulties in measuring temperature transient are related to the sensor time response and the signal interpretation. More recently, Morriesen and Deschamps (2009) measured instantaneous temperature and velocity inside of refrigerating compressor running at 3600 rpm, and analyzed the main phenomena associated with the flow during the suction cycle.

Concerning numerical methodologies, the proposals in the literature can be divided into three different classes. Integral numerical models (Todescat *et al.*, 1992; Ooi, 2003) apply the energy conservation equation over some conveniently chosen components of the compressor, which are then linked to each other. With the exception of very simple geometries, for which classical heat transfer correlations are available, experimental data is usually necessary to complement this type of analysis. Another difficulty of integral approaches is that complex interaction between the components brought about by conduction heat transfer cannot be accounted for in the thermal energy balances. Hybrid numerical models were developed to solve this issue, by combining an integral approach to evaluate the gas thermodynamics and a differential model to solve the three-dimensional heat transfer in solid components (Almbauer *et al.*, 2006; Ribas, 2007). Nevertheless, experimental data are still needed for the purpose of calibration in the hybrid methodology. Finally, full differential numerical models simultaneously solve heat transfer in the fluid and solid domains, eliminating the necessity of empirical data, such as convection heat transfer coefficients. Naturally, this alternative of modeling offers a greater flexibility in the analysis of changes in compressor layout. However, the computational processing time is quite high, making it difficult to be used for optimization purposes.

A considerable number of studies dealing with thermal management of hermetic compressors are available in the literature, including the analysis of superheating in suction mufflers. Most of such investigations adopt global heat transfer coefficients to evaluate the gas temperature variation in the muffler. Although the gas temperature in the suction chamber is enough to estimate the compressor performance, information about main sources of heating is crucial for developing new designs.

The present paper considers the numerical prediction of heat transfer in the suction muffler of a small reciprocating compressor, based on a three-dimensional differential formulation. Measurements for instantaneous temperature and velocity in the suction chamber and wall heat flux are also carried out with the purpose of providing data required to validate the simulation methodology. The main goal of this paper is to verify the current modeling capability to predict superheating in compressors.

2. EXPERIMENTAL SETUP

The experimental setup was planned with two objectives: i) measurements of temperature and heat flux at the suction muffler wall and ii) measurements of gas temperature and velocity in the suction chamber. Both measurements were carried out for a 3600 rpm reciprocating compressor, with a cooling capacity of 270 W, operating with R134a. A calorimeter facility was employed to submit the compressor to the required operating condition, which is represented by a pair of evaporation and condensation temperature, -23.3°C and 54.4°C, respectively. The experimental procedure is described below.

2.1. Measurement of heat flux and temperature at solid walls

Thermocouples were adopted to measure temperature at four positions of the muffler wall, identified by points 3, 4, 5 and 6 in Fig. 1, and whose values are presented in Tab. 1. The test was repeated several times so that each temperature value represents in fact an average.

Heat transfer at the walls was measured with heat flux sensors (HFS's). Such thermal sensors supply a self-generated output voltage (E) proportional to a heat flux excitation ($q'' = E/S$), where q'' is heat flux [W/m²] and S is the HFS sensitivity [V/W/m²]. The HFS sensitivity is commonly obtained through calibration procedures (Holmberg and Womeldorf, 1999). In the present work, the commercial HFS's employed were commercially supplied with their sensitivity values. According to the manufacturer, the uncertainty associated with the sensitivity is about ±5%. More details about HFS's working principle and the instrumentation issues can be found in Dutra and Deschamps (2009b).

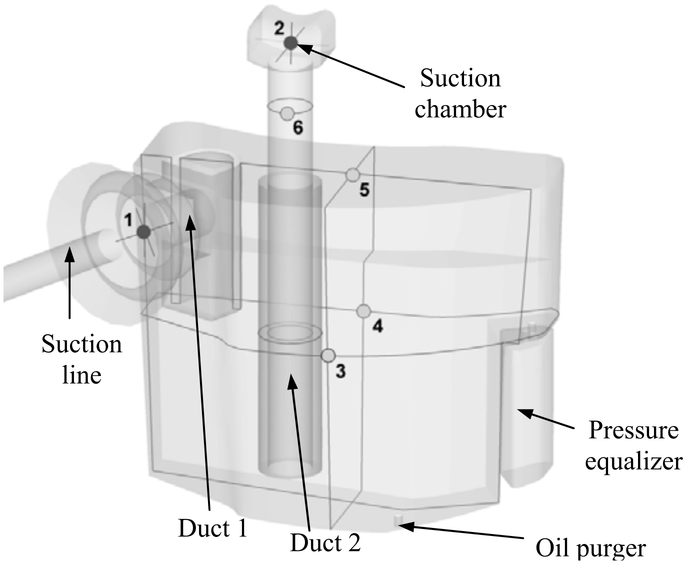

Figure 1. Suction muffler geometry and measurement positions.

Table 1. Experimental data for temperature at different positions of the muffler surface.

Muffler wall temperature	
(3) Front wall	57.0 ºC
(4) Back wall	54.0 ºC
(5) Top wall	64.9 ºC
(6) Neck wall	78.1ºC

2.2. Measurement of velocity and temperature transients

Measurements of gas temperature transients in the suction chamber were conducted with a DANTEC anemometer system composed by a constant current module (90C20). For velocity measurements, a DANTEC mini-CTA system (54T30) was adopted. A single 5 μm diameter tungsten wire sensor (55P11) was employed for both velocity and temperature measurements. Additionally, a piezoelectric pressure transducer was installed in the suction chamber and a sensing winding was installed in the crankshaft to measure the crank angle.

As illustrated in Fig, 2, the anemometry sensor was assembled at the entrance of the suction chamber, allowing measurements of velocity and temperature. During each experiment, 1000 measurements of temperature, velocity and pressure in the suction chamber were collected for every 240 operation cycles of the compressor. Measurements of velocity at the entrance of the suction chamber were used to allow a correction in the experimental data for temperature, associated with the sensor thermal inertia.

3. NUMERICAL METHODOLOGY

Differential governing equations for mass, momentum and energy conservation were applied to describe the gas flow inside mufflers. The compressible turbulent flow that prevails in the suction muffler was solved through the concept of Reynolds-averaged quantities, in which the value of a computed variable represents a time/ensemble average over a sufficiently long period of time. The turbulence transport contribution was modeled through the concept of an eddy viscosity model evaluated with the SST turbulence model. The choice of this turbulence model was based on the ground that it has been extensively used and validated in many practical problems of engineering, especially those in the presence of heat transfer. An equation of state for an ideal gas completes the system required to solve the compressible flow.

The system of equations was solved with the commercial CFD code Fluent (ANSYS, 2008), which is based on the finite volume methodology. A first-order upwind scheme was adopted to interpolate flow quantities needed at the control volume faces. The momentum and continuity equations were solved via a fully implicit coupled multigrid algorithm.

The geometry complexity of an actual suction muffler demanded the adoption of unstructured grid (Fig. 3). The geometric model was first generated with CAD software and then imported into a pre-processor available in the Fluent code to generate the computational grid. Tetrahedral elements were used to discretize the solution domain, keeping cell skewness bellow recommended upper limit values. The final mesh was formed by approximately 140,000 volumes.

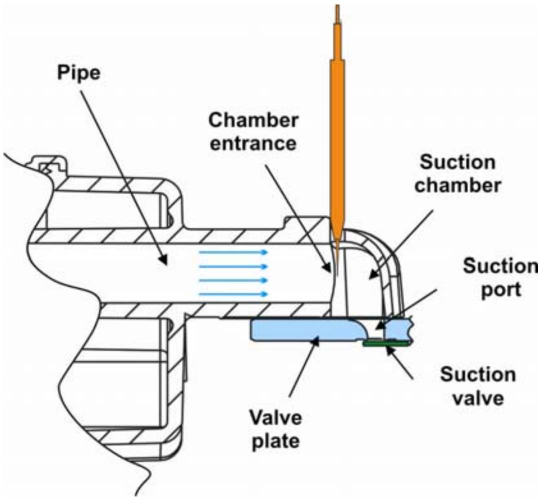

Figure 2. Detail of anemometry sensor assembly.

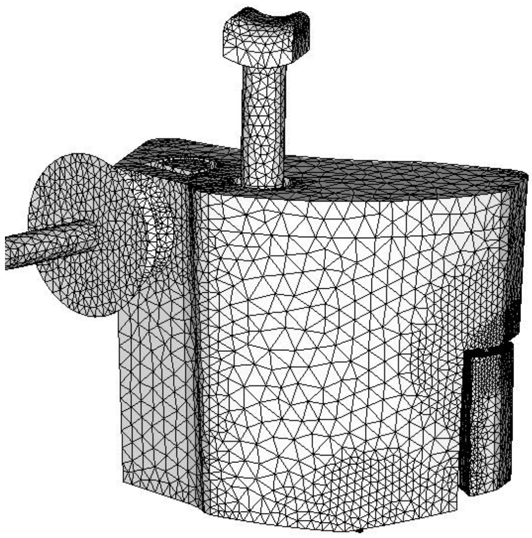

Figure 3. Computational grid for the three-dimensional formulation.

A turbulence intensity of 3% and turbulence length scale corresponding to the tube hydraulic diameter were adopted as the inlet boundary condition for the turbulent flow in the mufflers. Evaporation pressure was imposed as the inlet conditions for the gas in the suction line, oil purger and pressure equalizer.

The inlet gas temperature in the suction line was prescribed with reference to experimental data. This was a necessary procedure since there is no information about how much gas from the crankcase enters the suction muffler. Temperature distribution at the external surface of the muffler was prescribed according to experimental data at the four positions, shown in Tab. 1. A simple one-dimensional conduction equation was applied for an estimate of heat transfer through the muffler walls. Accordingly, the temperature at the internal wall surface in contact with the gas is calculated from the concept of a thermal resistance, $\Delta x/k$, where Δx is the wall thickness and k is the wall thermal conductivity.

Mass flow rate was prescribed as boundary condition in the suction chamber outlet. This information is obtained from a companion computational program especially developed for the thermodynamics compressor simulation. The companion program and the CFD suction muffler model run in a coupled manner. The solution domain flow field was initialized by considering the gas properties equal to the conditions at the suction line inlet. A total of three cycles were simulated to guarantee a fully periodic flow condition.

4. RESULTS AND DISCUSSION

Initially, the three-dimensional model was employed to analyze unsteady effects on the heat transfer and also on the flow pattern throughout the muffler. Since the heat transfer phenomena in the solid and fluid domains have quite different time scales, the solid domain of duct 2 was not considered in the transient simulation and, instead, the wall was admitted to be adiabatic.

Numerical results for velocity and temperature are shown in Figs. 4 and 5, respectively, at the same position where the wire sensor was located in the experimental setup. As can be seen, predictions for instantaneous velocity are in good agreement with the experimental data. It is also observed that the experimental data indicates a maximum velocity of 40m/s at the crankcase angle of 264°, whereas the simulation model predicts 34m/s at angle 262°. There is a second peak of velocity at angle 310°, with a certain phase delay between experimental and numerical results. Additionally, during the period in which the valve is closed, it becomes evident from measurements and predictions some small velocity fluctuations in the suction chamber.

Figure 5 depicts the instantaneous temperature in the suction chamber along a complete compressor operation cycle. Despite a clear difference between levels of temperature measured and predicted, it can be noticed a good agreement between both results as far as the temperature oscillatory behavior is concerned. It is interesting to observe the presence of a significant temperature increase during the period in which the suction valve is closed, which is a consequence of heat transfer between the suction chamber wall and the gas. When the valve opens, there is a sudden decrease of temperature associated with pressure drop in the suction chamber.

Table 2 shows a difference of 23-28% between experimental and numerical results for heat flux at the muffler walls, which reflects the difficulties associated with the measurements (Dutra and Deschamps, 2009b). On the other hand, heat transfer is a phenomenon more difficult to predict than fluid flow and, therefore, a further analysis of simulation parameters, such as turbulence model and grid refinement in the vicinity of walls, will be considered in future studies. It should be mentioned that no relevant fluctuations are observed on the heat flux along the cycle, with standard deviations of 3% and 6% for the forward and backward regions, respectively. All numerical results for heat flux were evaluated as an area weighted average, with dimensions similar to the area in which HFS's were assembled on the muffler surface.

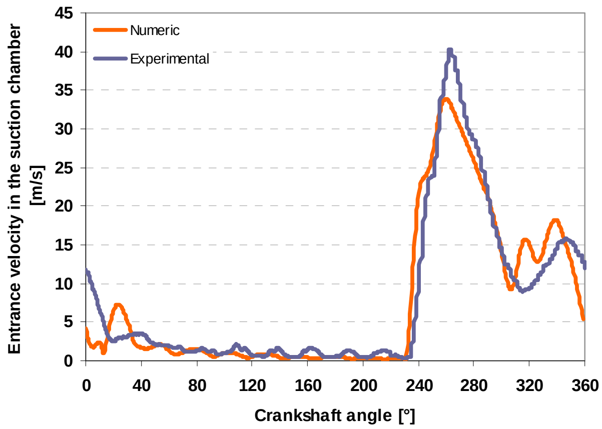

Figure 4. Instantaneous velocity in the suction chamber.

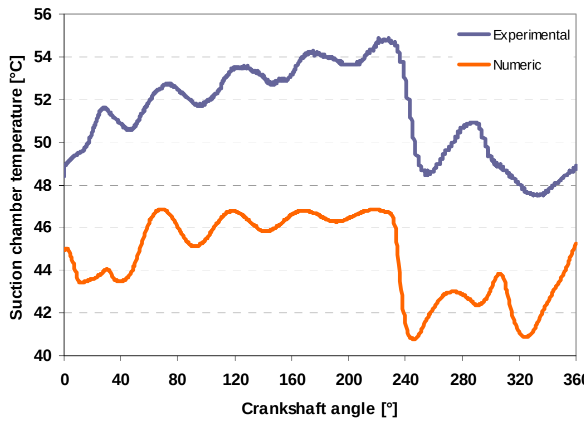

Figure 5. Instantaneous temperature in the suction chamber.

Table 2. Experimental and numerical results for heat flux at the muffler wall.

Position	**Experimental (W/m²)**	**Numerical (W/m²)**	**Deviation (%)** (related to experimental)
Forward	457	563	23
Backward	369	473	28

5. CONCLUSIONS

The present paper reported a numerical simulation, experimentally validated, of flow through the suction muffler of a small reciprocating compressor, based on the finite volume methodology. Heat flux sensors, cold wire and hot wire sensors were adopted to experimentally characterize thermal effects in the suction muffler and to validate the numerical solution. Predictions for velocity and temperature transients in the suction chamber and heat flux at the muffler wall were obtained and showed to be in good agreement with experimental data. The only exception is the much lower temperature levels found in the simulation, probably associated with an issue of boundary condition that must be addressed in a future investigation. Despite of that, temperature oscillations were shown to be in close agreement, providing further evidence that the fluid dynamics is correctly described by the simulation model.

6. ACKNOWLEDGMENTS

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