MEASUREMENT OF HEAT TRANSFER IN A HERMETIC RECIPROCATING COMPRESSOR

Thiago Dutra, dutra@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

Abstract. *In reciprocating compressors adopted for household refrigeration, the compression process generates a considerable amount of heat inside the compressor, increasing suction superheating and reducing the compressor efficiency. Several attempts have been made to provide a better account of compressor superheating, but such thermal analysis is very difficult due to the complex geometry, which does not allow simple modeling approaches. Moreover, the lubricating oil inside the compressor also takes part in the heat transfer process. The present paper reports temperature and heat flux measurements in several regions inside the compressor. A small reciprocating compressor operating with R134a was selected for the analysis, with operating conditions represented by two pairs of evaporation and condensation temperatures (-23.3°C/40.5°C; -10.0°C/90.0°C). The study also provides a clear evidence of the role played by the lubricating oil on the compressor heat rejection and the influence of the suction system on the heat transfer in the electric motor.*

Keywords*: heat flux measurement, superheating, reciprocating compressor*

1. INTRODUCTION

The overall efficiency of a compressor can be understood as being the result of three aspects: i) electrical efficiency, associated with the driving motor and its start up auxiliary device; ii) mechanical efficiency, related to the bearing system; iii) thermodynamic efficiency, due to irreversibilities in the suction, compression and discharge processes. A simple analysis of current efficiency levels of state-of-art household reciprocating compressors indicates an electric efficiency around 88%. The use of synchronous motors could increase even more this efficiency but, however, such alternatives sometimes are not adopted because of cost issues. Concerning the mechanical system, the efficiency levels are also quite high and can reach levels of up to 92%. It should be mentioned that linear compressors and variable speed compressors running at lower speeds can offer even higher mechanical efficiencies.

The thermodynamic efficiency is much lower and usually between 80 and 83%. Therefore, it is clear that future improvements in the compressor efficiency will very likely be associated with the reduction of thermodynamic losses. Ribas Jr. *et al.* (2008) presented the sources of thermodynamic inefficiency in a high-efficiency compressor with a capacity of 900 BTU/h (ASHRAE / LBP) operating with R134a. They showed that approximately 47% of the thermodynamic losses is originated in the suction and discharge processes, due to viscous losses in the vapor path from the suction line to the cylinder and from the cylinder to the discharge line, respectively. Much effort has been directed to reduce the energy losses in the suction and discharge systems, mainly by decreasing flow restrictions. On the other hand, the compressor performance is quite affected by its thermal profile due to the suction gas superheating and its contribution to the overall compressor thermodynamic loss is 49%. Superheating is associated with heat generated and released inside the compressor, such as in the compression process and by 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 can 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. Furthermore, in addition of being a lubricating agent for bearings the oil inside the compressor also acts to directly transfer heat from the compressor kit to the shell. As shown in Fig. 1, in the reciprocating compressor selected for the present analysis, the lubricating oil, stored in the sump, is collected by an oil pump, which is coupled to the crankshaft. As the shaft spins, the lubricating oil flows inside the pump by centrifugal action until it reaches the other extremity, leaving the crankshaft as a jet. The oil jet impinges against the shell upper surface, referred to as cover in this paper. Then, a portion of oil returns to the sump by flowing over the kit and other portion flows as an oil film along the shell sidewall.

A detailed thermal analysis of the suction system can lead to a considerable improvement in its performance, mainly through the reduction of gas superheating. However, a detailed knowledge about thermal interaction among the compressor components is required before one can propose alternatives to improve the compressor performance. An excellent review of research on this subject up to 1998 has been presented by Shiva Prasad (1998), including an account about major developments of theoretical, numerical and experimental methodologies. The author pointed out that much progress was still required to understand the heat transfer phenomenon in compressors, as well as to develop numerical models and experimental methodologies to accommodate this aspect in the compressor design. He concluded that heat transfer in reciprocating compressors is perceived as a technology issue aimed at developing new materials for improving the compressor reliability rather than its energy efficiency.

A great number of papers in literature are concerned with numerical models to predict heat transfer in reciprocating compressors. Some of such models (Todescat *et al.*, 1992; Ooi, 2003) are based on integral formulations, in which the compressor domain is divided into control volumes. By applying mass and energy balances to each one of the control volumes, the thermal profile of the compressor is obtained. Other studies are based on differential formulation for the entire domain, including heat transfer in the compressor components and in the gas, but with a high computational cost (Raja *et al.*, 2003). Finally, a third group of methodologies (Almbauer *et al.*, 2006; Ribas Jr., 2007) combine integral and differential formulation to form the so called hybrid model. Recently, Pizarro-Recabarren (2007) developed a numerical model to evaluate the influence of the lubricating oil on the compressor thermal profile.

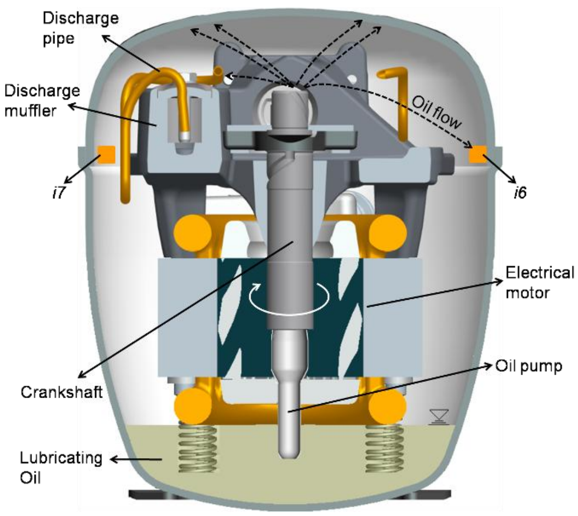


Figure 1. Schematic 3D view of a typical reciprocating compressor.

Heat transfer in reciprocating compressors has also been reported through experimental works. Meyer and Thompson (1990) and (Kim *et al.*, 2000) adopted thermocouples to measure the gas temperature in different components inside the compressor. Then, by applying energy balances, the heat transfer process through the component wall is characterized by the concept of an equivalent thermal conductance (*UA*). Shiva Prasad (1992) carried out measurements of instantaneous temperature and heat flux at several parts of the compression chamber of a 900 rpm reciprocating air compressor. The major aim of the research conducted by Shiva Prasad (1992) was to verify the influence of the regenerative heat transfer on the compressor capacity. Dutra and Deschamps (2009) developed a combined experimental and numerical study to characterize heat transfer at the shell internal and external surfaces of a hermetic reciprocating compressor adopted for household refrigeration, under actual operating conditions. Measurements of heat flux were carried out in several regions of the shell surface and validated through energy balances and numerical simulation. The results indicated the lubricating oil is an important agent in the heat transfer process.

The present paper reports an experimental investigation aimed at characterizing heat transfer on the electric motor and on the external surface of the compression chamber of a household reciprocating compressor, under two operating conditions. Additionally, the effect of the lubricating oil on the heat transfer on the compressor shell is also assessed.

2. EXPERIMENTAL PROCEDURE

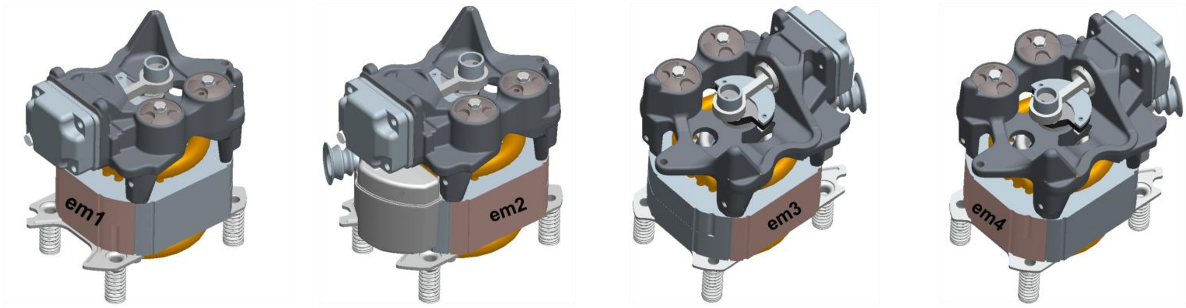
The working principle of a HFS consists on a self-generated voltage output proportional to a heat flux excitation. Such a voltage signal (E) derives from a temperature difference (ΔT) across the HFS thickness, which is captured by a thermocouple serial association ($E = N\, S_T\, \Delta T$), being N and S_T the number of thermocouple junctions and the thermoelectric sensitivity of the materials, respectively. On the other hand, the one-dimensional steady-state heat flux perpendicular to the HFS surfaces, q'', depends on the HFS thermal conductivity, k, on the HFS thickness, t, and on the temperature difference between its surfaces ($q'' = k\, \Delta T / t$). Then, by combining the aforementioned equations, one obtains a linear relation between heat flux and voltage output ($q'' = E / S$), where S ($= N\, S_T\, t / k$) is the sensitivity associated to a HFS. The value of S, which is provided by the sensor manufacturer with an accuracy of 5%, can also be obtained from a calibration procedure. In the present work, some HFS's were calibrated and, because no difference higher than 10% was observed in relation to the values of S specified by the manufacturer, the supplied values were considered in the measurements.

The surfaces of the electric motor and compression chamber were divided into four regions, as shown in Figs. 2 and 3, and in each region a HFS was assembled. Great part of the HFSs employed in the present study was provided with a thermocouple to measure its surface temperature. The instrumentation of HFSs on the compressor internal components is not trivial. Due to the high temperature levels and the lubricating oil flow, sensors must be well fixed. In this respect, an epoxy-adhesive was necessary to attach the HFS to the surfaces and their wires carefully positioned inside the compressor, so as to minimize changes in the lubricating oil flow path, as illustrated in Fig. 4. This is a quite relevant aspect, because the lubricating oil sensibly affects the heat transfer process inside the compressor and amplifies the heat deliver to the external ambient. The flow of oil in the compressor is promoted by a pump, which collects oil stored in the sump and, by centrifugal action, takes it to the upper parts of the compressor shell (Fig. 1).

Several thermocouples were also instrumented in the compressor internal ambient, in order to establish a suitable reference temperature, T_∞, to estimate the local heat transfer coefficient, h_c, in each region, i.e.:

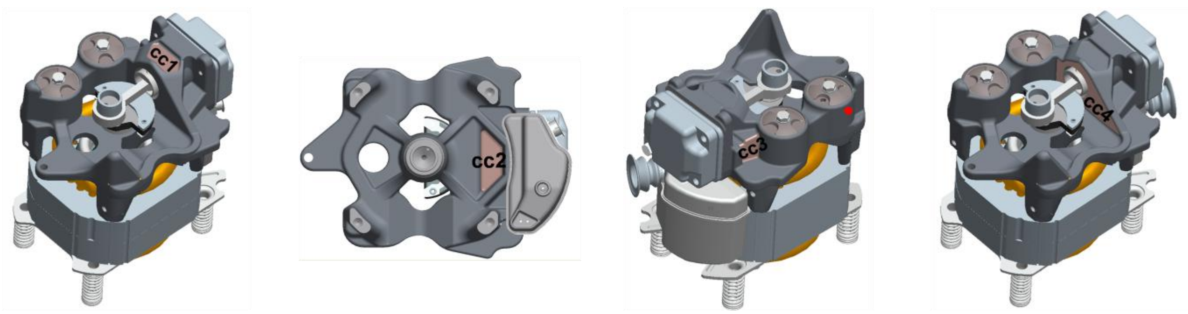
$$h_C = \frac{q''}{T_S - T_\infty} \tag{1}$$

where T_S is the sensor surface temperature. It should be mentioned that in most regions inside the compressor there is a thin film of oil covering the surface (Fig. 4). Therefore, since the reference temperature, T_∞, is measured in the gas, the heat transfer coefficient h should concern to a global heat transfer coefficient indeed. For the shell external surface there is no such an issue and the heat transfer coefficient is referenced to the external air temperature.



em1 ***em2*** ***em3*** ***em4***

Figure 2. Measurement regions on the electric motor.



cc1 **cc2** **cc3** **cc4**

Figure 3. Measurement regions on the compression chamber.

3. RESULTS

A 60Hz small reciprocating compressor operating with R134a was selected for the analysis, being submitted to two operating conditions, represented by two pairs of evaporation and condensation temperatures: (-23.3ºC/40.5ºC) and (-10.0ºC/90.0ºC). A calorimeter facility, built according to parameters established by standards, was employed to test the compressor. The uncertainties associated with measurements taken with the calorimeter are lower than $\pm$2% for mass flow rate and power consumption. Additional details of the experimental facility are described in (Dutra, 2008).

The compressor was tested five times for each operating condition. Results for heat flux, temperature and heat transfer coefficient are presented with an uncertainty bar corresponding to a 95% confidence interval.

3.1. Validation of experimental results

The shell internal surface of the compressor shell was divided into eleven regions, whereas ten regions were adopted for the external surface, and in each region a HFS was assembled. The heat transfer rate rejected through the compressor shell, $\dot{Q}_c$, can be calculated by summing up contributions of local heat flux measured in each of the regions in the internal and external surfaces. Thus,

$$\dot{Q}_c = \sum_{i=1}^{n} q''_i \, A_i \tag{2}$$

where q''_i is the local heat flux on the *i*-nth region of the shell surface with an area equal to A_i. Values of A_i needed in equation (2) are obtained from a CAD model. On the other hand, the heat rejected through the shell, $\dot{Q}_c$, can also be estimated from an energy balance applied to the compressor itself:

$$\dot{W}_c = \dot{m}\left(h_d - h_s\right) + \dot{Q}_c \tag{3}$$

where $\dot{W}_c$ is the compressor power consumption, $\dot{m}$ is the refrigerant mass flow rate and h_s and h_d are the refrigerant specific enthalpies at the suction and discharge lines, respectively. The power consumption is measured with a power meter and the specific enthalpies are determined from measurements of temperature and pressure at the corresponding suction or discharge line. According to Fig. 5, there is reasonable agreement between results for $\dot{Q}_c$ obtained from measurements, equation (2), and from the energy balance, equation (3). The greater deviation is regarded for the value measured at external surface, which is approximately 15% higher than that given by the energy balance. Despite some differences observed between local heat fluxes on the external and internal surfaces of the shell, one notices that the heat flux levels on external surface are also increased when condition -23.3ºC/40.5ºC is changed to -10.0ºC/90.0ºC.

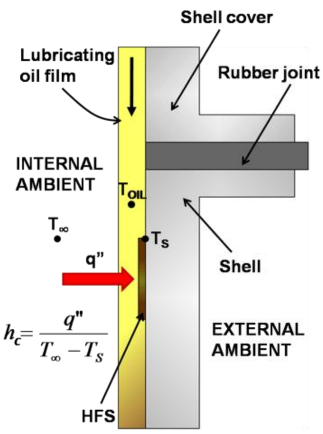


Figure 4. Lubricating oil film on the internal surface of the compressor shell.

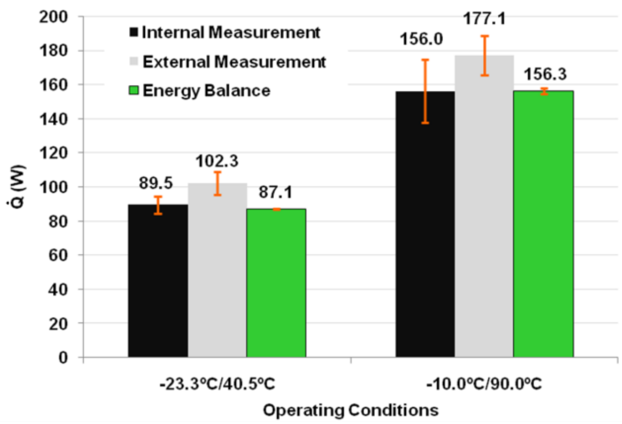
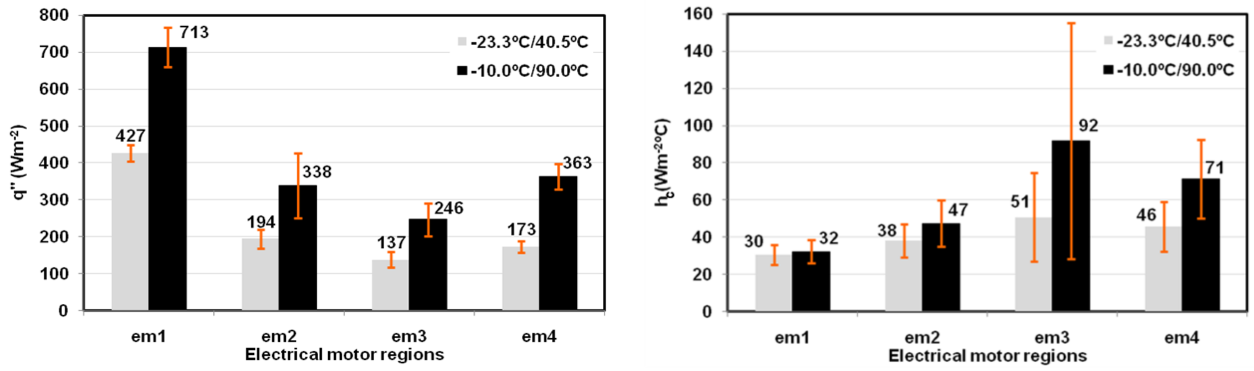


Figure 5. Heat transfer rate at the compressor shell for different operation conditions.

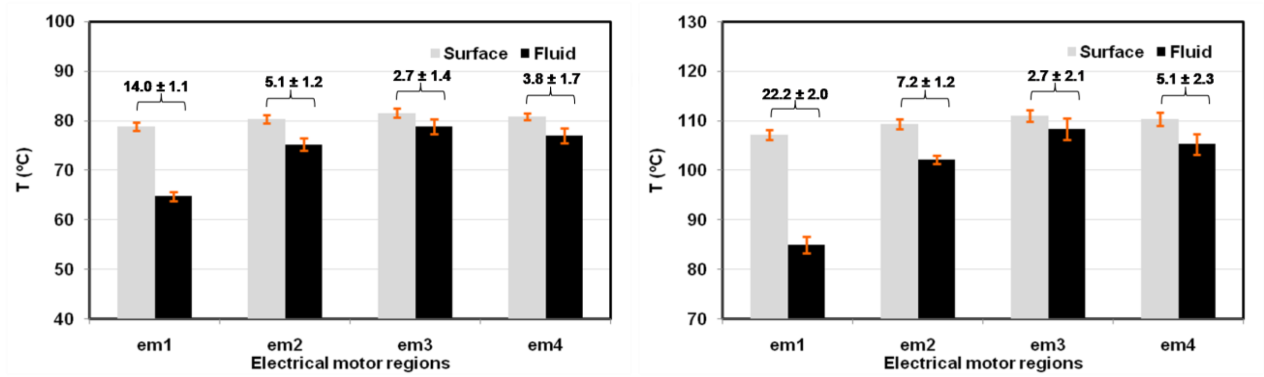
3.2. Heat transfer at the electric motor and compression chamber surfaces

Regarding the electric motor, Figs. 6 and 7 indicate that region *em1* presents the higher heat flux level and the lower surface temperature among the four surfaces analyzed. This can be explained by the proximity of this region to the suction muffler, where the lowest temperature is found inside the compressor. The remaining surfaces of the electric motor present nearly uniform heat flux and temperature profile. Figure 7(b) shows the local heat transfer coefficients which are roughly the same for the four regions. A high level of measurement uncertainties are seen for the heat transfer coefficient, mainly for region *em3*. This is caused by the low temperature difference between the solid surface and the refrigerant gas in the internal ambient, which has the same magnitude of the temperature sensor uncertainty (approximately 2°C). In the same way as shown for the compressor shell, when the operating condition is changed to -10.0°C/90.0°C there is also an increase of heat flux in the electric motor, which is specially noticed on region *em1* (increase of 70%). Moreover, one can verify that the increase of heat flux for region *em1* is linked to the increase of temperature difference.



(a) (b)

Figure 6. Measurements at the electric motor surface: (a) heat flux; (b) heat transfer coefficients.



(a) (b)

Figure 7. Difference between the temperatures in the fluid and on the electric motor surface:
(a) -23.3ºC/40.5ºC; (b) -10.0ºC/90.0ºC.

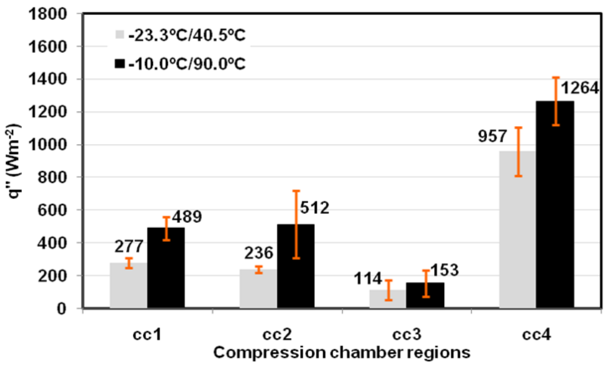
By examining Fig. 8(a), one can notice that the highest heat flux at the compression chamber is found at region *cc4*, which is about three times greater than the heat flux at region *cc1*. Such a high heat flux is related to the rotational movement of the crankshaft that increases the gas velocity next to surface *cc4* and also spatters oil droplets against the same surface. This conclusion is supported by results for convective heat transfer coefficients shown in Fig. 8(b), where the highest coefficient is verified for surface *cc4*.

As the operating condition is changed to -10.0°C/90.0°C, heat flux is increased at all four surfaces of the component, with the surface *cc4* maintaining the highest level. As can be seen in Figs. 8(c) and 8(d), the rise of heat flux is associated with a small increase of temperature difference between the compression chamber surface and the gas.

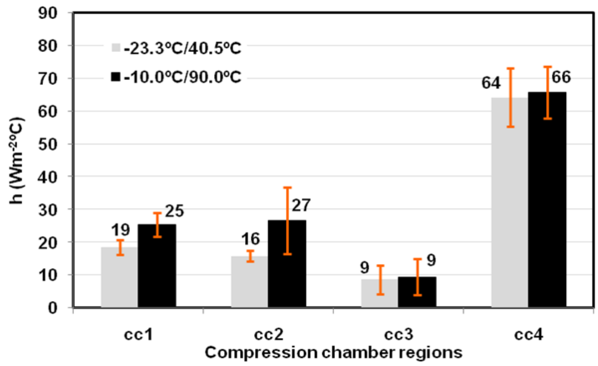
The surface c*c3* displays a quite distinct behavior in relation to the other surfaces, since virtually no variation in the heat flux and surface temperature is observed as the operating condition is changed. Fig. 9 shows that the temperature of surface *cc3* is higher than that measured with the thermocouple to characterize the other surfaces. Considering that the communicating pipe that connects the discharge chamber to the discharge muffler is very close to the surface *cc3* (Fig. 10), the temperature at the surface *cc3* can be estimated to be about 3°C higher than the measured value. Therefore, the heat transfer coefficient at region *cc3* is possibly even lower than the experimental data of Fig. 8(b). The region in which surface cc3 is located makes it difficult gas motion, besides being inaccessible for the lubricating oil flow.

3.3. Effect of lubricating oil on heat transfer

A baffle was carefully installed close to the top of the crankshaft, in order to block the lubricating oil flow and, therefore, avoiding the oil from reaching the upper and side shell internal surfaces. This allows the evaluation of the effect of the lubricating oil flow on the shell heat transfer. Fig. 11 shows the baffle installed on the compressor crankcase. Local heat flux and temperatures were measured with and without (standard condition) the baffle. A comparison between measurements for both operating conditions is presented in Fig. 12, with results for temperature (a) and heat flux (b) at several positions of the compressor shell internal surface.

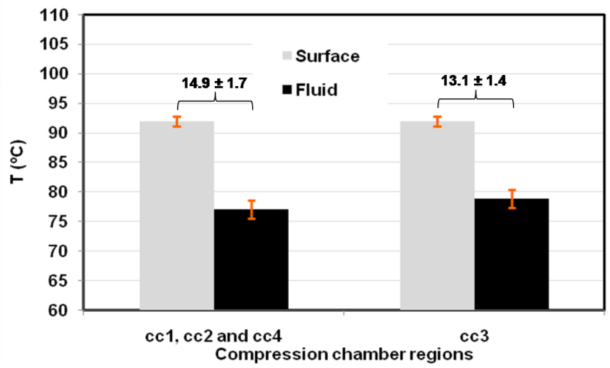


(a)

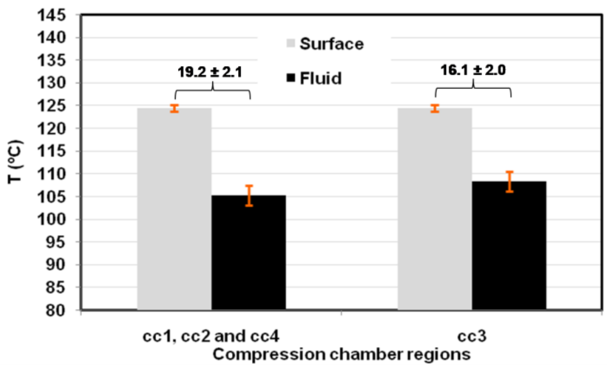


(b)

Figure 8. Measurements at the compression chamber: (a) heat flux; (b) heat transfer coefficients.



(a)



(b)

Figure 9. Difference between the temperatures in the fluid and on the compression chamber:
(a) -23.3ºC/40.5ºC; (b) -10.0ºC/90.0ºC.

From Fig. 12, one can notice high flux levels at the border of the shell cover (*i9*, *i10*, *i11*) and at region *i6*. After leaving the pump, the oil hits against the cover surface, forming an impinging jet in the form of volute that enhances heat transfer in that regions. Figure 8(b) depicts higher values of heat transfer coefficients on surfaces *i9*, *i10* and *i6*, confirming the previously mentioned oil effect. The heat transfer coefficient was not evaluated at region *i11* because the HFS had no capability for temperature measurement. It is noticed that when the compressor operating condition is changed from -23.3ºC/40.5ºC to -10.0ºC/90.0ºC, the levels of heat transfer and temperature also increase.

It can be clearly seen that the absence of lubricating oil flow on the upper surface (*i8*, *i9*, *i10* and *i11*) and on the side surface (*i1*, *i2*, *i3* and *i4*) provides a reduction of heat flux and temperature, especially on the upper surface. One exception of this behavior is the surface of region *i1*, where there is an increase in the heat flux when the lubricating oil flow is blocked by the baffle. In the upper surface, the presence of the baffle brings about a reduction of 50% in the heat flux and a decrease of more than 10°C in the temperature levels. On the other hand, at the bottom surface (*i5*) the heat flux and temperature are increased. The baffle avoids the oil jet from impinging against the upper surfaces of the shell and, therefore, from flowing as a film over the side surfaces. Thus, in this situation the lubricating oil returns to the sump by flowing over the compressor kit.

A comparative analysis of the heat transfer distribution among the upper (cover), side and bottom (sump) surfaces of the shell was carried out considering the standard and the baffle test conditions. It was found that for the standard situation (without baffle), the upper surface contributes 37% of the total heat rejection, whereas the bottom surface (oil sump) is responsible for only 14% of the total heat loss. When the baffle is installed, the lubricating oil does not interact with the upper surfaces and its contribution to the heat loss decreases to 23%. Since the oil returns to the sump with a higher temperature than that for the standard condition, the heat loss of the bottom surface increases from 14% to 30% of the total heat. In these situations, the heat loss through the side wall is virtually the same, with 49% for the standard configuration and 47% with the presence of a baffle.

Besides affecting the heat transfer distribution among different regions of the shell, the lubricating oil affects the total heat loss through the shell. Accordingly, the compressor rejects 87W to the external environment for the standard condition and 78W (10% lower) when the baffle is assembled. Due to this heat loss reduction, the discharge temperature increases from 89.1ºC to 94.6ºC (6% higher).

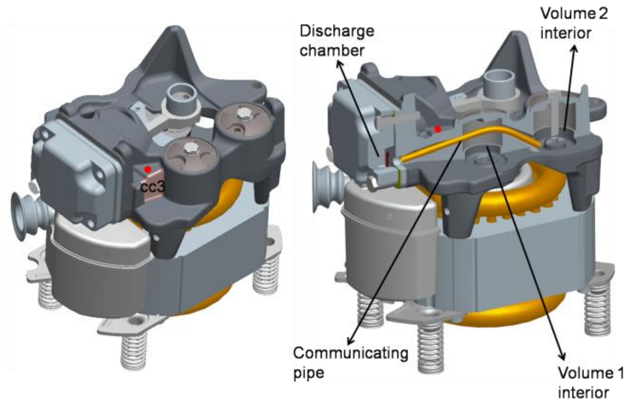


Figure 10. Communicating pipe connecting discharge chamber and discharge muffler

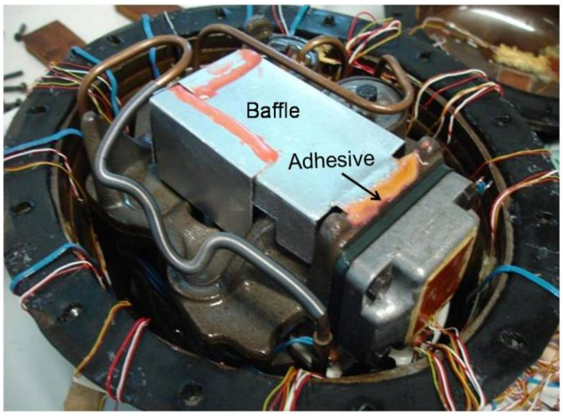
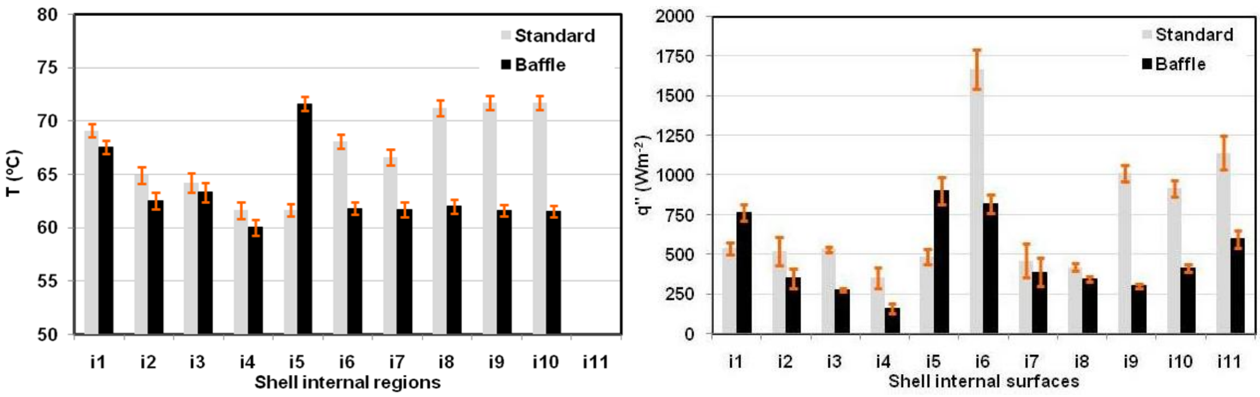


Figure 11. Baffle installed inside the compressor.



(a) (b)

Figure 12. Influence of lubricating oil on (a) temperature and (b) heat flux at the internal surface of the compressor shell.

4. CONCLUSIONS

This paper reported an experimental investigation of heat transfer at the surfaces of the electric motor and compression chamber of a small reciprocating compressor. To this purpose, heat flux sensors and thermocouples were employed, allowing the local characterization of heat transfer in several regions of each compressor component. The results have shown the influence of the suction system on the electric motor heat transfer and the action of the crankshaft mechanism on the heat transfer enhancement at the external surface of the compression chamber. Additionally, it was observed that the lubricating oil flow considerably affects the total heat loss through the shell and the heat transfer distribution among upper, side and bottom regions of the compressor shell surface.

5. 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.

6. REFERENCES

Almbauer, R. A., 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. *Proc. Int. Compressor Engineering Conf. at Purdue*, paper C079.

Cavallini, A., Doretti, L., Longo, G. A., Rosseto, L., Bella, B., Zannerio, A., 1996. Thermal analysis of a hermetic reciprocating compressor. *Proc. Int. Compressor Engineering Conference at Purdue*, pp. 535-540.

Dutra, T., 2008. *Experimental Investigation of the Heat Transfer in Reciprocating Compressors Adopted for Household Refrigeration*. M.Sc. Thesis, PPGEM, Federal University of Santa Catarina, 234p.

Kim, S. H., Sim, Y. H., Youn, Y., Min, M. K., 2000. An Experimental Study on Internal Temperature Distribution and Performance Characteristics in a Reciprocating Compressor for a Domestic Refrigerator. *Proc. Int. Compressor Engineering Conf. at Purdue* (1), pp. 245-252.

Kurazumi, Y.; Tsuchikawa, T.; Ishii, J.; Fukagawa, K.; Yamato, Y.; Matsubara, N., 2008. Radiative and Convective Heat Transfer Coefficients of the Human Body in Natural Convection. *Building and Environment* 43 (12), 2142-2153.

Marinoski, D. L.; Güths, S.; Pereira, F. O. R.; Lamberts, R., 2006. Improvement of a Measurement System for a Solar Heat Gain through Fenestrations. *Energy and Buildings* 39 (4), 478-487.

Melo, C.; Silva, L.W.; Pereira, R.H., 2000. Experimental Evaluation of the Heat Transfer through the Walls of Household Refrigerators, *Proc. Int. Compressor Engineering Conf. at Purdue*, pp. 353-359.

Meyer WA, Thompson HD. 1990, “An Experimental Investigation into Heat Transfer to the Suction Gas in a Low-Side Hermetic Refrigeration Compressor”, Proc. Int. Compressor Engineering Conf. Purdue, pp. 908-916.

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

Pizarro-Recabarren RA. 2007, Influência do óleo lubrificante na transferência de calor em um compressor hermético alternativo, M.Sc. Thesis (in portuguese), Departament of Mechanical Engineering, Federal University of Santa Catarina, Florianopolis, SC, Brazil.

Raja, B., Sekhar, S. J., Lal D. M., Kalanidhi, A. A., 2003. Numerical Model for Thermal Mapping in a Hermetically Sealed Reciprocating Refrigerant Compressor. *Int. J. Refrigeration* 26 (6), pp. 229-236.

Ribas Jr. FA. 2007, “Thermal Analysis of Reciprocating Compressors”, International Conference on Compressors and Their Systems, London, pp. 277-287.

Sabau, A. S.; Wu, Z., 2007. Evaluation of a Heat Flux Sensor Spray Cooling for the Die Casting Processes. *J. Materials Processing Technology* 182 (1-3), 312-318.

Shiva Prasad, B. G., 1992. Regenerative Heat Transfer in Reciprocating Compressors. *Paper based on original version presented at the 1992 PCRC*.

Shiva Prasad, B.G., 1998, Heat Transfer in Reciprocating Compressors - A Review, *Proc. Int. Compressor Engrg. Conf. at Purdue*, pp. 857-863.

Silberstein, R.; Held, A.; Hatton, T.; Vilney, N.; Sivapalan, M., 2001. Energy Balance of a Natural Jarrah (Eucalyptus Marginata) Forest in Western Australia: Measurements during the Spring and Summer. *Agricultural and Forest Meteorology* 109 (2), 79-104.

Sim YH, Youn Y, Min MK, 2000, A study on heat transfer and performance analysis of hermetic reciprocating compressors for refrigerators, Proc. Int. Compressor Engineering Conf. Purdue, West Lafayette, pp. 229-236.

Todescat ML, Fagotti F, Prata AT, Ferreira RTS. 1992, Thermal Energy Analysis in Reciprocating Hermetic Compressors, Proc. Int. Compressor Engineering Conf. Purdue, West Lafayette, pp. 1419-1428.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.