EXPERIMENTAL CHARACTERIZATION OF HEAT TRANSFER DURING THE THERMAL TRANSIENT OF A RECIPROCATING COMPRESSOR

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Abstract. *The efficiency of reciprocating compressors adopted in household refrigeration systems is greatly affected by gas superheating that takes place along the flow through the suction system and inside the cylinder. The present paper reports an experimental investigation aimed at characterizing heat transfer in a small reciprocating compressor during the thermal transient regime that takes place after the compressor is started up. Measurements of heat flux and temperature are carried out in several components of the compressor. The investigation reveals the presence of different time scales associated with the heat transfer process in each component during the thermal transient. It has also been observed that both the lubricating oil pumped from the sump and the gas motion induced by the crankshaft mechanism inside the shell have a major effect on the phenomenon.*

Keywords: *reciprocating compressor, heat flux, thermal transient*

1. INTRODUCTION

The compression process in a hermetic reciprocating compressor generates a significant amount of heat that increases gas superheating and, hence, negatively affecting the compressor performance. According to Ribas Jr. *et al.* (2008), superheating may represents as much as 49% of the thermodynamic loss in reciprocating compressors. Therefore, the study of such phenomenon is necessary to design more efficient compressors and, as consequence, systems of smaller power consumption. Several simulation models have been developed to provide a better understanding of heat transfer inside the compressor. However, thermal analysis of compressors is a difficult task because the associated geometry is very complex and does not allow simple modeling approaches.

Several studies of heat transfer in reciprocating compressors are focused on numerical modeling. For instance, Todescat *et al*. (1992) adopted an integral formulation and, by applying mass and energy balances to selected control volumes, obtained the compressor thermal profile. Sim *et al.* (2000) and Ooi (2003) developed simulation strategies with heat transfer coefficients in each component being evaluated from classical correlations available in the literature. However, with the exception of simple geometries, it is difficult to find adequate correlations for all of the compressor components. Furthermore, there must be also a good understanding of the main heat transfer mechanisms in each component in order to select a proper heat transfer correlation.

A limitation present in the methodologies proposed by Todescat *et al.* (1992) and Ooi (2003) is the impossibility to predict the interaction between the components by heat conduction. In the model of Todescat *et al.* (1992) such effect is incorporated somehow via calibrated thermal conductances but this technique is of very little utility if different compressor layouts has to be analyzed. In the strategy developed by Ooi (2003), a simple one-dimensional heat conduction model is adopted but it is not sufficient to correctly predict heat transfer in some components, such as the cylinder head. In order to circumvent such limitations, Ribas Jr. (2007) proposed a hybrid model in which a differential formulation is used to solve heat conduction in the solid components and an integral formulation is applied to evaluate temperatures of the gas as it flows throughout the compressor.

Experimental studies have considered different aspects of heat transfer in reciprocating compressors. Meyer and Thompson (1990) used thermocouples to measure the gas temperature in different components and by applying energy balances characterized the heat transfer process through the concept of an equivalent thermal conductance (*UA*). However, the use of thermocouples is not enough for a detailed description of superheating. Dutra and Deschamps (2010) describe an experimental investigation in which heat flux sensors and thermocouples were jointly applied to obtain local heat transfer coefficients.

The present paper reports an experimental investigation aimed at characterizing heat transfer in different components of a reciprocating compressor during the thermal transient regime that occurs after the compressor is started up. Such a study is relevant because the compressor in domestic refrigeration applications usually works in an on-off mode. The aim is to understand the preferential path of heat during the initial operation of the compressor. For this purpose, measurements of temperature and heat flux were carried out at several regions of the compressor, such as the shell, electric motor, suction muffler and cylinder.

2. EXPERIMENTAL PROCEDURE

2.1. Heat flux sensors

The working principle of a heat flux sensor (HFS) is based on a thermocouple serial association. This configuration amplifies the output voltage signal for a temperature difference and its output signal E can be expressed as:

$$E = N\alpha\Delta T \tag{1}$$

where N is the number of thermocouples junctions, α is the thermoelectric sensitivity of the materials and ΔT is the difference of temperature between the lower and the upper surfaces of the HFS.

The heat flux across the sensor can be calculated by the Fourier Law:

$$q'' = \frac{k\Delta T}{d} \tag{2}$$

where q'' is the heat flux per unit of area across the HFS, k is the thermal conductivity of the filling material and d is the sensor thickness. By combining Eq. (1) and Eq. (2), one can obtain a relationship between the heat transfer q'' and the voltage output signal E:

$$E = \frac{N\alpha d}{k} q'' \tag{3}$$

The sensitivity of the HFS is defined as:

$$S = \frac{E}{q''} = \frac{N\alpha d}{k} \tag{4}$$

The thermal inertia of sensors can be characterized by its response time t and modeled as an electric circuit composed by a resistor R and a capacitor C:

$$t = RC \tag{5}$$

Based on the thermal analogy of Ohm's law, the resistance R can be understood as the ratio between the temperature difference and the heat flux:

$$R = \frac{\Delta T}{q} \tag{6}$$

By denoting $q'' = q/A$, where A is the heat change area, and substituting Eq. (2) into Eq. (6) results:

$$R = \frac{d}{kA} \tag{7}$$

The heat capacity is the required energy to change the temperature of a body by one degree. In the case of a HFS it is given by:

$$C = \rho\, A\, d\, c \tag{8}$$

where ρ is the density and c is the specific heat of the HFS filling material. Finally the response time of a heat flux sensors is:

$$t = \frac{d^2 \rho c}{k} \tag{9}$$

According to Eq. (9), the thickness is the most influential parameter on the response time. In order to reduce the response time the sensor needs to be thin and made from a material with high thermal conductivity and low specific heat.

2.2. Compressor instrumentation

Heat flux sensors (HFS) and thermocouples were fixed onto the surfaces of different compressor components, such as the shell, suction muffler, motor and cylinder. As far as the shell is concerned, heat flux sensors were fixed in three regions of both the internal and external surfaces, as shown in Fig. 1. The positioning of all other HFS adopted for the suction muffler, motor and cylinder are depicted in Fig. 2. All heat flux sensors employed in the present study are provided with a thermocouple to measure its surface temperature.

The instrumentation of HFSs on the internal components is not a simple task, due to the presence of high temperature levels and lubricating oil. Epoxy-adhesives were used to attach HFSs to the surfaces and the wires of each sensor were carefully positioned to avoid disturbances in the lubrication oil flow. This is of crucial importance because the oil significantly affects the heat transfer process inside the compressor, increasing heat loss to the external ambient. The flow of oil is promoted by a pump, which collects oil in the sump and takes it to the upper parts of the compressor shell by centrifugal action. Figure 3 presents a schematic view of the compressor components and the indication of the lubricating oil flow. In addition to HFSs, thermocouples were also adopted to measure the temperature of the gas in the internal ambient.

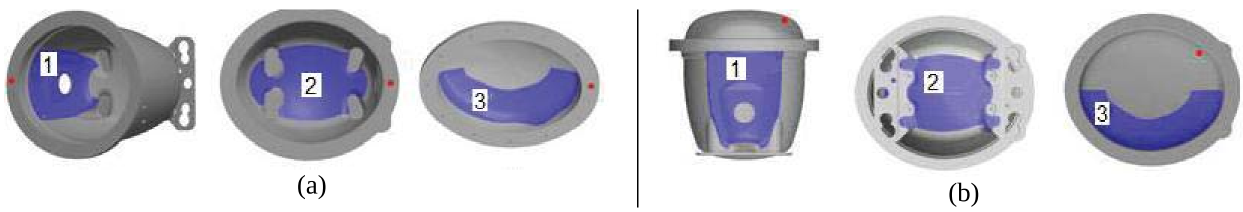

Figure 1. Measurements locations on the compressor shell: (a) internal surface; (b) external surface.

Figure 2. Measurements locations on internal components.

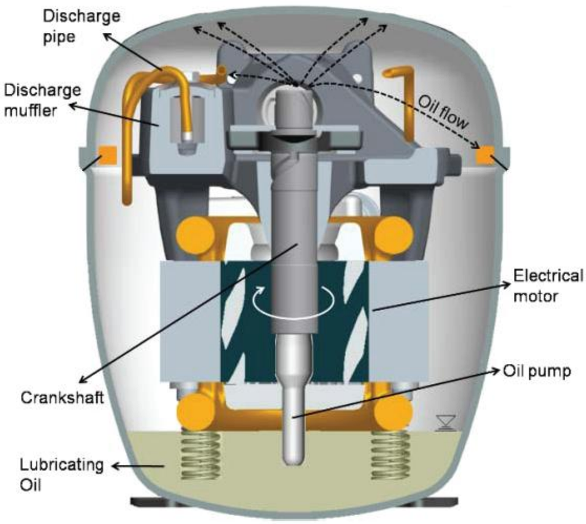

Figure 3. Compressor schematic view.

3. RESULTS

A 60 Hz reciprocating compressor operating with refrigerant 134a was selected for the analysis, being submitted to an operating condition represented by evaporating and condensing temperatures equal to -23.3°C and 40.5°C, respectively. A calorimeter facility was employed to test the compressor. The uncertainties associated with measurements are lower than 2% for mass flow rate and power consumption and around 1% for suction and discharge pressures. The measurements were acquired along a period of two hours after the compressor startup. The results shown in this section represent the average of two measurements.

3.1. Measurements on the shell

The results of heat flux and temperature at the selected regions inside and outside the compressor shell (Fig. 1) are shown in Figs. 4 and 5, respectively. It should be noticed that negative values in Fig. 4 express that heat flux is towards the surface and vice-versa if the signal is positive. As can be seen from Figs. 4 and 5, heat transfer on the internal surface a3 of the shell cover increases very fast after the compressor startup because lubricating oil exiting the pump (Fig. 3) hits the surface and transfers energy more effectively than on the other internal surfaces (a1 and a2). For this reason, region a1 reaches a temperature above 70^{o}C. The corresponding external surface of the shell cover (b3) is associated with the same phenomenon and has also the influence of an air flow supplied by a fan installed inside the calorimeter compartment in which the compressor is tested.

3.2. Measurements on internal components

The surfaces of the internal components were divided into three parts: suction muffler (s1, s2), electric motor (s3, s4, s5) and cylinder (s6, s7, s8) as identified in Fig. 2. Results for heat flux and temperature on the muffler are shown on Figs. 6 and 7. It can be clearly seen that heat flux on surface s1 is higher than that on s2. This occurs because s1 interacts more with the gas flow than s2, as suggested by fluctuations in the signal of heat flux. In fact, surface s2 is adjacent to surface s3 on the electric motor and the proximity between them leaves very little space for air motion. Figure 7 presents measurements of temperature on surfaces s1 and s2, as well as in the internal ambient provided by a thermocouple located in the gas near of the muffler, showing similar temperature transients.

Results for the electric motor (Figs. 8 and 9) show that surface s3 releases heat during all the period in which the compressor is kept switched on. As already mentioned s3 is very close to s2 and, therefore, does not interact with the internal ambient as surfaces s4 and s5 do. On the other hand, surfaces s4 and s5 receive heat on the first minutes and eventually start to release. This inversion of the heat flux signal can be understood with the assistance of Fig. 9. The temperatures on the electric motor surfaces are very similar to each other and for this reason just that of surface s5 is shown in Fig. 9. Yet, the temperatures of the internal ambient were measured in three different points. Figure 9 shows that after the compressor starts up the temperature of the internal ambient increases because of heat being released by the cylinder, discharge system and bearings. Both the lubricating oil pumped from the sump and the gas motion induced by the crankshaft mechanism inside the shell intensifies this heat transfer process. Initially, heat is transferred to the electric motor, but because heat is also generated inside the motor its temperature becomes greater that the temperature of the internal ambient after a certain period. This is precisely the moment in which occurs the inversion of heat flux signal verified for surfaces s4 and s5.

Finally, Figs. 10 and 11 show that high heat flux takes place on surface s8 of the cylinder when compared with the other surfaces (s6 and s7). This occurs because heat transfer on s8 is dominated by the effect of both the oil and gas flow induced by the crankshaft. All three surfaces (s6, s7 and s8) reach a maximum value of heat flux within 30 minutes from the compressor startup. As Fig. 11 shows, the temperature of the cylinder increases much faster than those of the internal ambient along the first minutes and this is the reason for the indicated peak of heat transfer.

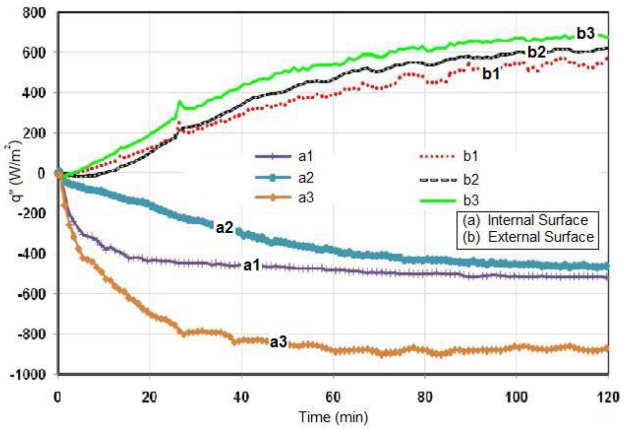

Figure 4. Heat flux on the shell.

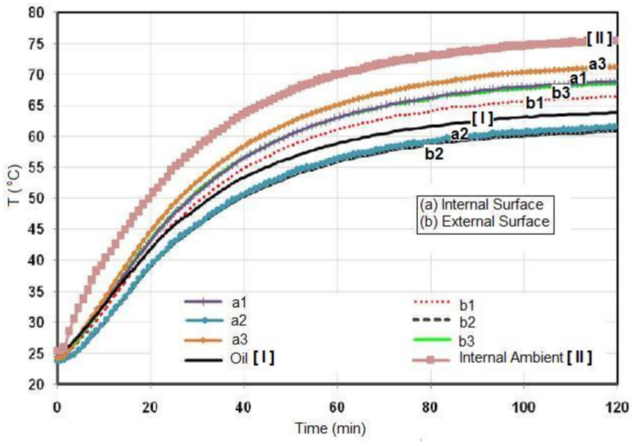

Figure 5. Temperature on the shell.

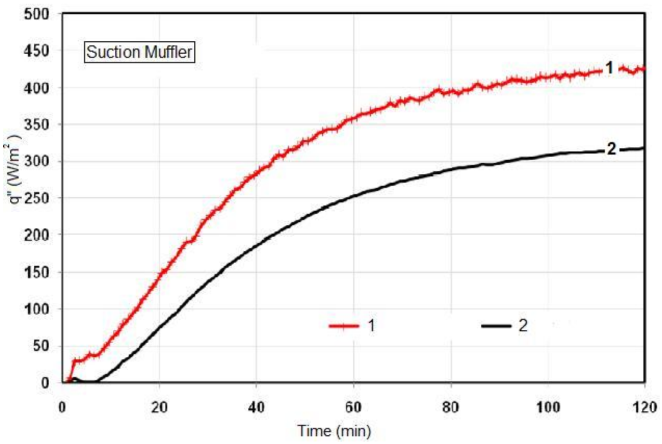

Figure 6. Heat flux on the suction muffler.

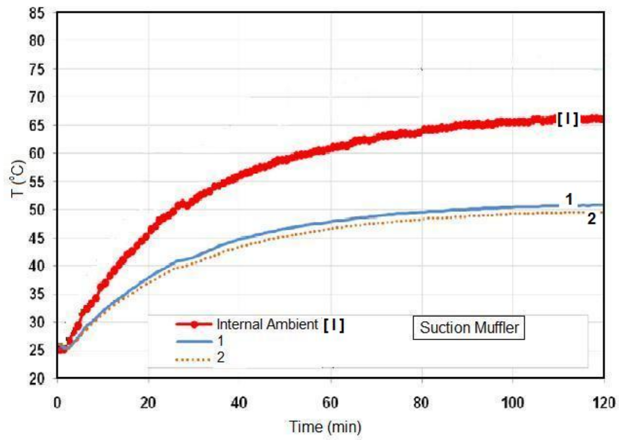

Figure 7. Temperature on the suction muffler.

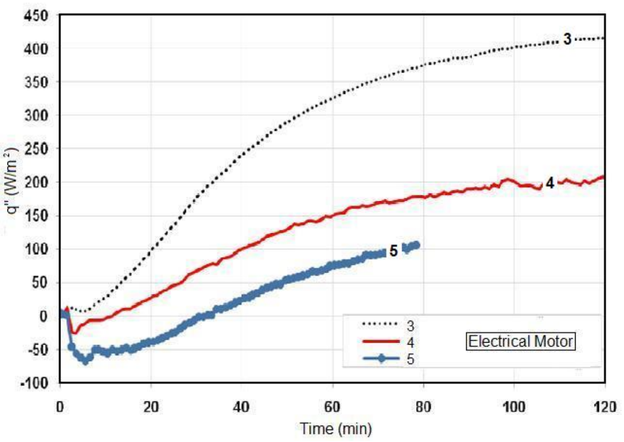

Figure 8. Heat flux on the electric motor.

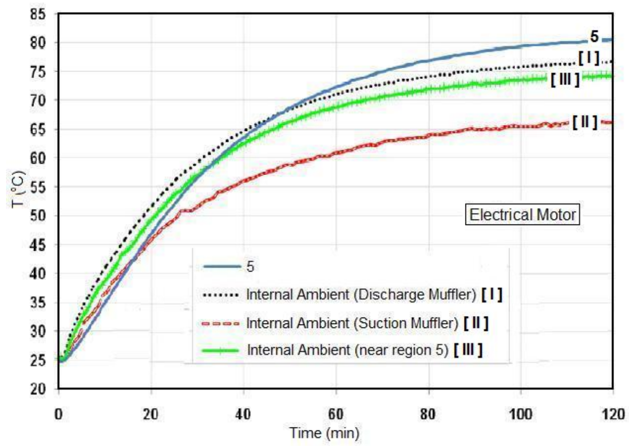

Figure 9. Temperature on the electric motor.

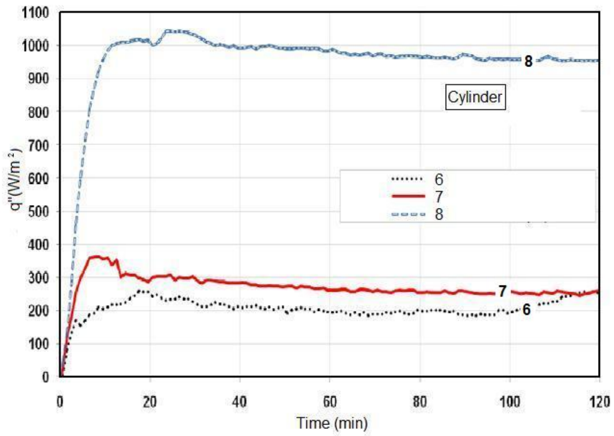

Figure 10. Heat flux on the cylinder.

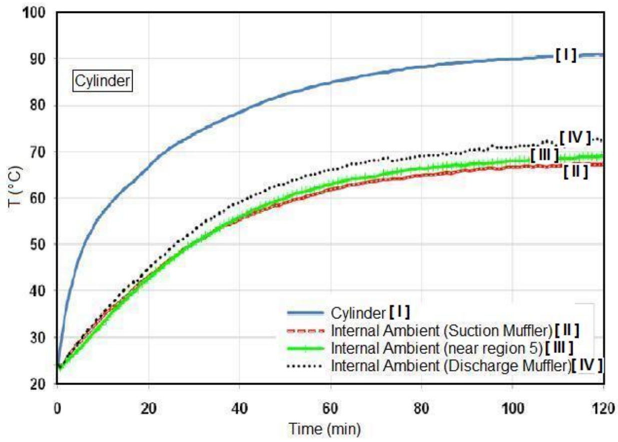

Figure 11. Temperature on the cylinder.

4. CONCLUSIONS

The present paper reported the results of an experimental investigation carried out to clarify some of the phenomena associated with heat transfer during the thermal transient of reciprocating compressors. Heat flux sensors and thermocouples were fixed onto the surfaces of different compressor components, such as the shell, suction muffler, electric motor and cylinder. The results reveal the presence of different time scales associated with the heat transfer process in each component. For example, it has been noticed that the gas temperature in the internal ambient increases much more slowly than the temperature of the cylinder surface. On the other hand, the temperature of the gas increases faster than in some of the electric motor surfaces. It has also been observed that both the lubricating oil pumped from the sump and the gas motion induced by the crankshaft mechanism have a major effect on the heat transfer process on the cylinder and on the compressor shell.

5. ACKNOWLEDGEMENTS

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