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Experimental Investigation of Heat Transfer in Components of a Hermetic Reciprocating Compressor under Thermal Transient

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ABSTRACT

Heat transfer has an important role on the efficiency and reliability of hermetic reciprocating compressors adopted for household refrigeration. Most analyses of such compressors are carried out for steady-state operating conditions based on established procedures and standards. However, since household refrigeration systems work by alternating periods in which the compressor is on (ON) and idle (OFF), the performance assessment of compressors should take into account the ON/OFF cyclic operation. This paper presents an experimental study of the heat transfer in components of a hermetic reciprocating compressor during thermal transients due to ON/OFF cyclic operation. The compressor was instrumented with thin-film heat flux sensors and thermocouples to evaluate time-varying heat transfer coefficients. We found that heat flux in some regions is significantly affected by the cyclic operating condition. Moreover, the investigation revealed the presence of different time scales due to distinct thermal inertia and heat transfer associated with each component.

1. INTRODUCTION

Heat transfer can considerably affect the efficiency and reliability of hermetic reciprocating compressors. In one hand, the superheating of the suction gas reduces the compressor volumetric and isentropic efficiencies. On the other hand, reliability issues limit the electrical motor and lubricant oil temperatures. Hence, significant research has been developed to better understand the heat transfer between the compressor components.

Thermal simulation models have been developed based on lumped-parameter formulation (Todescat *et al.*, 1992; Ooi, 2003), in which the heat transfer between lumped elements is approached from measurements or correlations available in the literature. Other simulation models adopt differential formulation (Raja *et al.*, 2003; Birari *et al.*, 2006), solving the heat conduction in solid components and the fluid flow via the finite-volume method. Such models do not require convective heat transfer coefficients to estimate heat transfer between the gas and solid components. Although a full differential formulation provides a detailed thermal characterization of the compressor, the numerical solution of the governing equations is associated with high computation cost. Hybrid simulation models (Sanvezzo Jr. and Deschamps, 2012) formed by lumped and differential formulations for the gas and solid components, respectively, have been proposed to reduce the computational cost while retaining a detailed characterization of heat transfer in the solid components.

Experimental studies have considered different aspects of heat transfer in reciprocating compressors. Meyer and Thompson (1990) used thermocouples to measure the gas temperature along the flow path and, by applying energy balances, determined the heat transfer rate through components. Dutra and Deschamps (2013) carried out measurements of heat flux and temperature in a single-speed small reciprocating compressor by using thin-film heat flux sensors (HFSs) and thermocouples, and combined such measurements to estimate local heat transfer coefficients.

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All the aforementioned investigations were conducted for the compressor operating under steady-state condition, following well-established standards for evaluation and certification criteria of compressors (EN12900:2005). Nevertheless, household refrigeration systems operate under cyclic ON/OFF conditions and such transients can affect the compressor performance. Available studies in the literature concerning the ON/OFF operation of refrigeration systems are focused on heat exchangers and expansion devices (Hermes and Melo, 2008). Only recently, Lohn *et al.* (2015) presented a hybrid thermal model to predict the temperature distribution in a hermetic reciprocating compressor under a cyclic ON/OFF condition.

The present paper reports an experimental investigation aimed at characterizing the heat transfer in a hermetic reciprocating compressor during transient conditions. Evaporating and condensing temperatures were fixed at -21°C and 40°C , respectively. The compressor was tested under steady-state operating condition and two ON/OFF cyclic conditions: (i) 12 min. ON/15 min. OFF; and (ii) 25 min. ON/30min. OFF. Measurements of temperature and heat flux were carried out in different regions and components of the compressor, allowing estimates of local heat transfer coefficients as a function of time.

2. EXPERIMENTAL PROCEDURE

2.1 Compressor Instrumentation

Several HFSs and thermocouples were fitted onto the surface of the electrical motor, suction muffler and discharge system, so as to obtain a detailed thermal characterization of the compressor: (i) three on the electrical motor surfaces: em1, em2 and em3; (ii) two on the suction muffler: sm1 and sm2; and (iii) five on the discharge system: dp1, dm1, dm2, dm3 and dm4. The location of each HFS on these components is shown in Figures 1-3.

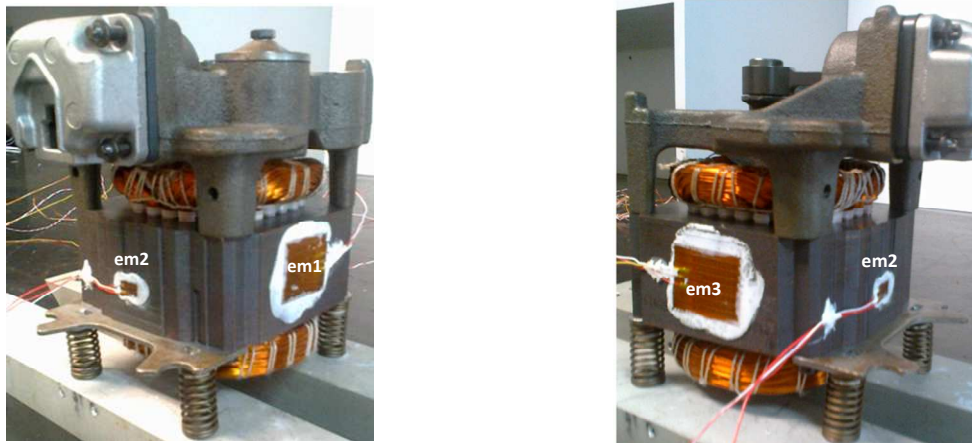


Figure 1: Instrumentation of the electrical motor.

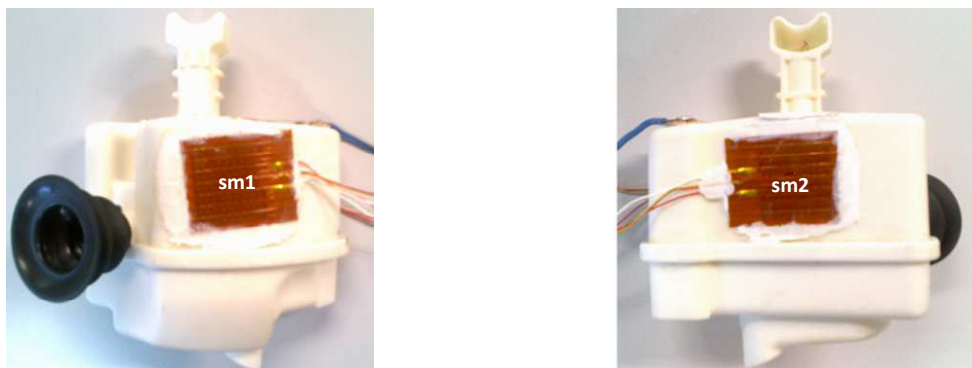


Figure 2: Instrumentation of the suction muffler.

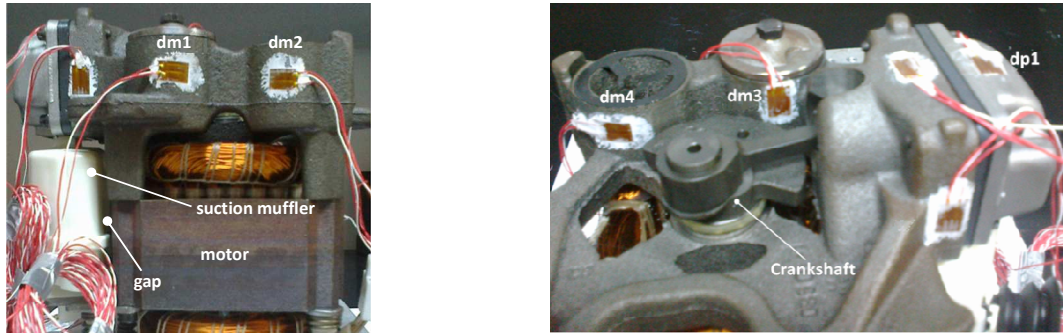


Figure 3: Instrumentation of the cylinder head and discharge muffler.

The operating principle of a thin-film HFS consists of a self-generated voltage output proportional to a heat flux through the sensor. The generated voltage (V) is the result of a temperature difference (ΔT) across the thickness of the HFS, which is measured by a thermocouple serial association ($V = N \alpha \Delta T$), being N and α the number of thermocouple junctions and the thermoelectric sensitivity of the materials, respectively. The heat flux through the HFS surfaces, q'' , is a function of its thermal conductivity, k , thickness, t , and temperature difference ΔT ($q'' = k \Delta T / t$). Therefore, a linear relationship between heat flux and voltage output is obtained ($q'' = V / S$), where $S (= N \alpha t / k)$ is defined as the HFS sensitivity. Although the value of S can be theoretically determined, this is not a common practice. The HFS sensitivity is usually assessed from a standardized calibration procedure performed by the sensor manufacturer which supplies a datasheet containing all the technical specifications, including the HFS sensitivity.

The instrumentation of HFSs on the compressor components is not a simple task, due to the high temperature levels and the presence of lubricant oil. An epoxy-adhesive was necessary to attach the HFSs onto the surfaces and their wires were carefully positioned inside the compressor to minimize disturbances in the lubricant oil flow. This is a relevant aspect, because the lubricant oil significantly affects the heat transfer inside the compressor, increasing the heat loss to the external ambient (Dutra and Deschamps, 2013). After the installation of the HFSs, thermocouples were also positioned in the gas within the compressor shell, in order to establish suitable reference temperatures, T_∞ , to estimate the local heat transfer coefficients:

$$h = \frac{q''}{T_s - T_\infty} \quad (1)$$

where T_s is the temperature of the sensor surface, measured by a thermocouple embedded in the HFS.

2.2 Experimental Setup

A calorimeter setup was employed to establish the operating conditions in which the compressor was tested. Figure 4 illustrates a schematic of the calorimeter composed of pipelines, control valves (CV), a mass flow meter (FM), heat exchangers (HX), a thermocouple (TC) and pressure transducers (PT). The compressor (C) raises the refrigerant pressure from point 1 up to point 2, which is then cooled by a heat exchanger (HX1) at point 3. After that, the refrigerant is throttled to an intermediate pressure level (point 4) by means of a control valve (CV1). The mass flow is measured by a Coriolis mass flow meter (FM) and then the refrigerant is cooled again with another heat exchanger (HX2) at point 5. The refrigerant is throttled again (CV2), reducing its pressure to the required evaporating pressure (point 6). Finally, an electrical heater (EH1) and a thermocouple (TC1) are used to adjust the level of superheating in the compressor suction line (point 1), completing the cycle. As can be seen in the pressure-enthalpy diagram of Figure 5, the refrigerant remains as superheated vapor along the entire thermodynamic cycle.

The first step in experimental test is to submit the compressor and the calorimeter pipeline to a suitable vacuum condition, in order to remove air and humidity within the system. After that, a refrigerant charge is supplied to the system and the operating condition is defined by setting the evaporating and condensing temperatures, as well as the ON/OFF period. The transient tests are carried out by switching on and off the compressor in an alternate mode for 8 ON/OFF cycles. The first 5 cycles are necessary to attain a periodically stationary condition and the last 3 are used to sample an average representative cycle. For the steady state condition, the compressor runs uninterruptedly for

approximately 4 hours until no temperature variation higher than 1°C is observed in one hour. After this requirement is met, data are acquired during 45 minutes and the test is finished. For both cyclic and steady-state conditions the sampling rate was 0.25 Hz.

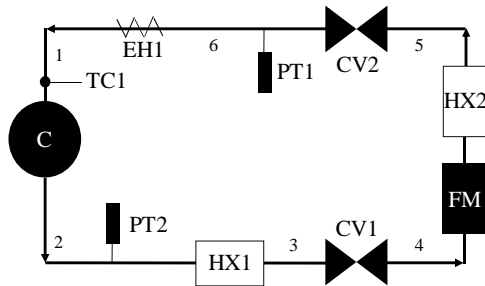


Figure 4: Schematic of the calorimeter facility.

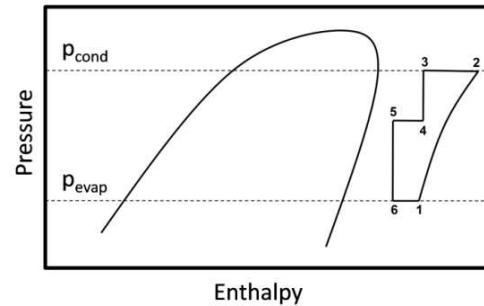


Figure 5: p-h diagram of the hot-cycle calorimeter.

3. RESULTS

A 60 Hz reciprocating compressor operating with R134a was adopted for the measurements. The suction and discharge pressures correspond to saturation pressures associated with the evaporating and condensing temperatures of -21°C and 40°C, respectively. These pressures were set and controlled within a range of ±1% whereas the temperatures of the gas in suction line and of the environment inside the calorimeter were controlled within ±0.5°C. Temperatures, heat fluxes and heat transfer coefficients are normalized in relation to steady-state measurements of the shell temperature, T_{shell} , (in °C), the heat transfer rate released through the shell, q''_{shell} , and the heat transfer coefficient between shell and environment, h_{shell} :

$$T^* = \frac{T}{T_{shell}} \qquad q^{**} = \frac{q''}{q''_{shell}} \qquad h^* = \frac{h}{h_{shell}} \qquad (2)$$

Results under steady-state condition are presented with uncertainty bars corresponding to a 95% confidence interval.

3.1 Gas inside the Compressor

Figures 6-8 show measurements for the gas temperature in four different positions inside the compressor shell under the cyclic ON/OFF conditions (12 min/15 min; 25 min/30 min) and the steady-state condition. The results show that the highest temperatures occur near the shell cover and discharge muffler, while the lowest temperatures are measured in the suction muffler and near the region em3 of the motor. Due to such temperature differences inside the compressor shell, the reference temperatures adopted to estimate the local heat transfer coefficients were those close to the surfaces of interest.

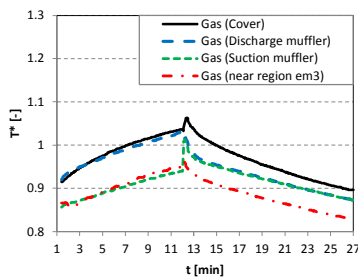


Figure 6: Gas temperatures - 12 min/15 min.

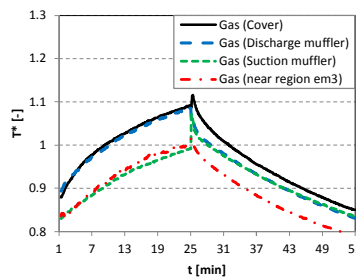


Figure 7: Gas temperatures - 25 min/30 min.

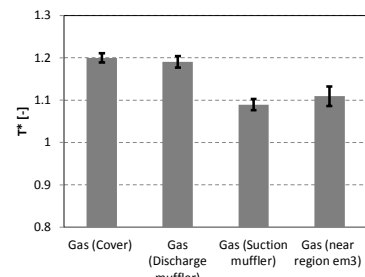


Figure 8: Gas temperatures - steady state.

3.2 Electrical Motor

Measurements of heat flux, temperature and local heat transfer coefficients for the electrical motor under the cyclic (12 min/15 min; 25 min/30 min) and the steady-state conditions are shown in columns (a), (b) and (c) of Figure 9.

During the ON period of the cyclic operation (12 min/15 min), the heat fluxes through the motor surfaces decrease in the first minutes and when this period is longer (25 min/30 min), the heat fluxes start to increase. This is a consequence of the temperature difference between the surfaces (em1, em2 and em3) and the gas inside the compressor shell, which is initially decreased and increased again in the case of a period of 25 min. Immediately after the compressor is switched off, the heat fluxes increase suddenly, since the gas temperature decreases at a much faster rate than the temperature of the solid components. The heat fluxes through the motor surfaces are on the same level when the compressor runs under the cyclic condition. On the other hand, the heat flux through surface em2 is much higher than the others when the compressor operates in steady-state condition. This is due to the proximity between the surface em2 and the suction muffler (Figure 3), which is the coldest component of the compressor under steady-state condition.

The temperature on the motor surfaces em1, em2 and em3 point out similar results, with a slightly higher value on em1, since this region is near the discharge muffler. The temperature of the gas close to the surface em2 was not measured, but estimated as the average of the measurements on the motor and suction muffler surfaces, em2 and sm2, respectively. This approach was adopted because the narrow gap between the electrical motor and the suction muffler (Figure 3) hinders the installation of a thermocouple in that location. Thus, the temperature in the gap is similar to that observed for the em2 region, making it difficult to correctly estimate the local heat transfer coefficient in the idle period. Since the difference between the surface and gas temperatures becomes very small when the compressor is switched off, the heat transfer coefficient suddenly increases and reaches excessive values. Despite this shortcoming, the estimates of heat transfer coefficients, h^* , for the remaining regions are reasonable, in the range of 0.5 - 2, being on the same magnitude for ON/OFF and steady-state conditions.

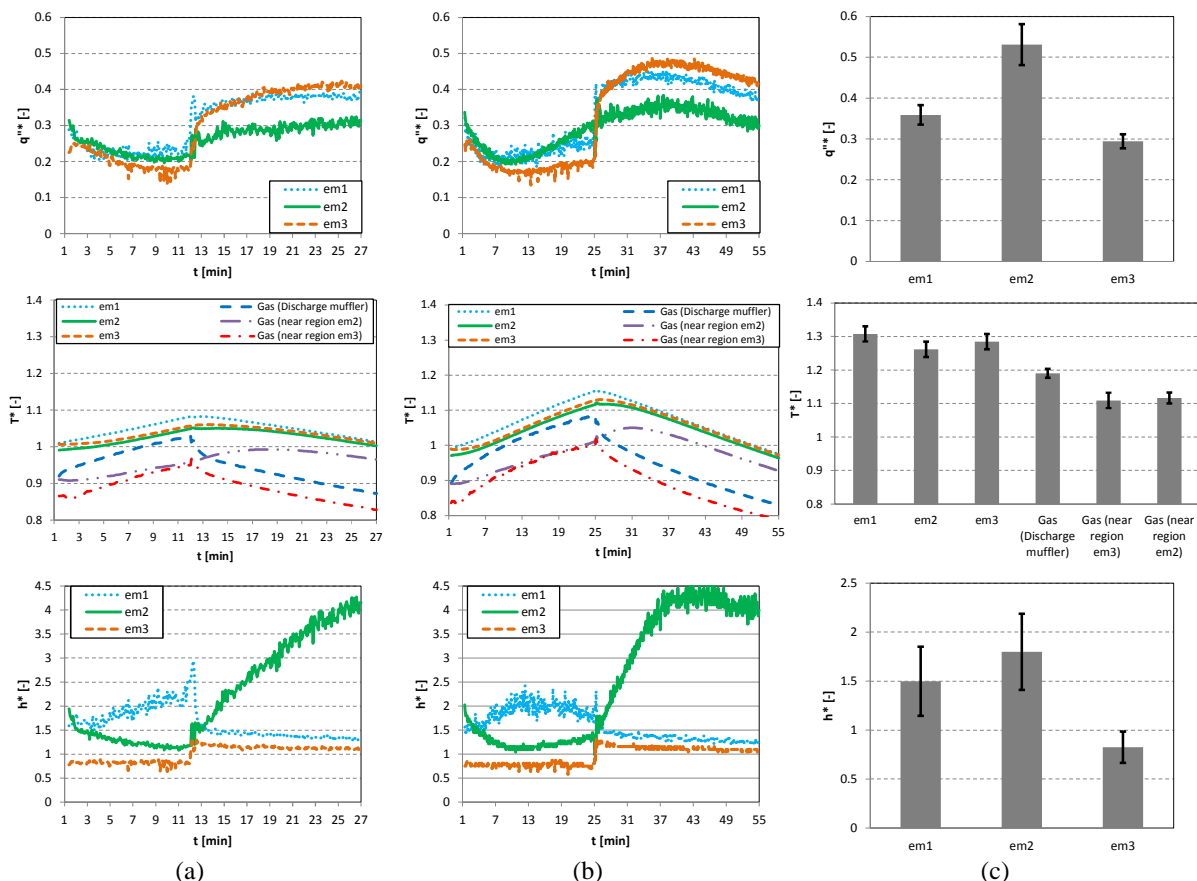


Figure 9: Electrical motor: Heat flux, temperatures and local heat transfer coefficients
(a) 12min/15 min; (b) 25min/30 min; (c) steady state.

3.3 Suction Muffler

The results of heat flux, temperature and local heat transfer coefficients associated with the suction muffler surfaces are shown in Figure 10. The measurements of heat flux suggest that the electrical motor thermally interacts with the suction muffler, since the surface sm2 is subjected to higher heat flux than the surface sm1. Moreover, there is a sudden decrease of heat flux immediately after the compressor is switched off, reaching negative values. When the compressor is idle (OFF), heat transfer occurs first towards the suction muffler, since the gas temperature is still higher than the temperature of the muffler wall. However, eventually during the idle period the temperature of the solid surpasses the gas temperature, and heat flux has its direction changed. These temperature variations can be clearly observed in Figure 10, with the exception of the gas temperature near the region sm2 that is the reference temperature for the surface em2.

As the surface and gas temperatures approach each other, a sudden increase in the heat transfer coefficient is observed on the surface sm1. Naturally, this is not associated with any modification of the flow pattern in the suction muffler vicinity, but simply a result of the approach used to estimate the coefficient, represented by Equation (1). More representative results for the heat transfer coefficients in the idle period can be obtained just before the compressor is switched on, $h^* \sim 1$. The heat transfer coefficients obtained for the ON period and the steady-state operation are on the same order of magnitude, 0.8 – 3.5, compared to those associated with the surface of the electrical motor.

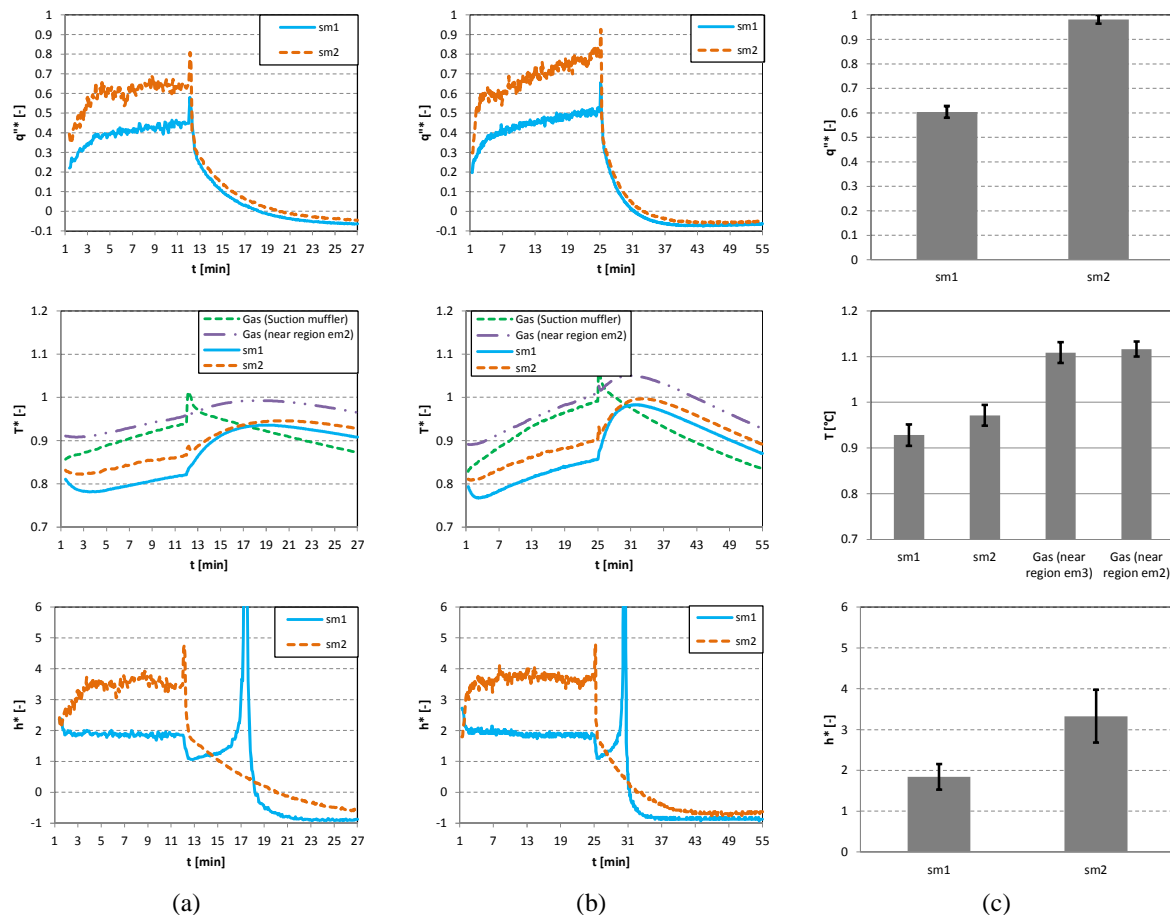


Figure 10: Suction muffler: Heat flux, temperatures and local heat transfer coefficients.
(a) 12min/15 min; (b) 25min/30 min; (c) steady state.

Measurements of temperature were also carried out for the gas in the suction chamber (sc) depicted in Figure 11. This temperature is an important parameter since gas superheating along the suction path reduces the compressor volumetric and isentropic efficiencies. Figure 12 shows the temperature of the gas in suction chamber during the ON

period for the two cyclic conditions and the steady-state condition. As can be seen, gas superheating varies according to how long the compressor runs before being switched off. For the 12min/15min ON/OFF condition, the non-dimensional temperature in the suction chamber reaches 0.72, whereas the temperature measured in the steady-state condition is 0.78.

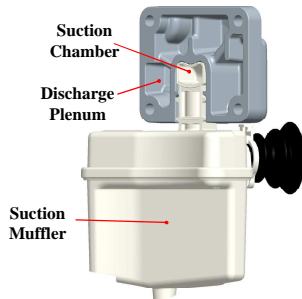


Figure 11: Suction chamber.

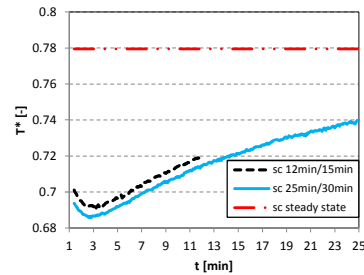


Figure 12: Suction chamber gas temperature.

3.4 Discharge System

Experimental data of temperature, heat flux and local heat transfer coefficients at the discharge system regions are shown in Figures 13-15 for the steady-state and ON/OFF operating conditions. As can be noted in Figure 13, the cylinder head, dp1, is the component of the discharge system with the highest temperature, regardless the compressor operating condition. This is the reason for the higher heat flux found at dp1 in comparison to the locations dm1 and dm2 (Figure 14). It is interesting to observe in Figure 15 that h^* is about 1 – 1.5 for the surfaces dm1 and dm2, which are much smaller than the values obtained at the surfaces dm3 and dm4 (in the range 7 - 12). Such a difference can be explained by the proximity of each surface in relation to the crankshaft, as depicted in Figure 3. Specific to this type of compressor, the oil stored in the sump is taken to the upper parts by the centrifugal action of an oil pump that 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 of the crankshaft. Part of the oil that leaves the crankshaft impinges against the surfaces dm3 and dm4, increasing the heat flux there. The highest heat transfer coefficient is found at dm3 because this surface is the closest to the crankshaft. On the other hand, the temperature at dm4 is higher than at dm3, and this explains its greater heat flux.

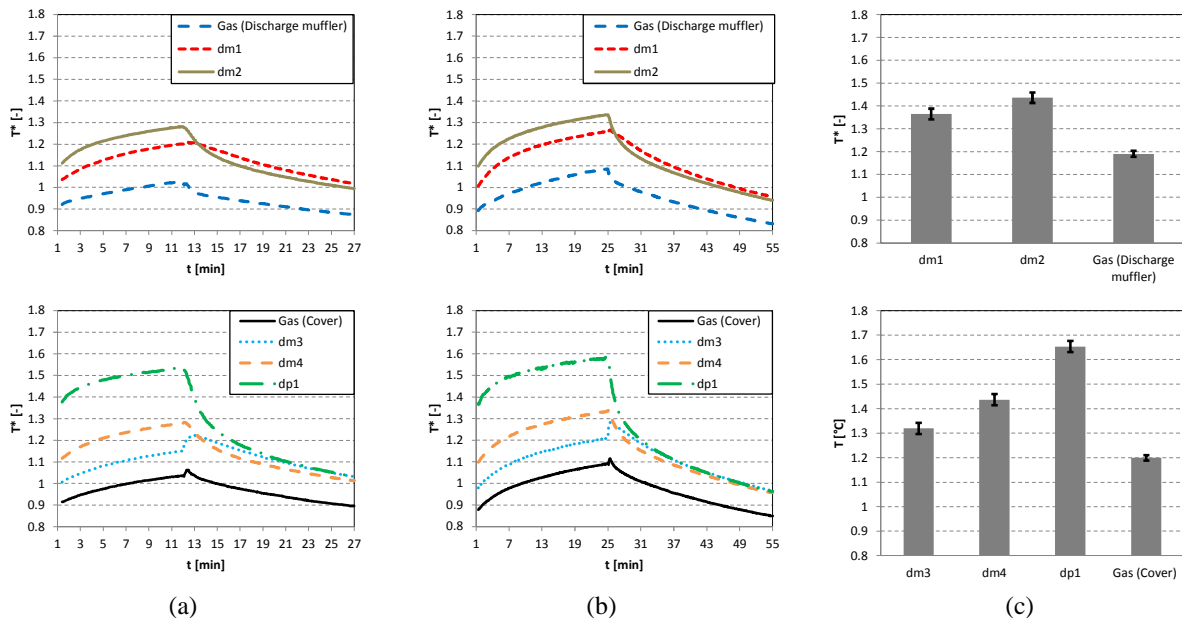


Figure 13: Temperatures at the discharge system wall: (a) 12 min/15 min; (b) 25 min/30 min; (c) steady state.

The heat transfer enhancement mechanism at the surfaces dm3 and dm4 due the crankshaft spinning suddenly stops when the compressor is switched off (Figures 14 and 15). During the idle period, the heat transfer coefficients for all the compressor surfaces become similar, within a range 0.8 – 1.5. This is a further evidence of the influence of the flow of lubricant oil on the heat transfer in the discharge muffler.

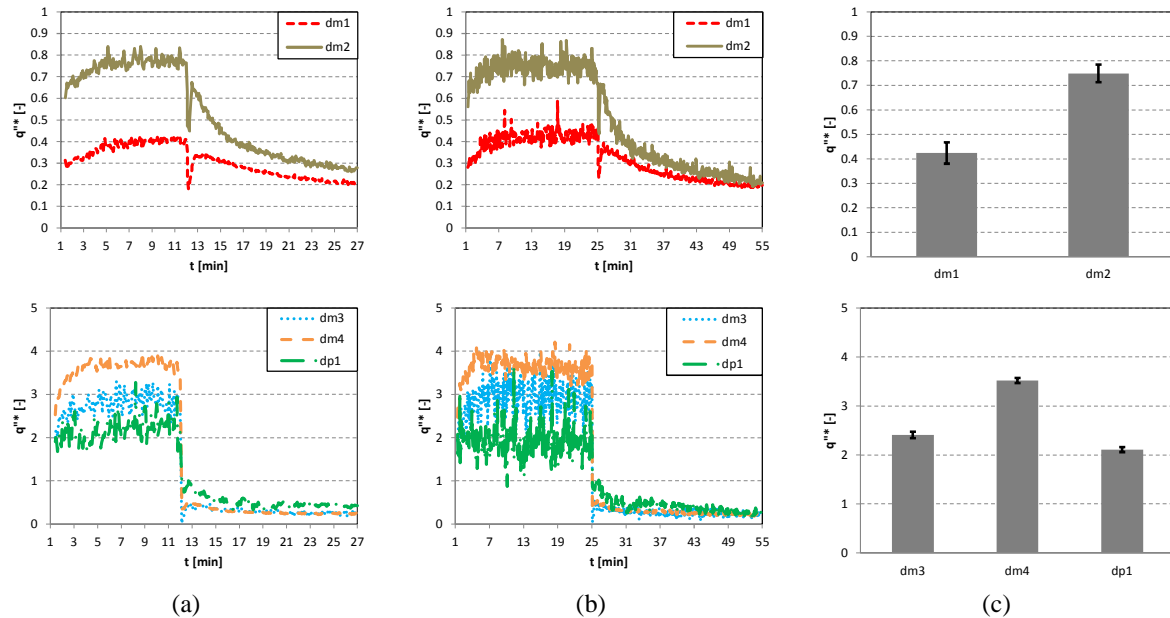


Figure 14: Heat flux at the discharge system wall: (a) 12min/15 min; (b) 25min/30 min; (c) steady state.

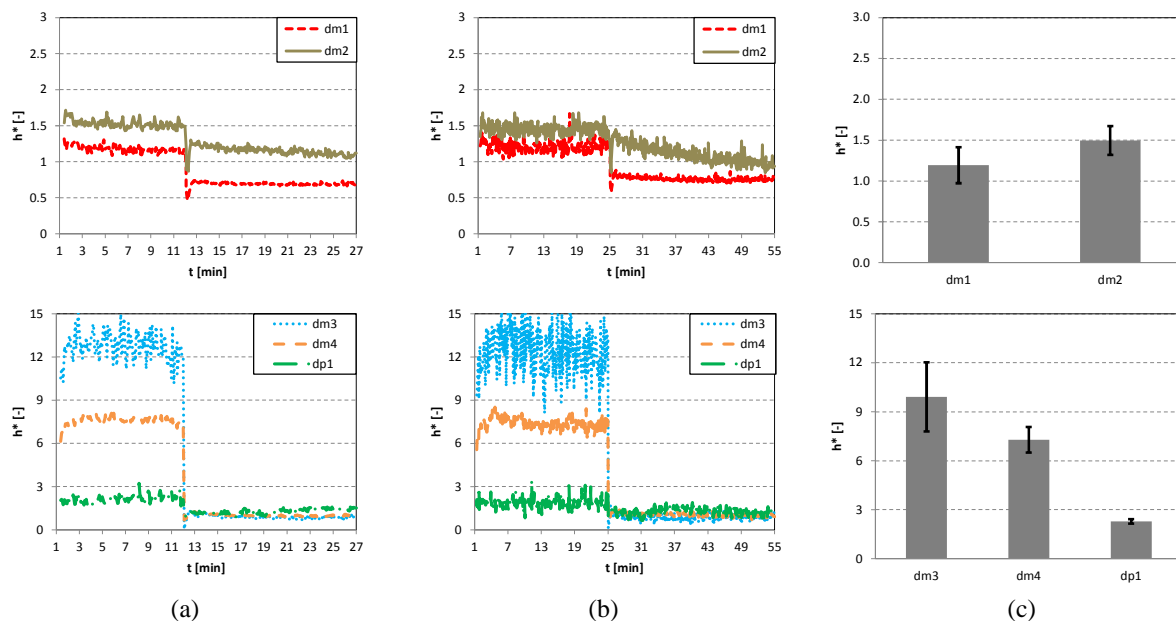


Figure 15: Heat transfer coefficients at the discharge system wall: (a) 12 min/15 min; (b) 25 min/30 min; (c) steady state.

3.5 Comparison of Temperatures and Heat Fluxes between Different Regions

Figure 16 illustrates the temperature difference ($T_s - T_\infty$) at several components of the compressor at the cyclic condition 25 min/30 min. These measurements show the presence of different time scales associated with the heat transfer process in each component. It is interesting to note that the temperature difference between the discharge

system components and the refrigerant gas is nearly constant during the ON period, inducing roughly constant heat fluxes (Figure 14), and which do not change regardless the operating condition, as shown in Table 1 for the surfaces dm1 and dp1. On the other hand, the temperature difference between the suction muffler (sm1) surface and the gas increases continuously, bringing about an increase of the heat flux at different operating conditions (Table 1). Moreover, the temperature difference between the electrical motor (em1) and the refrigerant gas first decreases and then increases. Such a slight variation do not affects the heat flux during the cyclic operation, as shown in Table 1. However, if the compressor is kept under operation for a longer period (i.e. until reaching the steady-state condition), the temperature difference continues to increase, bringing about a higher heat flux. It is worthwhile to note from Table 1 that the average heat fluxes at the surfaces sm1 and em1 are 50% higher in the steady-state condition in comparison to the ON period of the 12 min/15 min. This highlights the importance of taking into account thermal transients in analyses of compressor performance.

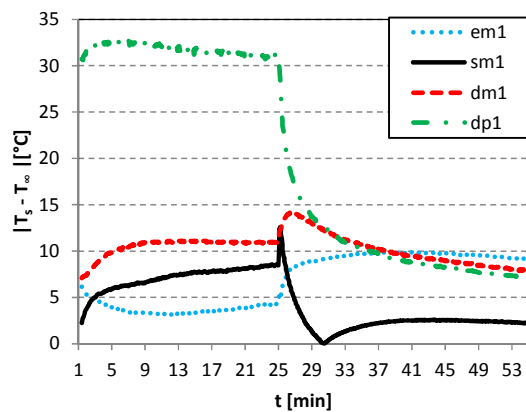


Figure 16: Temperature difference at different components for the 25 min/30 min operating condition.

Table 1: Average heat flux for cyclic and steady state operating conditions.

| | Mean heat flux [-] | | |
|-----|--------------------|-------------|--------------|
| | 12min/15min | 25min/30min | Steady State |
| em1 | 0.23 | 0.23 | 0.36 |
| sm1 | 0.39 | 0.44 | 0.60 |
| dm1 | 0.38 | 0.41 | 0.42 |
| dp1 | 2.17 | 1.93 | 2.11 |

4. CONCLUSIONS

The present paper reported an experimental investigation of heat transfer in a hermetic reciprocating compressor during cyclic and steady-state operating conditions. The compressor was instrumented with heat flux sensors and thermocouples fitted onto the surfaces of the suction muffler, electrical motor and discharge system. The results revealed the presence of two levels of heat transfer coefficients during the compressor operation: (i) 0.8 – 3.5 at most components and (ii) 7 - 12 at the surface of components closest to the crankshaft, brought about by forced convection of lubricant oil and refrigerant gas provided by the crankshaft motion. Furthermore, we found different time scales associated with the heat transfer process of each component, and the heat flux at the suction system and electrical motor during the cyclic condition are quite different of those for the steady state condition. This is an aspect that should be taken into account in the thermal modeling of compressors.

NOMENCLATURE

| | | | | |
|------|---|-----------------------|------------------|-----------|
| h | heat transfer coefficient | (W/m ² .K) | Subscript | |
| h* | non-dimensional heat transfer coefficient | (-) | s | surface |
| q'' | heat flux | (W/m ²) | shell | shell |
| q''* | non-dimensional heat flux | (-) | ∞ | reference |
| T | temperature | (°C) | | |
| T* | non-dimensional temperature | (-) | | |
| em | electrical motor | (-) | | |
| dm | discharge muffler | (-) | | |
| dp | discharge plenum | (-) | | |
| sm | suction muffler | (-) | | |

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