THE ADOPTION OF HEAT FLUX SENSOR TO ASSESS HEAT LOSS IN REFRIGERATION COMPRESSORS

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Abstract. *Arguably, the use of thermocouples is the most traditional and consolidated technique to characterize refrigeration compressors, allowing one to estimate heat flux in specific components from energy balances. Additionally, thermocouples can also indentify high temperature components that can critically affect the machine efficiency. However, such devices are unable to offer a complete analysis of heat transfer between the components that is required, for instance, in the development of more efficient layouts. Heat flux sensors have been increasingly employed in different applications, such as agrometeorology, material processing, building construction and refrigeration. The present paper gives an account of a study in which heat flux sensors were adopted to thermally characterize a reciprocating compressor. Details of the instrumentation and difficulties associated with the measurements are also presented.*

Keywords: *Thermal characterization, instrumentation, heat flux sensor.*

1. INTRODUCTION

The efficiency of refrigeration compressor is strongly related to its thermal behaviour. Therefore, it is very important to understand the thermal interaction between the machine components, by estimating heat transfer in specific components. This task has been traditionally accomplished by using thermocouples for temperature measurements, from which heat flux in each component can be estimated from energy balances. However, when heat flux has to be locally determined on a certain component, energy balances cannot be used. Additionally, the instrumentation of thermocouples for heat transfer analysis is, sometimes, a difficult task.

The first proposals of a heat flux sensor (HFS) appeared fifty years ago and since then HFS's have been increasingly employed in different areas. For instance, HFS is commonly adopted for agrometeorological applications (Silberstein *et al.*, 2001; Borges *et al.*, 2008), where energy balances at the soil/atmosphere interface is made possible from heat flux measurements through HFS's buried into the soil. Sabau and Wu (2007) applied HFS's for material processing measurements of heat flux associated with the lubricant spray cooling of die casting process. HFS's have been also used to characterize heat transfer between human body and environment under different postures (Kurazumi *et al.,* 2008), taking into account contributions from thermal radiation and convection. In the building construction sector HFS's were adopted to quantify solar heat gain through fenestrations (Marinoski *et al.*, 2006). Some investigations with HFS's are also available in refrigeration applications (Silva, 1998; Seidel, 2001), aimed at measuring cooling capacity and heat transfer through different parts of a household refrigerator cabinet.

Despite the aforementioned scenario about the application of HFS's for different interests, a literature review shows a single study of reciprocating compressors with heat flux sensors. Shiva Prasad (1992) presented measurements for instantaneous heat flux inside the compression chamber of a 900 rpm reciprocating air compressor. The major aim of his research was to verify the influence of regenerative heat transfer on the compressor volumetric efficiency.

The present paper is concerned with the application of HFS definition to characterize heat transfer in a small reciprocating compressor adopted for refrigeration. Additionally, the paper also provides details about the working principle, required instrumentation and major difficulties associated with HFS's. Results for heat flux and heat transfer coefficients are presented to allow a better understanding of thermal phenomena in the compressor.

2. HEAT FLUX SENSOR

Heat flux sensors are sensors which supply a self-generated output voltage proportional to a heat flux excitation. A typical HFS is built from thermocouples in a serial association, as can be depicted in Fig. 1. The voltage output of a thermopile, E [V], is a function of the thermoelectric sensitivity difference of the materials, S_T [V/K] , the number of thermocouple junctions, N, and the temperature difference between the HFS surfaces, ΔT [K], ($E = NS_T\Delta T$). On the other hand, the one-dimensional steady-state heat flux perpendicular to the HFS surfaces, q'' [W/m²], depends on the HFS equivalent thermal conductivity, k [W/mK], the HFS thickness, t [m], and the ΔT, ($q'' = k\Delta T/t$). Thus, by combining the aforementioned equations, one obtains a linear relation between heat flux and voltage output:

$$q" = \frac{E}{S} \tag{1}$$

where,

$$S = NS_T \frac{t}{k} \tag{2}$$

with S being the sensitivity [V/W/m²] associated with a certain HFS.

Eq. 2 indicates that one can directly calculate S, but this procedure is not usually done. In fact, it is not an easy task to determine with accuracy the values of S_T and k. Thus, the HFS sensitivity is commonly obtained from a calibration procedure, which can be performed under different layouts (Holmberg and Womeldorf, 1999). Most of the HFS's employed in the present work were commercially supplied with their sensitivity values. For those HFS's without indication of their sensitivity, the calibration procedure proposed by Güths and Nicolau (1998) was employed.

HFS's were installed on the external surface of the compressor shell by using only thermal grease and an aluminum tape. During the sensor fixation, care has to be taken to avoid the presence of air bubbles between the HFS and the surface, as illustrated in Fig. 2, otherwise deviation of heat flux lines may introduce errors into the measurement. Thus, thermal grease has to be applied between the surface and the HFS surface to guarantee a uniform distribution of heat flux lines. However, the thermal grease layer should be as thin as possible in order to prevent a significant increase of the thermal resistance, which would also negatively affect the measurement.

When thermal radiation is significant, it is necessary to know the emissivity of both the surface and the HFS's in order to evaluate the required correction. The aluminum tape used to attach the HFS's to the external surface of the compressor shell has an emissivity of approximately 0.2, whereas the shell surface emissivity is 0.9. Considering the compressor shell and the HFS as gray bodies, the thermal radiation flux can be computed as:

$$q"_R = \sigma\varepsilon\left(T_S{}^4 - T_V{}^4\right) \tag{3}$$

where σ is the Stefan-Boltzmann constant [W/m².K^4], ε is the emissivity [-], T_S and T_V are the surface and the vicinity temperatures [K], respectively. The compartment walls in which the compressor is placed for testing acts as a vicinity with temperature T_V.

Considering the aforementioned aspects, the heat flux at the external surface of the compressor shell is expressed as follows:

$$q"_S = q"_{HFS} - q"_{R\_HFS} + q"_{R\_S} \tag{4}$$

where $q"_{HFS}$ is the heat flux indicated by the HFS, whereas $q"_{R\_HFS}$ and $q"_{R\_S}$ are the radiation heat fluxes estimated for the HFS and the external surface of the compressor shell surface, respectively. By combining Eq. 3 and Eq. 4, one obtains the final expression for the heat flux at the external surface of the compressor shell:

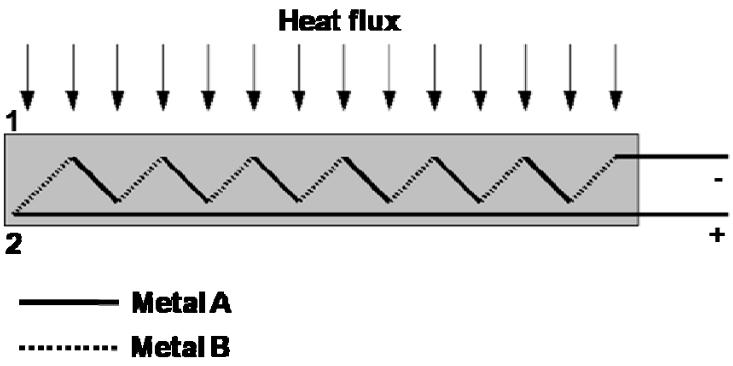
$$q"_S = q"_{HFS} + \sigma\left(\varepsilon_S - \varepsilon_{AT}\right)\left(T_S{}^4 - T_V{}^4\right) \tag{5}$$

where the subscript "AT" refers to the aluminum tape. The external surface of the compressor shell was instrumented with several HFS's (Fig. 3a) and then, from measurements and Eq. 5, estimates for local heat flux could be obtained.

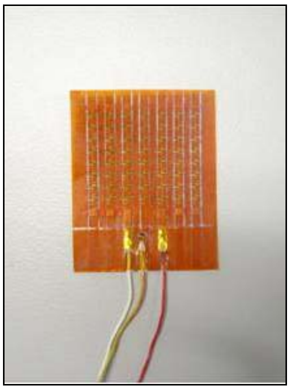
HFS's were also installed on the internal surfaces of the compressor shell (Fig. 3b). In this case, a cement epoxy-adhesive had to be employed, since the presence of high temperature levels and lubricating oil did not allow the use of thermal grease. The thermal radiation heat flux correction applied to the external surface of the compressor shell could not be carried out for the internal measurements due to the difficulty of determining the emissivity of the surfaces, especially for those in which as oil film is present. However, convective heat transfer is the dominant mechanism in many of the internal components, therefore reducing the importance of a correction due to thermal radiation.

The positioning of the HFS's wiring inside the compressor had to be carefully treated, so as to minimize changes in the lubricating oil flow path. This is a very important aspect, because the lubricating oil has a strong effect on the heat transfer process inside the compressor. The flow of oil inside the compressor is promoted by a pump, which collects oil stored in the carter and, by centrifugal action, takes it to the upper parts of the compressor. Figure 4 presents a three-dimensional schematic view of the compressor, with the identification of some of its main components, including the oil pump.

Other relevant issue is the variation of the HFS sensitivity with temperature. Therefore, when measurements are carried out at a temperature different from that in which the HFS was calibrated, a correction is required, since deviations can easily reach more than 5%. Generally, commercially available HFS's provide a temperature-sensitivity correction curve.

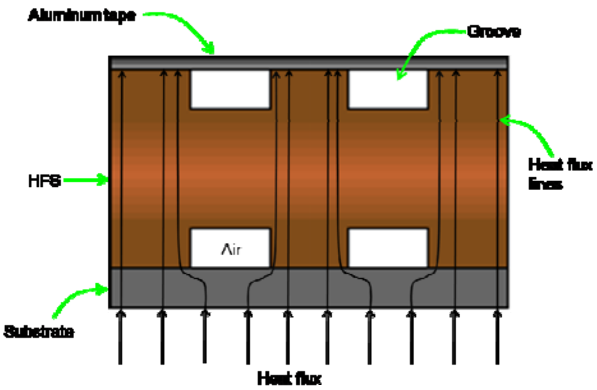


(a)



(b)

Figure 1. (a) HFS cross-section schematic (Hukseflux, 1999) in which 1 and 2 refer to the HFS hot and cold surfaces, respectively. (b) Typically commercially available HFS.



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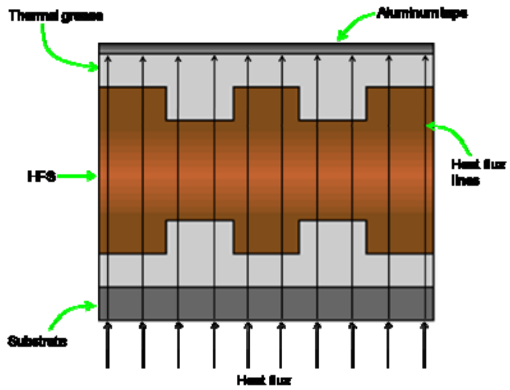
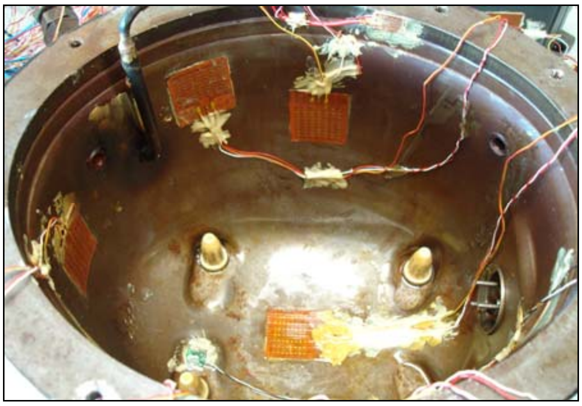


(b)

Figure 2. HFS instrumentation (a) without and (b) with thermal grease.

(a)



(b)

Figure 3. HFS assembled on the external (a) and internal (b) shell wall.

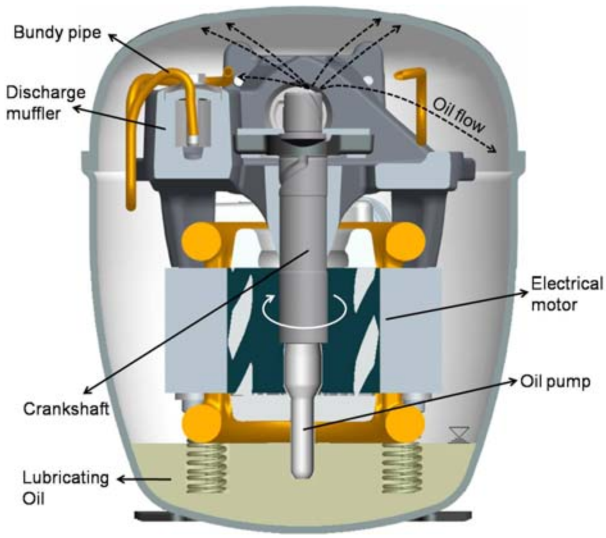


Figure 4. Compressor schematic view.

3. RESULTS AND DISCUSSION

A reciprocating compressor operating with R134a was selected for the analysis and submitted to different operating conditions, given by two pairs of evaporation and condensation temperatures: (-23.3ºC/40.5ºC) and (-10.0ºC/90.0ºC). All tests were conducted in a calorimeter facility following refrigeration test conditions established by standards. The uncertainties associated with measurements in the calorimeter are $\pm$ 2% for mass flow rate and power consumption. Further details of the experimental facility can be found in Dutra (2008). The compressor was tested five times for each operating condition. In what follows, results for heat flux and heat transfer coefficient are presented with an uncertainty bar corresponding to a 95% confidence interval.

Heat transfer coefficients were estimated from measurements of heat flux and temperature, the latter obtained with thermocouples installed in the compressor internal and external ambient. Therefore, suitable reference temperatures, T_∞, could be established for estimates of local heat transfer coefficient, h, in each region, i.e.:

$$h = \frac{q''}{T_S - T_\infty} \tag{6}$$

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

3.1. Experimental validation

By applying an overall energy balance to the compressor (Fig. 5a), the heat loss through the compressor shell, $\dot{Q}_C$, can be determined:

$$\dot{Q}_C = \dot{W}_C - \dot{m}\left(h_{DIS} - h_{SUC}\right) \tag{7}$$

where $\dot{W}_C$ is the compressor power consumption [W], $\dot{m}$ is the refrigerant mass flow rate [kg/s] and h_{SUC} and h_{DIS} are the refrigerant specific enthalpies at the suction and discharge lines [J/kg], 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.

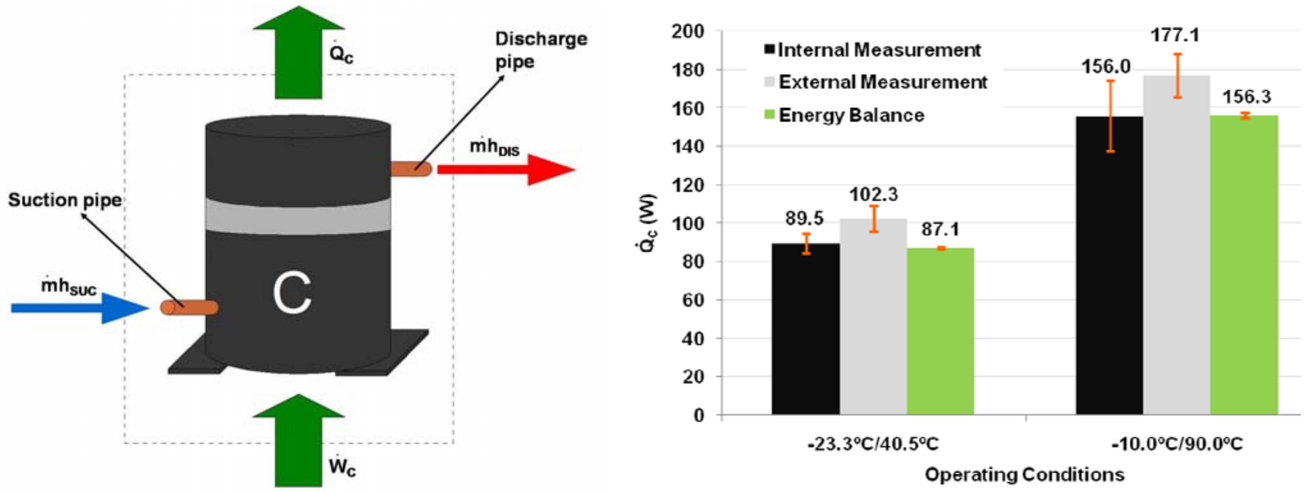
The heat transfer rate rejected through the compressor shell can be also evaluated by summing up contributions of local heat fluxes measured in different regions of the shell. Hence,

$$\dot{Q}_C = \sum_{i=1}^{n} q''_i \, A_i \tag{8}$$

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 Eq. (4) are obtained from a CAD model. Each region of the shell is represented by a heat flux measurement of a HFS.

Results for $\dot{Q}_C$ obtained from energy balance and integration of heat fluxes are presented in Fig. 5b for the two aforementioned compressor operating conditions. As Fig. 5b reveals, there is a satisfactory agreement between both results, with the larger deviation (approximately 15%) occurring for the external surface.

A number of uncertainty sources are present in the measurements. As far as the heat flux sensor is concerned, the manufacturer specifies an accuracy of 5% for the HFS sensitivity. Random errors associated with instrumentation were treated by repeating five times each test condition and then applying a 95% confidence interval to the measurement (Dutra, 2008). Thermal radiation is also an issue since there is a difficulty to characterize the emissivity of surfaces, especially inside the compressor where oil is present on almost all surfaces. Yet, for the external surface, infrared thermography was combined with thermocouples to quantify the emissivity. Finally, the regions chosen to provide data for Eq. (8) must be in enough number to properly discretize the heat transfer on both surfaces of the compressor shell. Therefore, for the purpose of energy balance, local measurements of heat transfer are not a trivial task since one should have information about the thermal field before installing the heat flux sensors. Naturally, if the aim is to obtain only local values for heat transfer, then this issue is not relevant. Considering all the aforementioned difficulties in this type of measurement, the level of agreement between estimates for $\dot{Q}_C$ obtained from Eq. (7) and Eq. (8) is considered acceptable.



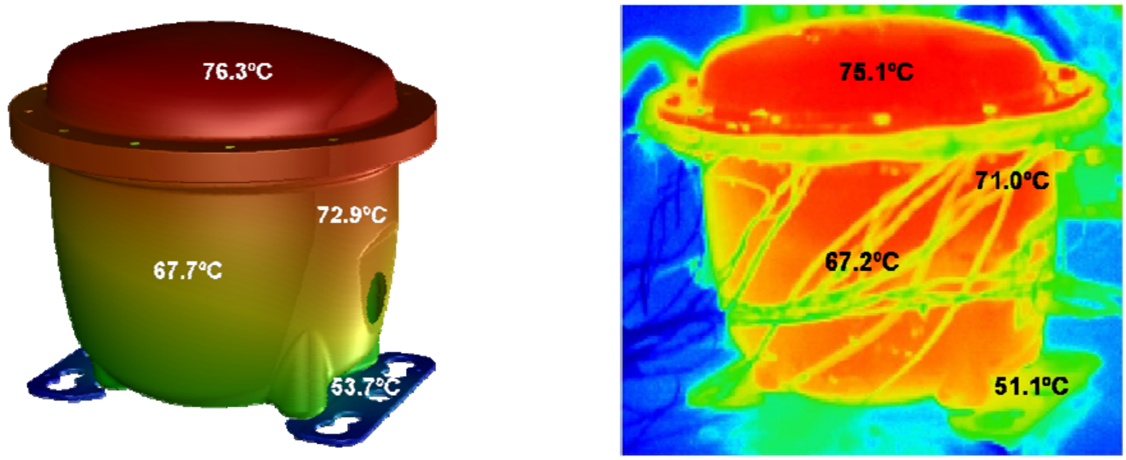
(a) (b)

Figure 5. (a) Compressor overall energy balance and (b) compressor heat transfer rate for two operating conditions.

Heat conduction in the compressor shell was simulated with the commercial code FLUENT 6.3 (Ansys Inc, 2008), which adopts a finite volume methodology to discretize the governing equations. Boundary conditions of local convective heat transfer coefficient and ambient temperature for both internal and external surfaces of the compressor shell were prescribed with reference to measurements. A three-dimensional model of the compressor shell was made available from a CAD software. Two computational grids were tested for assessment of truncation errors, one with 1.3×10^6 and other with 2.7×10^6 volumes. Results for temperature [ºC] obtained with each grid were in an agreement within 1% in the whole domain and, therefore, the numerical solution was considered to be representative of the heat conduction equation.

Figure 6 shows a comparison between results for temperature field on the external surface of the compressor shell via numerical prediction and from infrared thermography combined with thermocouples. Good agreement is seen between the experimental and numerical results, with an average deviation of 2%. The greatest difference was found at the base plate, corresponding to approximately 5% (2.6°C). Such a level of agreement provides further evidence about the measurement accuracy.

Figure 7 depicts experimental and numerical results of local heat flux on the external surface of the compressor shell and again good agreement is observed for the different regions on the external surface, depicted in Fig. 8a. A typical deviation of 2% is found between the results, whereas the major discrepancy (approximately 12%) occurs in the compressor base plate (referenced as *e6*).



(a) (b)

Figure 6. Temperature field on the compressor shell (a) numerical prediction; (b) measurements.

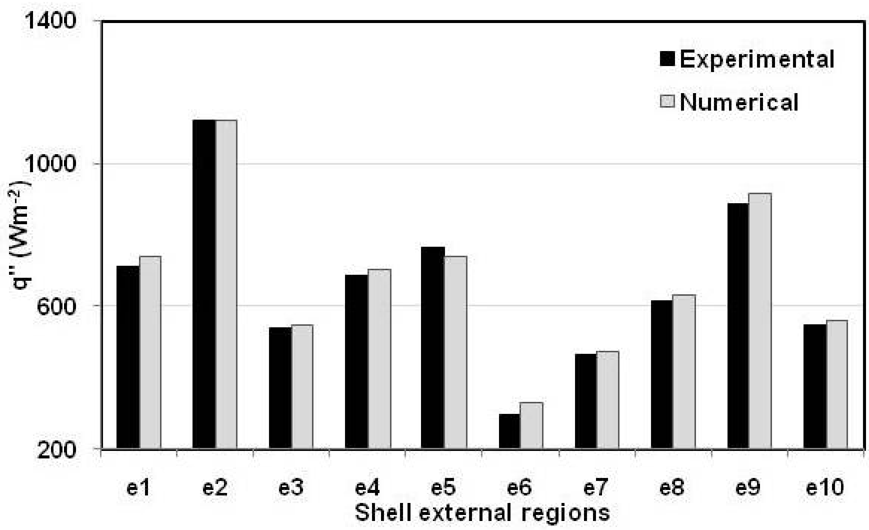


Figure 7. Experimental and numerical results for heat flux on the shell external surface.

3.2. Local heat transfer analysis

Figure 8 presents results for heat transfer related to the operating condition -23.3ºC/40.5ºC. It can be noticed relatively low levels of heat transfer coefficient varying between 10 and 40 W/m²K. This is an expected result since the compressor was tested inside a temperature controlled compartment, in which low velocity air stream is maintained.

However, the air flow over the compressor shell is not uniform and this can be clearly noticed in Fig. 8c, where some regions of the shell like *e2*, *e4*, *e5* and *e9*, present higher levels of heat transfer coefficient due to a more intensive exposure to the air flow stream. As a consequence, higher levels of heat flux are evident in these regions (Fig. 8b).

Due to low heat transfer coefficients in some regions, associated with high temperature difference between compressor shell and its vicinity, thermal radiation becomes an important heat transfer mechanism. The vicinity temperature, represented by T_V in Eq. (3), is around 30°C whereas the compressor shell temperature varies between 60°C and 70°C, depending on the region. Following the methodology presented in Section 2, thermal radiation heat transfer were estimated for the shell regions and compared with the total heat loss in each one. Results in Fig. 9 show that thermal radiation corresponds to more than 30% of the total compressor heat loss.

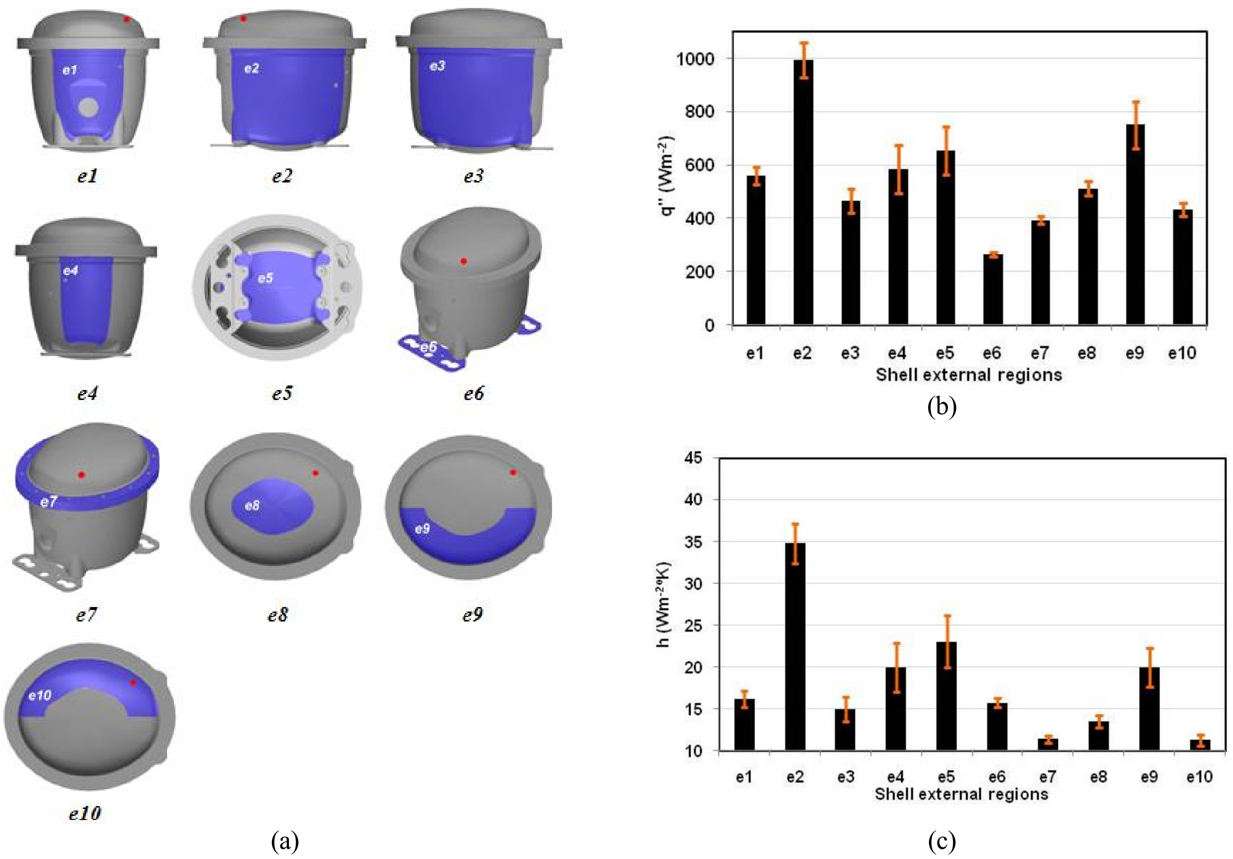


Figure 8. (a) Measurements regions on the shell external surface; (b) local heat flux; (c) local heat transfer coefficient.

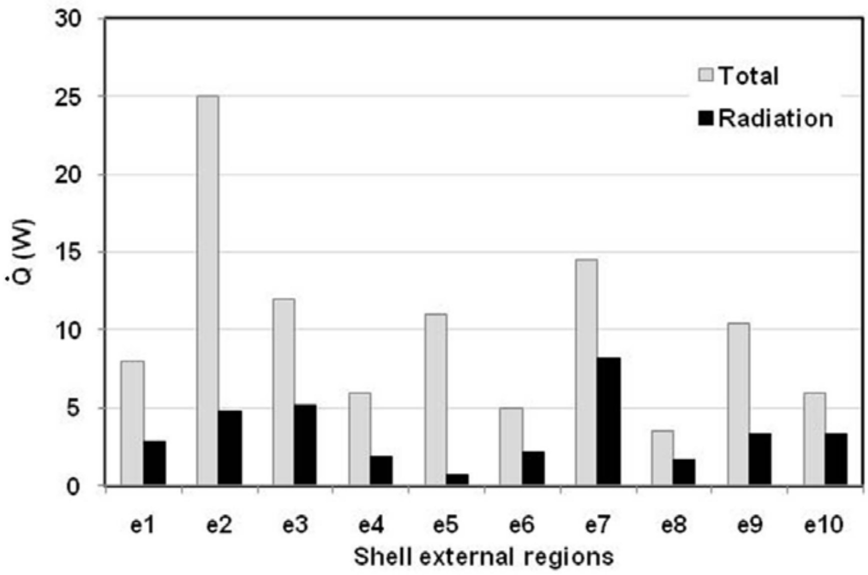


Figure 9. Total and thermal radiation heat transfer in regions of the shell external surface.

4. CONCLUSIONS

This paper reported the use of heat flux sensors (HFS's) to thermally characterize a reciprocating compressor. Although such sensors have been increasingly employed in several applications, they are not commonly used for measurements in compressors. A number of uncertainty sources were identified, such as the presence of thermal radiation, sensor wiring and thermal resistance between sensor and substrate. In the present study, experimental data were validated through an overall energy balance applied to the compressor, complemented with numerical simulations based on boundary conditions obtained from measurements.

5. ACKNOWLEDGEMENTS

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