Experimental Evaluation of the Heat Transfer Through the Walls of Household Refrigerators

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ABSTRACT

This work presents an experimental analysis of heat transfer paths from the surroundings to the interior food compartment of a 230 liters all refrigerator under closed door conditions. The overall refrigerator thermal conductance was determined from conventional reverse heat loss rate measurements as well as from unconventional measurements taken with specially manufactured heat flux sensors. The heat transfer rate through each of the various pathways (walls, door, gasket, etc.) was also quantified.

INTRODUCTION

In response to the concerns of the global community over greenhouse emissions, efforts are being made to produce refrigerators with low energy consumption (Vineyard et al., 1998). The conventional approach is to improve the thermodynamic performance of each one of the components (evaporator, condenser, etc.) as well as the overall thermodynamic performance of the refrigeration unit. However, the energy efficiency of refrigerators not only depends on the thermodynamics of the refrigeration cycle but also on the heat transfer characteristics of the refrigerator cabinet and door gaskets (Hessami, 1993). Knowing that the heat exchange from the surroundings to the interior of the refrigerator accounts for a significant amount of the compressor power, its minimization through better insulation and gaskets improves the unit's energy efficiency.

The heat exchange between the inside and outside of the refrigerator is characterized by the refrigerator overall thermal conductance, \( UA \), usually determined from reverse heat loss rate measurements (NTB00119, 1992). This kind of measurement, however, does not distinguish the heat fluxes through the refrigerator walls, door and gasket.

The work effort reported herein focuses on a similar experimental technique to evaluate the refrigerator overall thermal conductance but using heat flux sensors adequately distributed over the refrigerator external surfaces. This technique provides overall results comparable to the ones given by the reverse heat loss rate tests and also provides enough data to quantify the heat fluxes through each one of the various heat transfer paths.

REVERSE HEAT LOSS RATE TESTS

The procedure for measuring heat loss rate involves placing a refrigerator in a climated chamber with airflow, humidity and temperature fluctuations within the specifications of the ISO standard 7371, 1987. A controlled electric heater is employed to maintain desired temperatures in the cabinet (see Figure 1). Cabinet temperatures and power measurements along with ambient temperatures, are recorded as the cabinet temperatures achieve desired levels. Once the cabinet temperatures achieve steady state, data are averaged for a 30-minutes interval to determine the overall heat loss rate.

The overall refrigerator thermal conductance, \( UA \), is then evaluated from the following expression:

\[
UA = \frac{\dot{q}_{\text{heater}}}{(T_1 - T_A)} = \frac{\dot{q}_{\text{walls}}}{(T_1 - T_A)}
\]  

The cabinet temperature, \( T_1 \), corresponds to the arithmetic mean of the readings of the thermocouples T1, T2, T3, T4, and T5 while the ambient temperature, \( T_A \), corresponds to the arithmetic mean of the readings of the thermocouples TT, TD, TF, and TE, placed 10 cm from the center of the refrigerator walls (see Figure 2).
Three reverse heat loss rate tests were performed under different cabinet and ambient temperatures but maintaining a temperature difference of approximately 25°C (NTB00119, 1992). The results of these tests are shown in Table 1.

Table 1 – UA values

<table>
<thead>
<tr>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{q}_{\text{walls}} = 42.1$ W</td>
<td>$\dot{q}_{\text{walls}} = 45.5$ W</td>
<td>$\dot{q}_{\text{walls}} = 46.5$ W</td>
</tr>
<tr>
<td>$\bar{T}_{A} = 42.7$ °C</td>
<td>$\bar{T}_{I} = 50.0$ °C</td>
<td>$\bar{T}_{I} = 56.9$ °C</td>
</tr>
<tr>
<td>$\bar{T}_{A} = 18.5$ °C</td>
<td>$\bar{T}_{A} = 25.2$ °C</td>
<td>$\bar{T}_{A} = 31.7$ °C</td>
</tr>
<tr>
<td>$\text{UA}=1.74 \pm 0.01$ W/°C</td>
<td>$\text{UA}=1.83 \pm 0.01$ W/°C</td>
<td>$\text{UA}=1.85 \pm 0.01$ W/°C</td>
</tr>
</tbody>
</table>
HEAT FLUX METERS

Most of the commercial heat flux meters in use are based on monitoring a temperature differential across a thermal barrier layer. Fourier's one-dimensional law of conduction can then be used to determine the heat flux through the medium provided the thermal properties are known. A variation on the traditional heat flux sensor concept is the planar thermal gradient heat flux sensor where the temperature difference is measured across two locations on the substrate surface.

Very thin (~200 μm) and very sensitive (20μV/W/m² for a 50x50 mm sensor) planar thermal gradient heat flux sensors, developed by Güths, 1994, have been used in the present work.

In these sensors a constantan strip is placed on a kapton support with copper deposits on it. Periodic volumic gaps etched into the copper plate allow an asymmetrical constriction of any heat flow line reaching the sensor surface. Therefore, the copper/constantan plated thermoelectrical circuit is subject to tangential thermal gradients. So the heat or cooling rate can be directly read out as a function of voltage. When these thermoelectric elements are placed in series, an electrical motive force proportional to the thermal flux is created.

Each sensor has been calibrated according to the method described by Lassue et al., 1993.

![Planar heat flux sensor](image)

**Figure 3 – Planar heat flux sensor**

EXPERIMENTS USING HEAT FLUX METERS

The heat exchange between the inside and outside of the refrigerator was also evaluated using specially manufactured heat flux meters (Güths et al., 1994). Thirty-one heat flux sensors were installed on the test unit; twenty-five (50x50mm) on the refrigerator walls and six (1.8x15.4cm) on the gasket (see Figure 4). Tests were performed with cabinet temperatures higher and lower than the ambient temperatures.

**Cabinet temperatures higher than the ambient temperatures**

During this experiment the average cabinet temperature was kept around 50°C while the average ambient temperature was maintained around 25°C, resulting in a temperature difference of approximately 25°C. Once the temperatures achieved steady state, data were recorded through a computerized data acquisition system and averaged over a 30 minutes interval.

From equation (1), and with \( \dot{q}_{heater} = 45.2 \text{ W} \), \( T_i = 50.4 \text{ °C} \), \( T_f = 25.4 \text{ °C} \), a value of 1.81 W/°C was found for the overall cabinet thermal conductance. This value is quite close to the one shown in Table 1 for test # 2.
The heat flux through each one of the sensors, $q_i^*$, was determined from the following equation:

$$q_i^* = \frac{C_i V_i}{A_i}$$  \hspace{1cm} (2)

where $C_i$ is the calibration constant [W/mV]; $V_i$ is the voltage [mV] and $A_i$ is the surface area of each sensor [m$^2$].

The average heat flux through each wall corresponds to the arithmetic mean of the readings of the sensors shown in Figure 4. Table 2 shows the heat flux, $q^*$, the heat transfer rate, $\dot{q}$, and the percentages of the total heat transfer rate for each pathway.

Table 2 – Heat flux and heat transfer rate distribution ($\overline{q_i} > \overline{T_A}$)

<table>
<thead>
<tr>
<th>Heat transfer path</th>
<th>Area [m$^2$]</th>
<th>$q^*$ [W/m$^2$]</th>
<th>$\dot{q}$ [W]</th>
<th>% of total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Door</td>
<td>0.6721</td>
<td>12.38</td>
<td>8.32</td>
<td>19.7</td>
</tr>
<tr>
<td>Top wall</td>
<td>0.2397</td>
<td>11.53</td>
<td>2.76</td>
<td>6.5</td>
</tr>
<tr>
<td>Side wall</td>
<td>0.6885</td>
<td>12.90</td>
<td>8.88</td>
<td>21.0</td>
</tr>
<tr>
<td>Side wall (*)</td>
<td>0.0799</td>
<td>20.40</td>
<td>1.63</td>
<td>3.9</td>
</tr>
<tr>
<td>Bottom wall</td>
<td>0.5593</td>
<td>11.66</td>
<td>6.52</td>
<td>15.5</td>
</tr>
<tr>
<td>Top wall (*)</td>
<td>0.1128</td>
<td>13.72</td>
<td>1.68</td>
<td>4.0</td>
</tr>
<tr>
<td>Gasket</td>
<td>0.0915</td>
<td>13.72</td>
<td>1.26</td>
<td>3.0</td>
</tr>
<tr>
<td>TOTAL [W]</td>
<td></td>
<td>42.22</td>
<td></td>
<td>100</td>
</tr>
</tbody>
</table>

(*) Compressor compartment

It can be seen that the heat flux is uniformly distributed over the refrigerator walls and gasket, with exception to the top wall of the compressor compartment. The higher heat flux through this surface is due to the electric heater that was installed on it. The heat transfer rate through the side walls, door and gasket account for 42%, 19.7% and 3% of the total heat transfer rate measured by the heat flux meters, respectively.
Comparing the total heat transfer rate measured by the heat flux meters (42.22 W) with the heat released by the electric heater (45.22 W) it is clear that there are heat transfer paths that were not taken into account. These heat transfer paths account for 3.0 W or 6.6% of the total heat transfer rate and are related to the conduction heat transfer along the door and wall steel frames (Boughton et al., 1996).

**Cabinet temperatures lower than the ambient temperatures**

During this experiment the cabinet temperature was kept below the ambient temperature by a refrigeration test facility developed by Silva, 1998 (see Figure 5).

![Figure 5 - Refrigeration test facility](image)

The facility runs with HFC-134a and consists of a hermetic reciprocating compressor (COMP), a water cooled condenser (COND), two evaporators (EVAPP, EVAPS), and three expansion devices (VEP, VES1, VES2). Two oil separators (SO1, SO2) and an oil filter (FO) are placed between the compressor and the condenser. The pressure in the condenser is established by the water flow, which is controlled by a pressure regulating valve. A subcooler (SRL) and an electric heater (RA1) are used to fine tune the refrigerant temperature at the expansion valve inlet. The evaporating pressure is controlled by the aperture of the expansion valve VEP. The other two expansion valves (VES1 and VES2), bypass the desired amount of refrigerant to control the system cooling capacity. The mass flow rate of the refrigerant is measured by a coriolis type mass flow meter (FM), with a maximum uncertainty of ± 0.03 kg/h. Strain gage pressure transducers (TA, TBA, TBD) are used to measure the absolute pressures, with a maximum uncertainty of ± 0.05 bar. The temperatures are measured by type T thermocouples (TPID, T1, T2), 0.13 mm in diameter, with a maximum uncertainty of ± 0.2 °C. The instantaneous power to the electric heater RA2 is measured by a power transducer whose accuracy is within ± 0.25% of the reading.

The heat transfer rate through the refrigerator walls was then calculated from the following equation:

\[ \dot{q}_{walls} = \dot{q}_{evap} - \dot{q}_{heater} \]  
\[ \text{(3)} \]

where \( \dot{q}_{evap} \) is the evaporator heat transfer rate.

The evaporator heat transfer rate was determined from the following energy balance:

\[ q_{evap} = \dot{m} (h_{in} - h_{out}) \]  
\[ \text{(4)} \]
where \( m \), \( h_{in} \) and \( h_{out} \) are the refrigerant mass flow rate and the specific enthalpies at the inlet and outlet of the evaporator, respectively.

The specific refrigerant enthalpies are determined from the pressure and temperature measurements at the inlet and outlet of the evaporator.

From equation (3) and (4), and with \( \dot{Q}_{\text{heater}} = 4.7 \text{ W} \), \( m = 1.16 \text{ kg/h} \), \( h_{out} = 254.0 \text{ kJ/kg} \) and \( h_{in} = 93.6 \text{ kJ/kg} \), values of 51.7 \( \text{W} \) and 47.0 \( \text{W} \) were found for the evaporator and wall heat transfer rates, respectively.

From equation (1) and with \( \dot{Q}_{\text{wall}} = 47.0 \text{ W} \), \( T_L = 4.7 \text{ °C} \), \( T_A = 31.2 \text{ °C} \), a value of 1.77 W/°C was found for the overall cabinet thermal conductance. This result is quite close to the one shown in Table 1 for test #3, which was performed for approximately the same ambient temperature \( T_A \).

The distribution of the heat flux and heat transfer rates between the various pathways followed the approach previously described. The results are shown in Table 3.

<table>
<thead>
<tr>
<th>Heat transfer path</th>
<th>Area [m²]</th>
<th>( q^* ) [W/m²]</th>
<th>( \dot{q} ) [W]</th>
<th>% of total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Door</td>
<td>0.6721</td>
<td>12.92</td>
<td>8.68</td>
<td>20.7</td>
</tr>
<tr>
<td>Top wall</td>
<td>0.2397</td>
<td>10.10</td>
<td>2.42</td>
<td>5.8</td>
</tr>
<tr>
<td>Side wall</td>
<td>0.6885</td>
<td>13.34</td>
<td>9.18</td>
<td>21.9</td>
</tr>
<tr>
<td>Side wall</td>
<td>0.6885</td>
<td>13.34</td>
<td>9.18</td>
<td>21.9</td>
</tr>
<tr>
<td>Back wall</td>
<td>0.5593</td>
<td>15.18</td>
<td>8.49</td>
<td>20.2</td>
</tr>
<tr>
<td>Bottom wall</td>
<td>0.1598</td>
<td>7.05</td>
<td>1.13</td>
<td>2.7</td>
</tr>
<tr>
<td>Top wall (*)</td>
<td>0.0799</td>
<td>11.70</td>
<td>0.93</td>
<td>2.2</td>
</tr>
<tr>
<td>Bottom wall (*)</td>
<td>0.1128</td>
<td>7.00</td>
<td>0.79</td>
<td>1.9</td>
</tr>
<tr>
<td>Gasket</td>
<td>0.0915</td>
<td>12.33</td>
<td>1.13</td>
<td>2.7</td>
</tr>
<tr>
<td>TOTAL [W]</td>
<td></td>
<td></td>
<td>41.93</td>
<td>100</td>
</tr>
</tbody>
</table>

(*) Compressor compartment

It can be seen that the heat fluxes through the bottom wall and through the compressor compartment walls are quite different from the values given in Table 2. This is directly related to the lower power released by the electric heaters during this test. The largest heat transfer rate was the one through the side walls (44%) and the smallest was the heat conduction through the bottom wall of the compressor compartment (1.9%).

Comparing the total heat transfer rate measured by the heat flux meters (41.93 W) with the wall heat transfer rate calculated from equation 3 (47.00 W) it may be also seen that there are heat transfer paths that were not taken into account.

These heat transfer paths account for 5.0 W or 10.7% of the total heat transfer rate and are also related to the conduction heat transfer along the door and wall steel frames (Boughton et al., 1996).

**CONCLUDING REMARKS**

Experimental and numerical works with the aim of improving the energy efficiency of household refrigerators are quite common in the literature. However, very few of these works pay the necessary attention to the heat transfer from the surroundings to the interior of the refrigerator. The usual practice is to estimate the heat transfer rates using empirical correlations and coefficients taken from the literature, that not always provides satisfactory results.

This work reports a unique methodology where the heat transfer rates from the outside to the inside of the refrigerator are evaluated using heat flux meters. This new technique is comparable, in global terms, to the results given by the reverse heat loss tests, but offers the advantage of quantifying the heat transfer rates through the various pathways.
The heat flux meters presented in this study are very sensitive and produce a negligible obstruction to the thermal flow. They have exhibited good potential for measuring the heat fluxes through the walls of household refrigerators.

It was observed clear that the heat transfer rates through the wall and door flanges account for a significant portion of the total heat transfer rate. Further numerical and experimental works have to be done to study this specific region of the refrigerator with greater care.

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