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HEAT AND MOISTURE LOADING OF A REFRIGERATOR CABINET DURING OPEN DOOR CONDITIONS

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ABSTRACT

Manufacturers have made substantial gains in energy efficiency over the last two decades. The gains have been based on several small, but cumulative effects including improved compressors, controls, heat exchangers, and cabinets. Continued improvement of refrigerators is dependent on tracking down all manners in which energy is transferred into a cabinet.

An experimental investigation of refrigerator cabinets is conducted using model cabinet test sections that cover a Rayleigh number range typical of domestic refrigerators. The “cabinets” are constructed from foamboard insulation with the interior covered with aluminum plates that act as calorimeters. A cavity is heated or cooled, then opened to the surrounding ambient air. Transient temperatures are recorded, which are then related to the heat transfer coefficient over the plate. For cooled cavities, when the cavity surface temperatures are below the ambient dewpoint temperature, the cavity weight is recorded. The addition of water condensation mass to the cavity is used to determine the overall cavity mass transfer coefficient.

NOMENCLATURE

A: area (m^2)
C: water concentration (kg/m^3)
C_p: specific heat (J/kg-K)
D_{ab}: mass diffusivity from A to B (m/s^2)
F: view factor
g: acceleration of gravity, 9.81 m/s^2
h: heat transfer coefficient (W/m^2-K) or (m/s)
h_{fg}: latent heat of vaporization (J/kg)
H: height of cavity (m)
j: diffusive mass flux (kg/s-m^2)
J: radiosity (W/m^2)
k: thermal conductivity (W/m)
L: width (m)
m: mass (kg)
$\dot{m}$: mass flux (g/s)
MW: molecular weight (kg)
Nu: Nusselt number
P: pressure (kPa)
q: heat transfer rate (W)
Ra: Rayleigh number
RH: relative humidity
T: temperature (°C)

Greek Symbols
α: thermal diffusivity (m^2/s)
β: thermal coefficient of thermal expansion (1/K)
β^*: thermal coefficient of expansion with concentration (m^3/kg)
ν: kinematic viscosity (m^2/s)

Subscripts
air: air
cond: conduction transfer
conv: convection transfer
exp: experimental
f: film properties
init: initial
mass: mass transfer
num: numerical
plate: AL calorimeter plate
wall: wall (or thickness) through which heat is being conducted
rad: radiation transfer
surf: surface

INTRODUCTION

The purpose of this research is to assess heat and moisture transport into a domestic refrigerator cabinet during open door conditions as well as determine sensible and latent refrigerator cabinet loading caused by objects removed and replaced into a refrigerator cabinet. The goal is to know how much water forms on the walls of the refrigerator when one opens the door. The air inside a refrigerator generally has lower water vapor pressure than the outside surroundings. When the door is opened, water vapor enters the cabinet; this water ends up on the evaporator in the form of frost. In addition to the energy load caused by moisture condensation, removal of the frost requires energy. The analytical and experimental study of heat/mass transfer in an open cavity is of interest not only in refrigerator cabinets, but also in other areas such as solar receivers, buildings, and electrical components.

CASE STUDY DESCRIPTION

Measurement Equipment

For this project, test cavities were designed using polished 6061-T6 aluminum plates to act as calorimeters (similar to work of M.R. Laleman (1992) and L.N. Knackstedt (1995)) to measure the heat transfer coefficient on the walls of the cavity. The dimensions of the aluminum plates are: 15.24 cm length, 15.24 cm width, and 0.3175 cm thick. Two holes were drilled in each plate at a 45° angle. Thermocouple wire (25 gauge copper-constantan) was inserted in each hole. Thermal epoxy was placed over the lead wires to hold the thermocouple bead in place and ensure good thermal contact between the wire and the plate. After the thermal epoxy cured, quick setting epoxy was then placed over the thermal epoxy for additional strength.

Three cavities were designed with various sizes (one plate per side, four plates per side, and nine plates per side). This allows a wider range of Rayleigh numbers to be investigated during testing of the cavities. The larger cavities also allow for the addition of "shelves" to simulate a refrigerator cabinet. The calorimeter plates were sealed to two-inch thick Styrofoam insulation with a thin layer of silicone gel. All plates were separated with 0.635 cm spacing to ensure no physical contact between plates. Silicone gel was also used between the plate spacing to smooth the surface.

Heat Transfer Testing on Cavity (no Mass Transfer)

A series of tests have been performed on the test cavities for comparison of results to those reported in the literature. Figure 1 shows different cavity orientations examined. The tests first involved placing a cover over the opening of the test cavity to seal the cavity from the ambient environment. Incandescent lights were positioned on the inside cover to warm the inside aluminum plates of the cavity. Fans were also located on the cover to help circulate air throughout the test section to create uniform temperature conditions. Once the desired cavity temperature was achieved, the cover was removed and the cavity was allowed to come in contact with the ambient air.

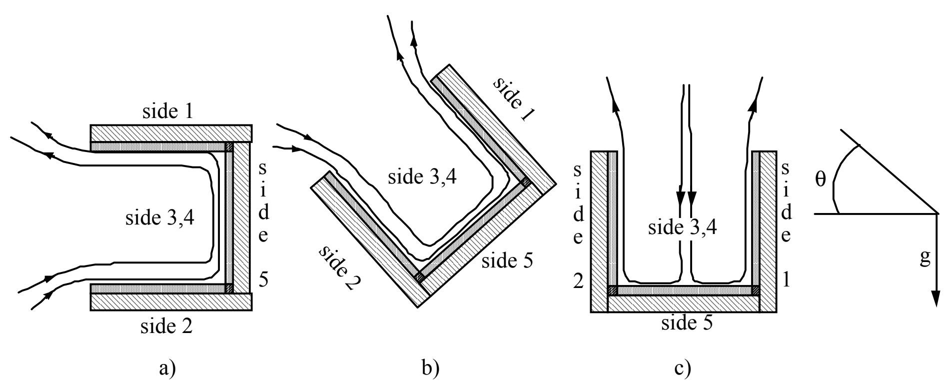

Figure 1: Various cavity angles of inclination: a) 0°; b) 45°; c) 90°.

The general energy balance for each plate in the cavity is:

$$m_{plate}C_{p,plate}\frac{dT_{plate}}{dt} = q_{conv} + q_{rad} + q_{cond} \tag{1}$$

where the capacity due to convective heat transfer is:

$$q_{conv} = h_{conv}A\left(T_{amb} - T_{plate}\right) \tag{2}$$

For conductive heat transfer through the wall insulation, Fourier's law is used:

$$q_{cond} = -kA\frac{dT}{dx} = \frac{k_{wall}}{L_{wall}}A\left(T_{surf} - T_{plate}\right) \tag{3}$$

For radiation, the view factor is determined for each plate to the surrounding plates and the ambient. The ambient surrounding is treated as a blackbody surface while the aluminum plates are assumed to be diffuse, gray surfaces. The net radiation transfer rate for each plate is determined as:

$$q_{rad} = \frac{E_{bi} - J_i}{\left(1-\varepsilon_i\right)/\varepsilon_i A_i} = \sum_{j=1}^{N}\frac{J_i - J_j}{\left(A_i F_{ij}\right)^{-1}} \tag{4}$$

where the radiosity (J_i) is determined from solving the simultaneous set of equations for each surface. The view factors are from Ehlert *et. al.*. Using equations 1 through 4, the only variable needed to solve for the convective heat transfer coefficient is dT/dt, which is determined experimentally. Temperatures of the plates are measured in thirty second or sixty-second intervals. Transient change of plate temperatures is based on least squares curve fit of four data points ahead of and behind each point. Tests end when temperature of the plates to reach close to ambient in order to obtain a complete temperature decay profile of cavity plates. Various levels of initial temperature differences between the cavity and surroundings were investigated.

Overall Moisture Loading in Open Cavity

For the combined heat and mass transfer analysis of the cavity as a whole, a container of liquid nitrogen is placed inside the test section to cool the aluminum plates (see Figure 2). A cover is then placed over the cavity to insulate the inside from the surroundings. Fans are also located on the cover to help circulate air throughout the test section to create uniform temperature conditions. The liquid nitrogen also dehumidifies the inside of the cavity, freezing moisture from the interior on the nitrogen container. The desired initial plate temperature is varied from 5°C to -5°C while the surrounding ambient relative humidity is varied from 50% and to 80%. Once the cavity plate temperatures reach the desired value, the cover is removed, allowing the cavity to come in contact with the ambient air. Ambient conditions as well as plate temperatures are recorded every minute. A scale is used to record the change in weight of the cavity due to condensation forming on the plates as time elapses.

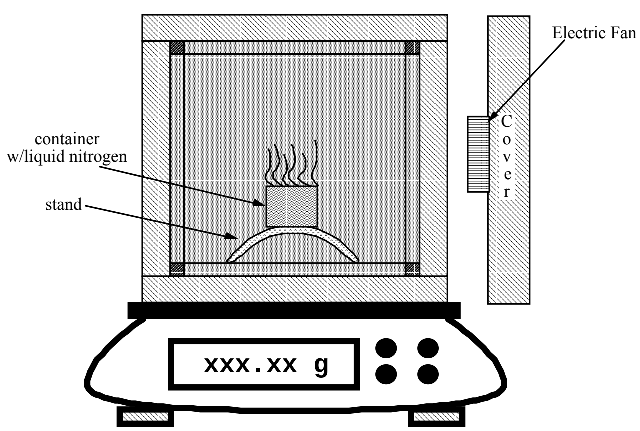

Figure 2: Set-up of mass transfer test for cavities.

The convective heat transfer is determined from the general energy balance equation (Equation 1) except an additional term is included to take into account mass transfer:

$$q_{mass} = h_{mass} h_{fg} A\left(C_{amb} - C_{plate}\right) \tag{5}$$

the mass transfer coefficient is related to the convective heat transfer through the heat/mass transfer analogy:

$$h_{mass} = h_{conv} \frac{D_{ab}^{2/3} \alpha^{1/3}}{k_{air}} \tag{6}$$

while the mass flux is determined from the following equation:

$$j_{mass} = h_{mass}\left(C_{amb} - C_{plate}\right) \tag{7}$$

RESULTS AND DISCUSSION

For the test with no mass transfer, results from typical test runs can be seen in Figure 3 and 4. For the four plate per side cavity, the plates are averaged. Plate (side) 2 typically has higher heat transfer coefficients since it is the first section to see the warmer ambient air entering the cavity. Plates (sides) 3, 4 and 5 have similar values most likely due to symmetry and orientation (vertical plates). Plate (side) 1 typically has lower values since air flowing over it has been warmed by other plates.

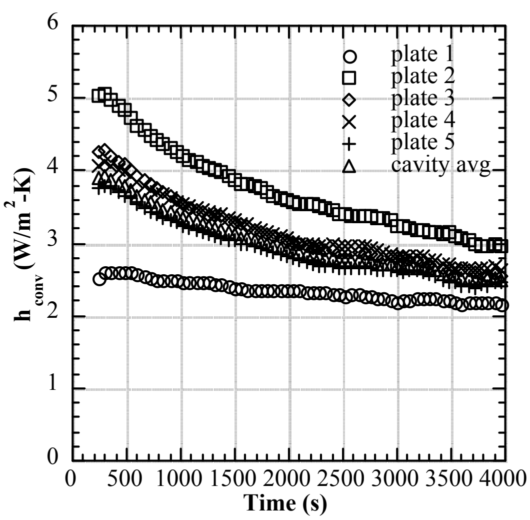

Figure 3: Convective heat transfer coefficients from a typical experimental run (w/no mass transfer) (1 plate/side cavity; 0° orientation; T_{init} ~ 80°C).

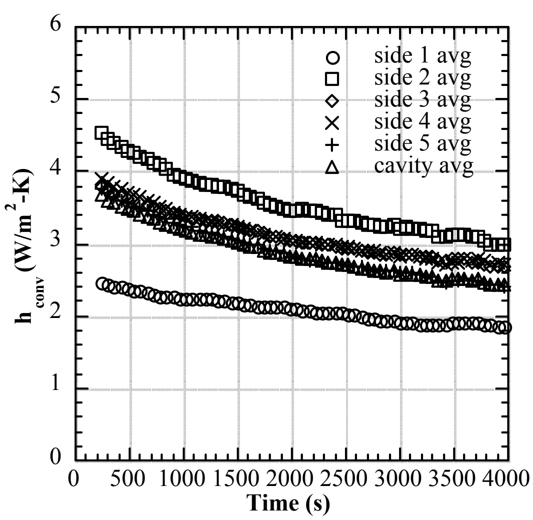

Figure 4: Convective heat transfer coefficients from a typical experimental run (w/no mass transfer) (4 plate/side cavity; 0° orientation; T_{init} ~ 80°C).

From the test runs, a Nussult number is obtained in the following form:

$$Nu_H = \frac{h_{conv} H}{k_{air}} \tag{8}$$

The Rayleigh number is determined from:

$$Ra_H = \frac{g\beta\left(T_{plate} - T_{amb}\right)H^3}{\nu_{air}\alpha_{air}} \tag{9}$$

where H is the characteristic height of the cavity. The air properties are determined from the film temperature:

$$T_f = \frac{T_{surf} + T_{amb}}{2} \tag{10}$$

and *volumetric thermal expansion coefficient:*

$$\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_{P,C} = \frac{1}{T_f} \text{ (Note that film temperature must be in Kelvin)} \tag{11}$$

Comparisons of experimentally determined cavity average Nusselt numbers between different orientations can be see in Figure 5 for the range of Rayleigh numbers tested. From the figure it appears that the 45° orientation produces slightly higher Nusselt values than the 0° and 90° orientation (which those two were similar). A least square fit to the experimental data is determined in the form:

$$\overline{Nu} = cRa_H^{1/3}$$

as well as the standard deviation and average deviation from the fit and the experimental data is shown in Table 1. The average difference between 0° and 45°/90° orientation is also presented. This shows that difference between 0° and 45° orientation can be contributed to more than error since the deviation of the curve fit is much less than the average difference of the orientations.

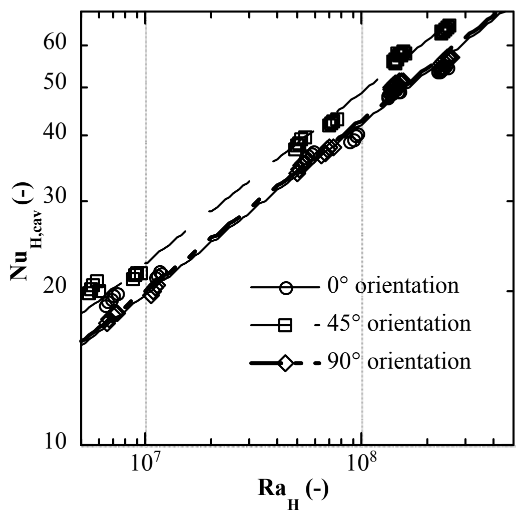

Figure 5: Cavity average Nusselt number for test data of various orientations.

Orientation of Cavity	Curve Fit of Data	Standard Deviation	Average Deviation	Average Difference from 0°
0°	$0.091391Ra_H^{1/3}$	±1.72	±4.67%	-
45°	$0.10528Ra_H^{1/3}$	±1.20	±2.86%	15.20%
90°	$0.093021Ra_H^{1/3}$	±1.23	±2.22%	1.78%

Table 1: Least squares fit to experimentally measured cavity average Nusselt numbers.

The cavity average Nusselt number for 0° orientation was compared to various other investigators and the results can be seen in Figure 6. Ranges in aspect ratio (H/L) for studies of 0.143, 1.0 and 1.5 are shown. For Skok, LeQuere, Penot, Krabel, and Cha, the cavity used in experiments or models had top, bottom and back wall heated while the side-walls were adiabatic. In Chan and Tien numerical model, the back wall was heated, while the other walls were adiabatic. The present study is quite different since all walls are heated.

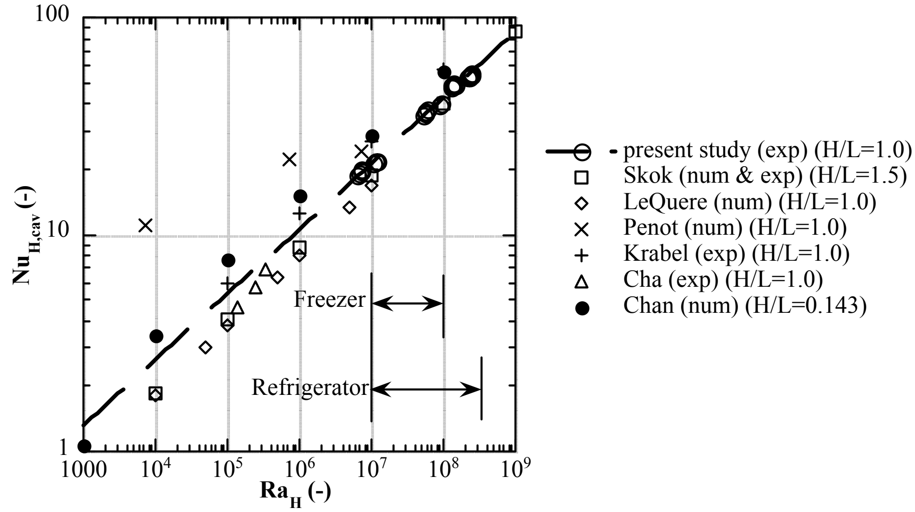

Figure 6: Cavity average Nusselt number for present study of 0° orientation and other studies.

For tests with mass transfer, the Rayleigh number is modified to take into account concentration effects.

$$Ra_H = \frac{g\beta\left(T_{amb} - T_{surf}\right)H^3}{\nu_{air}\alpha_{air}} + \frac{g\beta^*\left(C_{amb} - C_{surf}\right)H^3}{\nu_{air}\alpha_{air}} \tag{12}$$

where $$\beta^* = \frac{1}{\rho}\left(\frac{\partial \rho}{\partial C}\right)_{P,T} = \left[\frac{MW_{air}}{MW_C} - 1\right]\frac{1}{\rho} \tag{13}$$

In this study, as similar to Laleman's (1992) for Equation (12), the mass concentration difference buoyancy effect averaged 5-7% of the temperature effect. The heat transfer is shown in a similar fashion as in Figure 4. For mass transfer (cavity initially cooler than the ambient), plate 1 has the highest heat transfer coefficient due to it seeing the warmer ambient air while plate 2 has the lowest value (until around 1000 sec). Figure 8 shows the amount of water collected on the 1 plate-side cavity recorded by the scale.

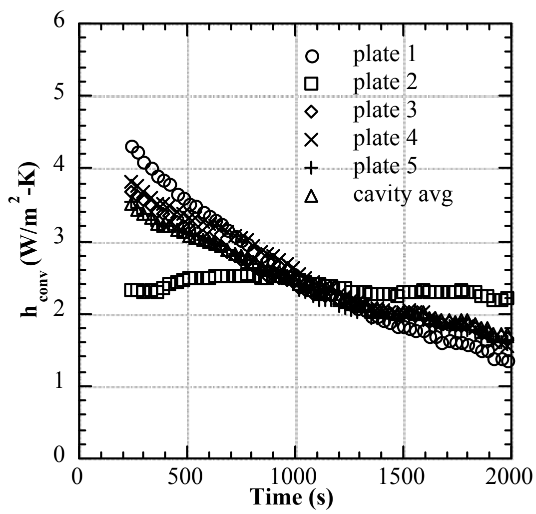

Figure 7: Convective heat transfer coefficients from A typical experiment run (w/mass transfer) (1 plate/side cavity, RH ~ 73%; T_{init} ~ -5°C).

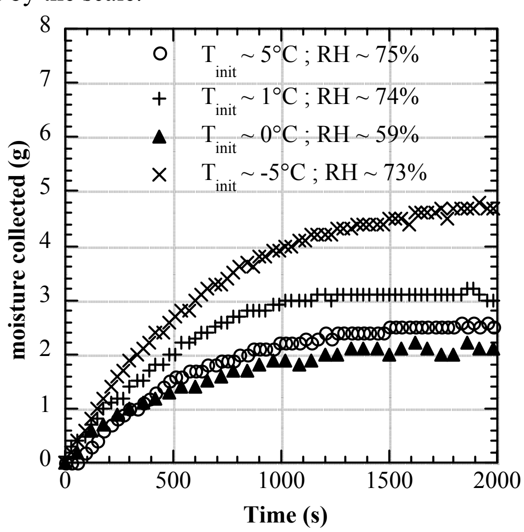

Figure 8: Moisture deposition on open cavity during initial tests.

From the data in Figure 8, the rate of mass accumulated on the cavity can be determined from a least square of points before the data at a given time. From this, the mass flux of water into the cavity can be determined:

$$j_{mass,\exp} = \frac{dm}{dt}_{\exp} \cdot h_{fg} = \dot{m}_{\exp} \cdot h_{fg} \tag{14}$$

The experimental mass flux of the data can be seen in Figures 9 and 10. These experimental mass flux values are compared to mass flux values obtained using the heat/mass transfer analogy using equations 6 and 7. From Figures 9 and 10, the data is in good agreement with the analogy.

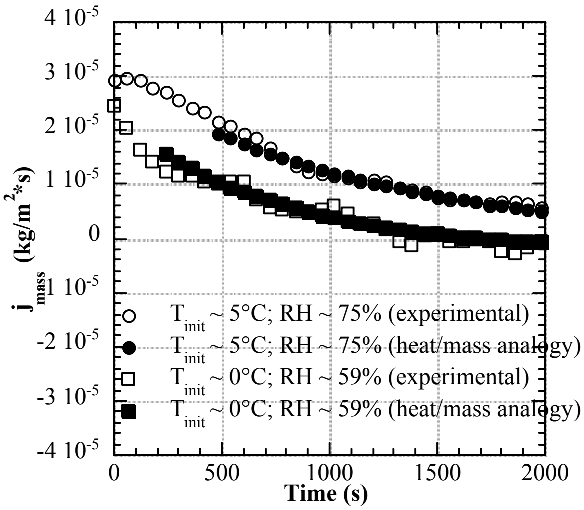

Figure 9: Cavity average mass flux for one plate/side cavity initial tests (test data 1 and 2).

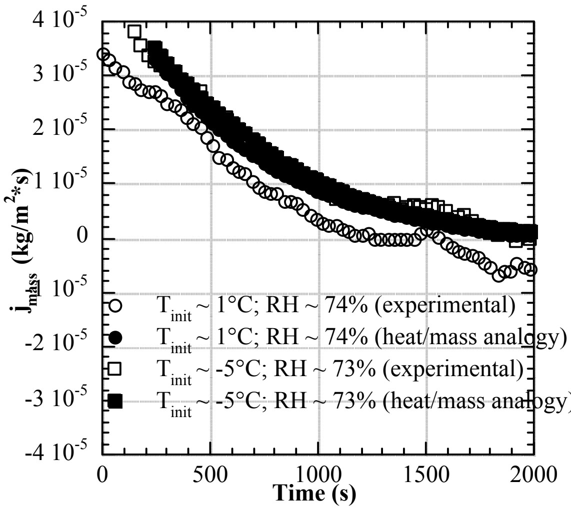

Figure 10: Cavity average mass flux for one plate/side cavity initial tests (test data 3 and 4).

CONCLUSIONS

Experiments were conducted to look at heat and mass transfer of cavities based on various sized cavities. Test on cavities with heat transfer only (no condensation) shows that cavity-average Nusselt numbers from a 45° orientation are slightly higher than at 0° and 90° orientations. Initial results from overall moisture loading in an open cavity so that at the conditions tested, the heat/mass analogy holds. Future work will involve further exploring cabinet heat and mass transfer of cavities. Also future experiments will determine local condensation on plates. Additional experiment on "objects" will provide information on heat and mass transfer to objects removed and them replaced into refrigerator/freezer cabinets.

ACKNOWLEDGEMENTS

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