2004

Heater Location Through Radiative Heat Optimization in a Finned Tube Evaporator

Ehab Mina
University of Illinois at Urbana Champaign

Cesar Gutierrez
University of Illinois at Urbana Champaign

HEATER LOCATION THROUGH RADIATIVE HEAT OPTIMIZATION IN A FINNED TUBE EVAPORATOR

Ehab MINA[1] and Cesar GUTIERREZ

Department of Mechanical and Industrial Engineering
University of Illinois at Urbana Champaign
140 MEB, MC-244, 1206 W. Green Street, Urbana, IL 61801, USA
Phone: (217) 265-5357, e-mail: ehabmina@uiuc.edu

ABSTRACT

Electric heaters are used to defrost the tubes of cooling coils. They represent about 15% of the annual energy usage of residential refrigerators. This paper investigates a new methodology to minimize the heat used during the defrost process for a given evaporator–heater configuration by locating the heater appropriately.

Two criteria are discussed and applied to determine the optimum heater(s) position, and their optimum share of the total power. The first criterion is maximizing the heat supplied to the tube that receives the minimum share of heat. The second criterion is minimizing the normalized standard deviation of the heat received by the tubes.

Two cases are analyzed: one heater and two heaters sharing the same total power as the original. For the two heaters case, the two criteria agree on the optimal heaters locations and optimal power share of 80% to the lower heater and 20% to the higher one.

1- INTRODUCTION

In the last 15 years domestic and commercial refrigerator manufacturers have faced different standards (DOE 1993 and DOE 2001) that have required design for lower energy consumption or environmental laws (phase out of CFCs and HCFCs) that have imposed energy penalties to existing products. While a refrigerator today consumes about one third of what it used to 15 years ago the pressure to get reductions is still present due to the increase in global competition and to some extent an increased awareness of consumers about energy use.

Energy consumption reductions have been attained, not by means of few big changes, but rather a number of small cumulative improvements. Changes to attain the required levels include high efficiency compressors, heat exchangers redesign, new insulation technologies, better door gaskets, perfection of air distribution systems, innovative control strategies, etc. (Radermacher and Kim, 1996).

Along with energy standards, nowadays consumers have more awareness of food preservation therefore temperature control is important. Some scholars, such as Flynn *et al* (1992) and Laguerre *et al* (2002), have been advocating counting the domestic refrigerator as part of the food supply chain, thus requiring temperature excursions during defrost operation and food loading to be kept within safe limits to avoid food spoiling.

It is because of these two reasons that manufacturers have tried to reduce the impact of defrost systems in frost free refrigerators. In order to remove frost formed in the evaporator, enough energy must be supplied to melt the ice and then dwell it outside the refrigerator. Defrost operation has a definite impact on refrigerators energy consumption due to: 1) energy supplied to defrost the heater (if used), 2) remnant heat in the freezer needing to be removed by refrigeration system; this remnant heat is also responsible for air temperature excursions that can be dangerous for food quality. The energy used to defrost the coil and later remove the remnant heat represents about 15 % of the annual energy usage of residential refrigerators.

Defforst efficiency and its effects on system performance are investigated in Ozyurt *et al* (2002). Different approaches have been reported to improve defrost efficiency or alternatively to reduce its impact. These efforts

---

[1] Corresponding author

followed three main approaches: retarding frost formation, designing new control strategies, and implementing new methods to defrost.

Watters *et al* (2001) described a geometry modification in which leading fins are separated to make evaporator more frost tolerant, Shin *et al* (2000) reported experimental on water hold up on surfaces having different hydrocharacteristics, the surface with better hydrophilicity resulted in less water holdup and higher defrost efficiency. Dyer *et al* (2000) studied the effect of coating a horizontal substrate on the frost growth rate. Based on their observations, they hypothesized that the water coverage of the substrate affects the thermal resistance of the mature frost layer and in turn affects the growth rate of frost. Jhee (2002) investigated the effect of surface treatment on the frosting and defrosting behavior in a fin-tube heat exchanger, hydrophilic surfaces mainly influence the frosting behavior while hydrophobic surfaces influence the defrosting behavior. From experimental results, evaporators with hydrophobic surface treatment improve defrost efficiency. Knoop *et al* (1988) described a defrost system that adapts to usage patterns and ambient conditions, Radcenco *et al* (1995) described optimization of defrost by intermittent operation. Norton (2000) described hot gas defrost as a simple method to defrost evaporators.

This paper investigates minimizing the heat used during the defrost process for a given evaporator–heater configuration. By locating the heater appropriately, the heat will be more evenly distributed among the tubes. The evaporator coil represented in figure 1 is used as an example of applying the technique described in the paper. The coil consists of two banks of eight tubes each, arranged in a triangular shape. Each tube is covered by spiral fins. A pan covered with reflective surface is placed below the coil to collect melted ice. The option of splitting the electrical power supplied to the heater into more than one heater is also investigated. Symmetry of the arrangement is exploited and therefore the analysis is carried over half the tubes and half the heater.

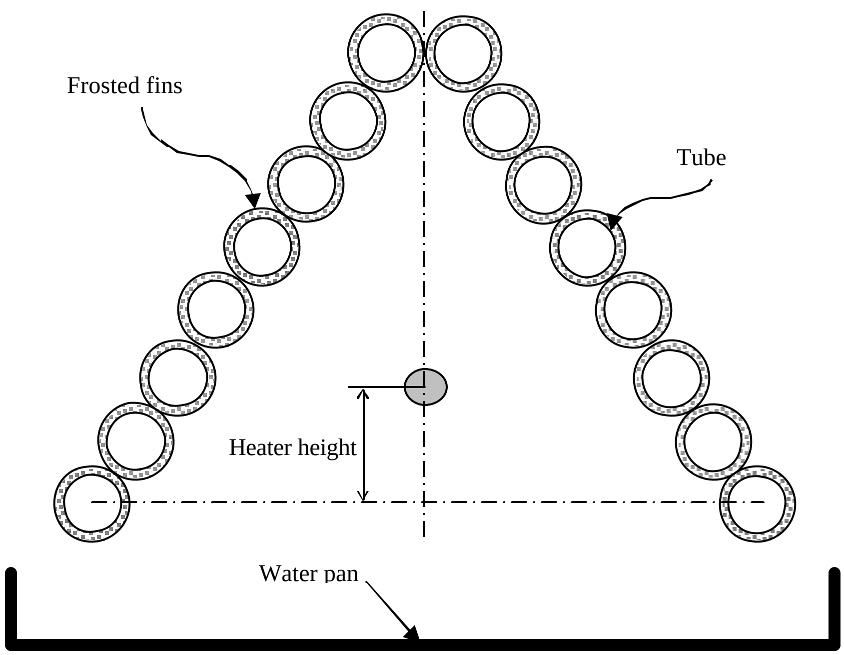

Figure 1 Typical configuration for the evaporator heater under study.

2- FACE FACTORS CALCULATIONS

The problem studied involves radiation heat transfer between three surfaces, heater, pan and tubes. And hence the face factors (view factors) between each two of the surfaces need to be computed. The length of the tubes, the heater and the pan is of the same order of magnitude of the refrigerator width (about 0.6 m); this length is much larger than the diameter of the tube and heater or the width of the pan; therefore the end effects are negligible. The radiation between any two surfaces can then be treated as two-dimensional.

2-1 Face Factor from the Heater to the Tubes

The heater area is much smaller than the surface of the tube and therefore the heater is approximated by a point source. The tubes are considered to be at the same temperature. However, heat radiated from the heater to each tube is important in this analysis and therefore the view factor between the heater and each tube row is calculated as a function of the heater position. The face factor is the ratio of the angle θ_i to half the total view angle of the heater (π rad.). The angle θ_i is the angle enclosed between the two tangents to the surface of the tube. Figure 2a illustrates the face factor from the heater to one tube.

2-2 Face Factor from the Heater to the Pan

In case of the pan, the heater area is also considered a point source and the face factor is the ratio of the angle between the lines connecting the center of the heater and the sides of the pan, θ_{pan}, to total angle of the heater (2π rad.). Figure 2b illustrates the face factor from the heater to the pan.

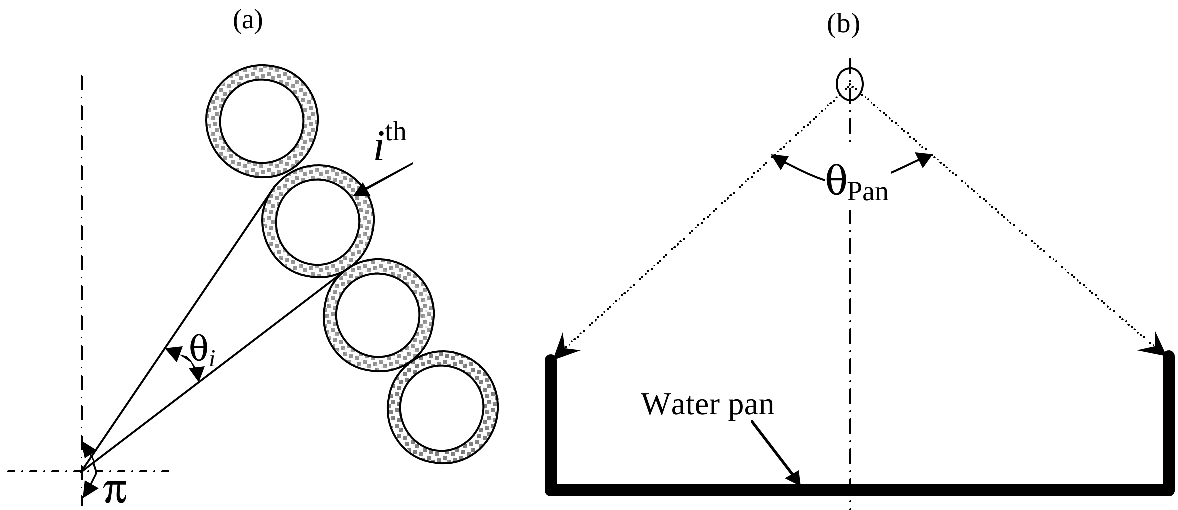

Figure 2 (a) the view factor from the heater to each tube (b) the view factor between the heater and the pan

2-3 Face Factor from Pan to Tubes

Since the radiation between the pan and the tubes is two-dimensional, the string method can be used to calculate the face factor from the pan to each tube. According to the string rule (Brewster, 1992), the face factor is given by equation (1).

$$A_i F_{ij} = \frac{1}{2}\left[\sum L_c - \sum L_u\right] \tag{1}$$

In equation (1) ΣL_c is the length of the two strings that cross each other when connecting the sides of the two surfaces and ΣL_u is the length of the two strings that does not cross each other when connecting the sides of the two surfaces. Figure 3 shows the crossed (AD and BC), and non-crossed (AC and BD) strings between the sides of the pan and one of the tubes.

The position of the heater is assumed not to affect the face factor calculations because of the small dimensions of the heater with respect to other surfaces. Results discussed later in the paper shows the small contribution of the pan to the heat going to the tubes. And hence justifies any approximation in the face factor calculations.

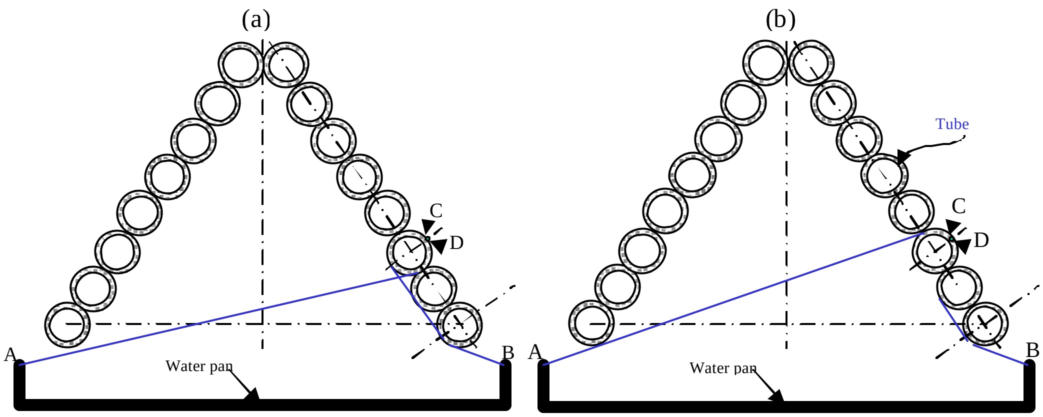

Figure 3 String rule used to calculate the face factor between the plate and one tube. (a) Crossed strings, and (b) non-crossed strings.

3- HEAT BALANCE

Radiation heat is transferred between the heater, the tubes and the pan. The medium is assumed to be non-participating. The difference in temperature between the ice accumulated on each tube is neglected and consequently the heat conducted from one tube row to another is also neglected. The convection heat transfer from the heater accounts for less than 2 %* of the total heat dissipated by the heater and hence neglected in the current analysis. Ice, water, and the heater surfaces have very high emissivities and hence they are treated as black bodies. Considering the above assumptions and simplifications the radiation interaction between the three surfaces can be schematically represented by figure 4. E_{Heater}, E_{Tubes} , E_{Pan} are the emissive power of the heater, tubes, and pan, respectively. The emissive power is calculated using equation (2), where σ is the Stefan-Boltzmann constant, and T_i is the temperature of surface *i* in Kelvin. The resistance to the radiative heat transfer between two surfaces *i* and *j* is calculated using equation (3), where A_i is the area of the surface *i* and F_{ij} is the face factor from surface *i* to surface *j*.

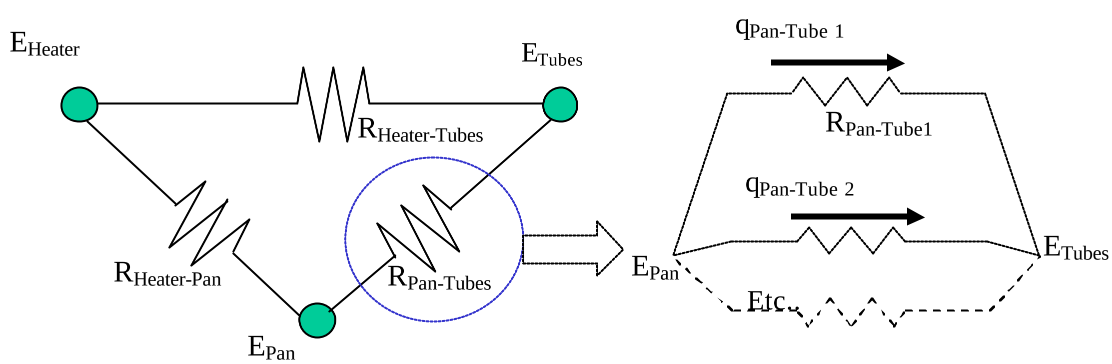

Figure 4 Analog circuit to the heat transfer problem. The detail shows the individual resistances between the pan and each tube. The same applies for the heater to tubes resistances.

$$E_i = sT_i^{\,4} \tag{2}$$

$$R_{ij} = \frac{1}{A_i F_{ij}} \tag{3}$$

* 1.8 % in case of 1000°C heater temperature and –20 °C air and tubes temperatures.

The temperature of the heater is computed from the known power. The temperature of the tube is taken as the average temperature of the ice accumulated on the tube surface during the defrost period. The pan temperature is unknown and is computed using the energy balance represented in equation (4).

$$q_{H-P} = q_{P-T} + q_{Cond} + q_{Water} + q_{Conv} \tag{4}$$

The left hand side of equation (4) represents the heat radiated from the heater to the pan. The right hand side of equation (4) represents the heat dissipated from the pan; radiated to the tubes ($q_{P\text{-}T}$), conducted to the foam (q_{Cpnd}), carried with the water (q_{Water}) or convected by the freezer air (q_{Conv}). In the later natural convection has been assumed and a coefficient of 5 W/m^2-K is used.

Figure 5 shows the heat received by each tube for a certain heater position (1 cm above the lowest tube centerline). Each tube receives heat from the heater as well as the water pan. Figure 5 shows the contributions of both sources to heat received by each tube. The contribution of the pan to the heat gained by the tubes is very small compared to the heater contribution. The heat radiated by the pan is concentrated on the lowest tube row, while the tube closer to the heater receives more of its heat. The total power of the heater is 680 W/m (400 Watts heater of about 58 cm long).

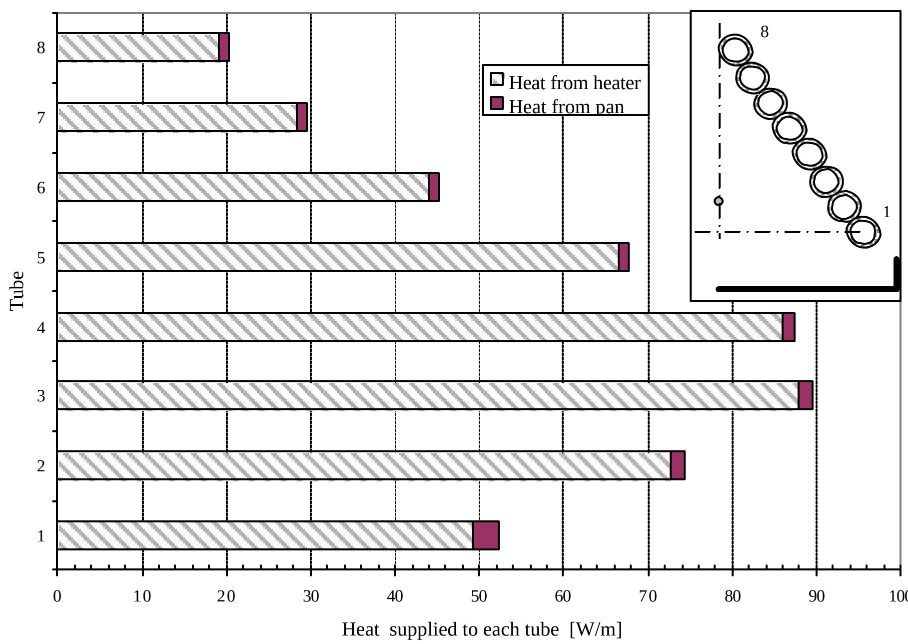

Figure 5 Tubes shares of heat for a heater position of 1 cm above the lowest tube centerline.

4- CRITERIA FOR DETERMINING OPTIMUM POSITION

In order to run the defrost process optimally the heat supplied should be as close to the heat needed as possible. This can be cast out into several criteria. Two of those criteria are discussed in this paper.

The first criterion is to maximize the heat received by the "critical tube", which is defined as the tube that receives the minimum share of heat. This tube is the last tube to defrost and hence its share of heat is key to determine the defrost time. Maximizing the heat supplied to the critical tube reduces the defrost time, and consequently saves energy.

The second criterion is to minimize the ratio between the standard deviation and the average of the heat received by the tubes. Increasing the average heat supplied to the tubes means reducing the heat lost (radiated to the freezer walls and air or to heat the drainage water). Reducing the standard deviation helps to distribute the heat more evenly among the tubes and hence reduce the amount of excess heat that will be removed by the system.

Figure 6 represents the two criteria applied to study the optimal heater position in an eight tube coil similar to the sketch shown in the Figure 1. The first criterion shows that a heater position of 2 cm above the centerline of the first row provides the maximum amount of heat to the critical tube (28.2 W/m). On the other hand, the second criterion favors a heater position of 1.2 cm above the same reference to be the optimum position. Such position will provide the most even distribution of heat among the tubes.

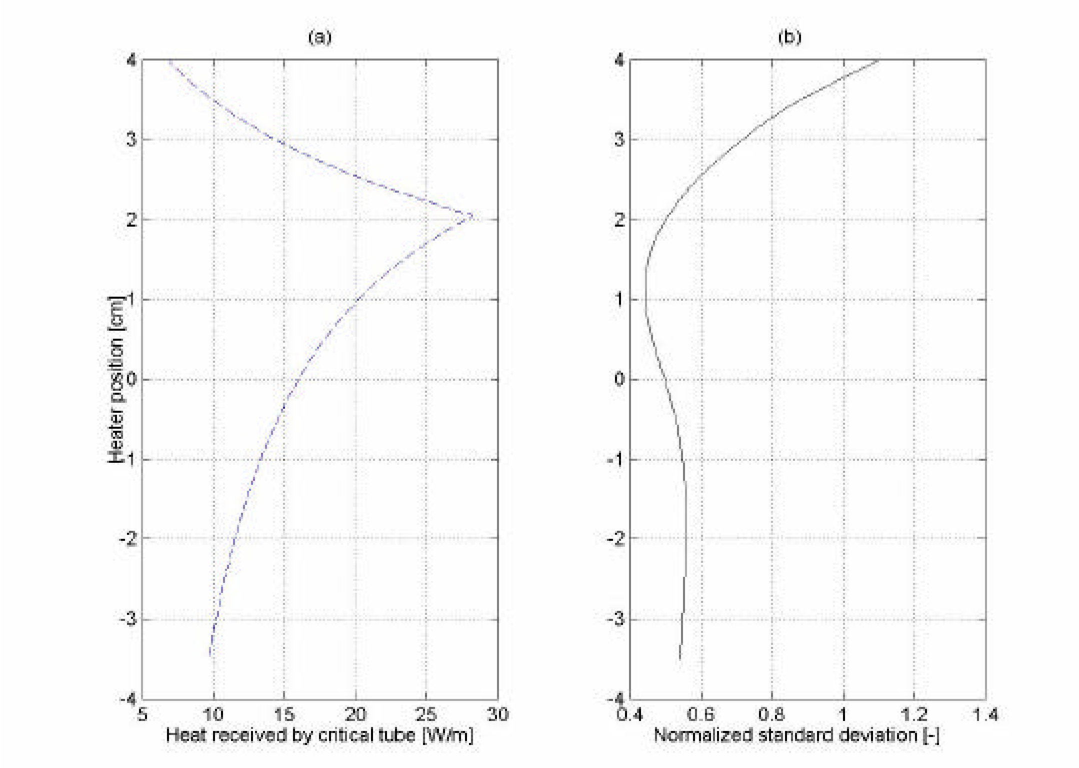

Figure 6 Two different criteria to select the optimal heater position: Left, maximizing the heat received by the "critical tube". Right, minimizing the ratio of the standard deviation to the average of the heat supplied to the tubes.

5- SPLITTING THE POWER OVER TWO HEATERS

The logic behind the two criteria used leads to believe that splitting the power into more than one heater will distribute the heat more evenly and increase the heat supplied to the critical tube. Several arrangements of two or more heaters could be studied. In this paper the investigation is limited to two heaters placed on the center line at different positions. The two heaters have the same total power as the previously discussed case of one heater. However the total power is not shared equally between the heaters. A power fraction is defined as the ratio between the power carried by one of the heaters and the total power.

Figure 7 represents the maximum heat supplied to the critical tube (first criterion) and the normalized standard deviation (second criterion) at different values of power fraction. The two criteria agree that splitting the total power into two heaters such that one of the heaters carries 80 % of the heat and the other carries 20 % is the optimal split of power. A close look to this ratio is seen on figure 8, where contours of the maximum heat supplied to the critical tube and the normalized standard deviation are plotted on part (a) and (b) respectively. The optimal arrangement is to have the heater with 80 % of the power at about 1 cm. and the second heater at about 4 cm above the centerline of the first row.

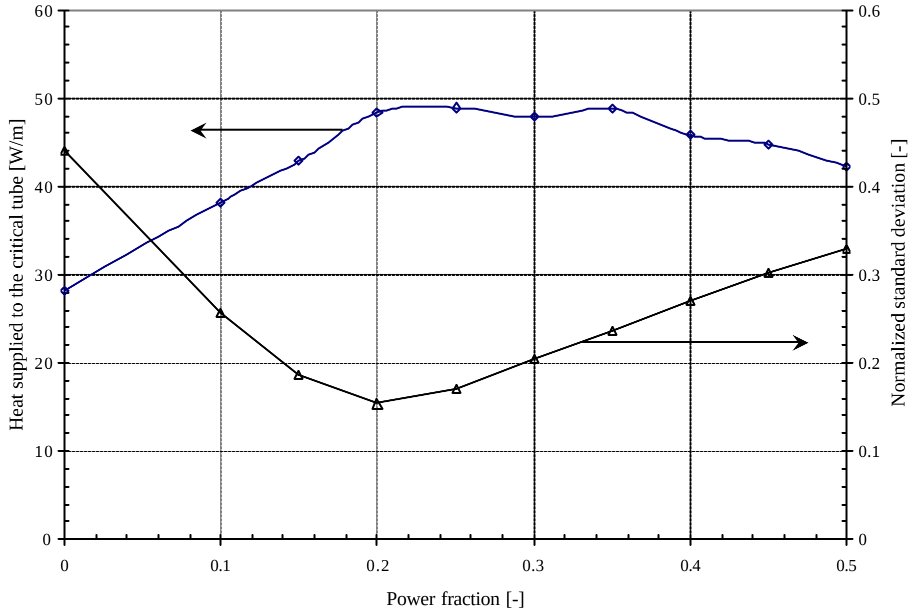

Figure7 Maximum heat supplied to the critical tube (first criterion) and normalized standard deviation (second criterion) at different values of Power fraction.

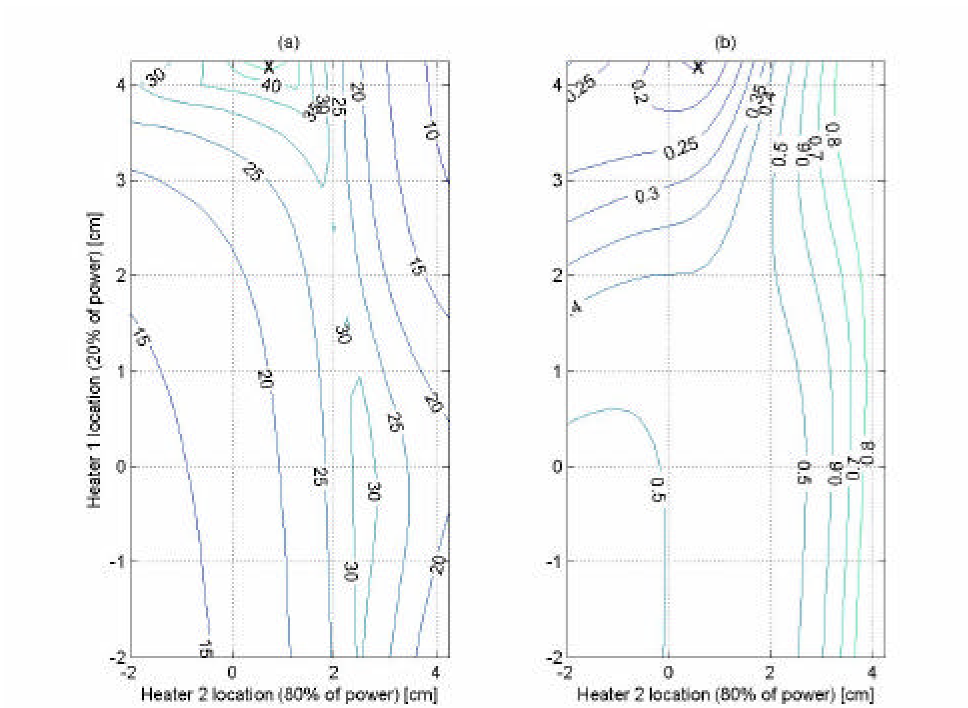

Figure 8 Contours of maximum heat supplied to the critical tube (first criterion) and normalized standard deviation (second criterion) at different heaters arrangements for a power fraction of (80/20)

6- CONCLUSIONS AND FUTURE WORK

This paper investigated the optimization of defrost process by proper placement of the defrost heater. Two simple criteria were developed to evaluate the heater position. The first criterion maximizes the heat supplied to the critical tube, while the second minimizes the normalized standard deviation of the heat supplied to the tubes.

The optimal position of a single heater was determined based on both criteria. The option of splitting the power over two heaters placed on the symmetry centerline was investigated. The two criteria agree that the optimal power split is 80 % for the lower heater and 20 % for the upper heater. The two criteria give similar heater positions for the optimal share. Both criteria require the heater with 80 % of the power to be placed at about 1 cm above the lower tube centerline and the heater with 20 % of the power at about 4 cm above the same datum

For future work a more elaborate objective function will be considered. The objective function will account for the power loss due to heat loss (conducted through the insulation and lost with the drained water) as well as the remnant heat which will be counted as a loss of the heater power as well as an extra load on the cycle. The effect of the heater position on reducing the defrost frequency will also be considered.

Another future task is to experimentally study the distribution of frost over the cooling coil and place the heater(s) so that the tube with denser frost receives more radiative heat.

Future work will analyze other coils arrangements and other heaters arrangements as well. For larger evaporators, three heaters placed on positions other than the centerline may be appropriate.

REFRENCES

Brewster M.Q. 1992. "Thermal Radiative Transfer and Properties" J. Wiley & Sons. New York.

Dyer, J.M., Storey, B.D., Hoke, J.L., Jacobi, A.M., and Georgiadis, J.,G. 2000. "Experimental Investigation of the Effect of Hydrophobicity on the Rate of Frost Growth in Laminar Channel Flows" ASHRAE Transactions, Vol 106, p. 143-151.

Flynn, O.M., Blair, I., and McDowell, D. 1992. "The Efficiency and Consumer Operation of Domestic Refrigerators" *Int. J. Refrig*. Vol.15: p. 307 – 312.

Jhee, S., Lee, K., Kim, W., 2002, "Effect of Surface Treatment on the Frosting/Defrosting Behavior of a Fin-Tube Heat Exchanger", *Int. J. Refrig*. Vol. 25, no. 8: p. 1047 – 1053.

Knoop, D.E., Tershak A., Thieneman M., 1988, "Adaptive Demand Defrost and Two-Zone Control and Monitor System for Refrigeration Products", *IEEE Transactions on Industry Applications* vol. 20, no. 2: p. 337 – 342.

Laguerrre, O., Derens, E., and Palagos, B. 2002. "Study of Domestic Refrigerator Temperature and Analysis of Factors Affecting Temperature: a French survey" *Int. J. Refrig*. Vol. 25: p. 653 – 659.

Norton, E., 2000, A Loot at Hot Gas Defrost, *ASHRAE Journal*, vol. 42, no. 10: p. 88.

Ozyurt, B., Karatas, H., Omam. C., Egrican, N., Hocaoglu, S. Cousins, S., 2002, Defrost Efficiency in a No-Frost-Type Household Refrigerator, *ASHRAE Transactions* vol. 108, no. 1: p. 460 – 466.

Radcenco, V., Vargas, J., Bejan, A., Lim, J., 1995, "Two Design Aspects of Defrosting Refrigerators", *Int. J. Refrig*. Vol. 18, no. 2: p. 76 – 86.

Radermacher, R., Kim, K., 1996, "Domestic Refrigerators: Recent Developments", *Int. J. Refrig*. Vol. 19, no. 1: p. 61 – 69.

Shin, J., Kim, C., Ha, S., Kim, J. 2000, A Study of Water Holdup on Two Surfaces with Different Hydrocharacteristics, *Journal of Flow Visualization and Image Processing*, vol. 7, no. 4: p. 343 – 351.

Watters, R., O'Neal, D., Yang, J., 2001, Effect of Fin Staging on Frost /Defrost Performance of a Two-Row Heat Pump Evaporator at Standard Test Conditions, *ASHRAE Transactions* vol. 107, no. 2: p. 240 – 249.