1996

An Experimental Study of Air-Side Heat Transfer and Friction Factor Correlations on Domestic Refrigerator Finned-Tube Evaporator Coils

H. Karatas
Arcelik A.S.

E. Dirik
Arcelik A.S.

T. Derbentli
Istanbul Technical University

AN EXPERIMENTAL STUDY OF AIR-SIDE HEAT TRANSFER AND FRICTION FACTOR CORRELATIONS ON DOMESTIC REFRIGERATOR FINNED-TUBE EVAPORATOR COILS

Hakan Karatas*, Engin Dirik*, Taner Derbentli**

* R&D Center, Arcelik A.Ş., Tuzla 81719, İstanbul, Turkey
** Assoc. Prof., Faculty of Mechanical Engineering, İstanbul Technical University, Gümüşsuyu 80191,İstanbul, Turkey

ABSTRACT

In this study, air-side heat transfer and pressure drop characteristics of finned-tube refrigerator-freezer evaporators operating under dry surface conditions were investigated. Four different evaporator coils were tested for both uniform and nonuniform velocity and temperature inlet conditions of the air flow. The heat transfer coefficient and the friction factor were correlated in terms of Reynolds number and finning factor - the ratio of outside total surface area to outside tube surface area of a coil - by using test data obtained under uniform inlet conditions. The Reynolds number varied from 300 to 1000 and the finning factor of the tested coils ranged from 1 to 6. Experimental results showed that the established correlation can also be used for the calculation of the heat transfer coefficient for the nonuniform flow situations by using mass flow averaged values of temperature and velocity at the evaporator inlet.

NOMENCLATURE

A : Surface area, m^2

d : Tube diameter, m

f : Friction factor, dimensionless

G_{max}: Mass flux at minimum flow area, $kg / m^2 s$

h : Heat transfer coefficient, $W / m^2 °C$

j : Colburn factor, dimensionless

k : Thermal conductivity, $W /m°C$

Nu : Nusselt number, $h_o d_o / k_a$

Pr : Prandtl number

Q : Heat transfer rate, W

Re: Reynolds number, $G_{max} d_o / \mu_a$

t : Thickness, m

T : Temperature, °C

Greek Letters

ε : Finning factor, dimensionless

ρ : Density, kg / m^3

μ : Dynamic viscosity

σ : The ratio of minimum flow area to face area of the coil.

θ : Dimensionless temperature parameter

Subscripts

a : Air

f : Fin

i : Inlet, inside

m : Mean

o : Outlet, outside

t : Tube

w : Water

INTRODUCTION

The evaporator coil considered in the present study is used in a two-door top-mount refrigerator-freezer unit. The fresh-food and freezer compartments are maintained at different temperatures. The air flow rates circulating in the individual compartments are also different. Such an operation results in a distorted flow situation at the evaporator inlet. Furthermore, the axial fan mounted downstream of the coil causes nonuniform flow distribution through the coil. Consequently, the actual operating conditions of evaporators in the refrigerators are considerably different than the usual wind tunnel testing.

Most of the research presented in the literature are given for the heat exchangers used in commercial refrigeration and air conditioning applications. A comprehensive review of the literature has been given by Webb [1]. In these studies, the flow velocities and geometrical parameters such as fin type, row number and tube arrangement are considerably different than those of a refrigerator evaporator. A recent parametric study conducted by Rite [2] considers the effects of frost accumulation on the overall conductance and the pressure drop characteristics of domestic refrigerator evaporator. In his study, performance parameters are not expressed in terms of correlations which can be utilized for design purposes. The aim of this study is to develop air-side heat transfer and friction factor correlations to be used in design. To investigate the influence of the nonuniform

temperature and velocity field at the evaporator inlet, an experimental setup is designed and constructed. In this setup, the heat transfer mechanism in the evaporator is thermally reversed by circulating the hot water in the tubes. Neglecting the natural convection effects is justified based on the mixed convection studies given in the literature.

EXPERIMENTAL SETUP AND PROCEDURE

Apparatus

A schematic diagram of the apparatus is shown in Figure 1. Laboratory air is drawn by a fan located upstream of the evaporator through the air ducts of the mullion of the selected refrigerator unit. Mullion is made of styrophor. After leaving the evaporator, air flows in a circular tunnel constructed according to IEC 866 standard [3]. An orifice plate is positioned downstream of the flow straightener to measure the air flow rate. Since the left and right air ducts in the mullion have equal cross sectional areas, the air split ratio is controlled by measuring and changing the air flow rate in the middle duct alone. Another tunnel is constructed upstream of the middle duct to measure the air flow rate which is regulated by a booster fan located at the inlet of the tunnel.

Air heaters are used in the ducts with equal cross sectional areas for the nonuniform temperature inlet condition at the evaporator inlet. A flow mixing device is used to achieve a uniform temperature distribution in the duct. Air flow straighteners are placed in each of the three ducts. To minimize the heat losses to the surroundings, the test section is insulated with a 50 mm thick layer of glass wool. Both of the fans are driven by direct current motors. The hot water circulated in the tubes of the coil is supplied from a hot water tank whose temperature is kept constant within ∓ 2 °C of the desired temperature by a thermostat activated heater. A pump is used to circulate the water in the test coil and the mass flow rate is varied with a valve.

Test Coils

The evaporator coils tested are products of the same manufacturer. The geometrical characteristics of the coils are given in Table 1. The overall dimensions of the coils are 50x248x535mm. The tubes have 8.0 mm outside diameters and 0.64mm wall thicknesses. The fins are flat aluminium with collars and the tube material is also aluminium. The bond between the fins and the tubes is mechanical and there is no metallurgical bonding. The fin thickness is 0.19 mm. The width of the collars is 1.2 mm. The flow of air and water in the coils is counter-cross/parallel-cross flow.

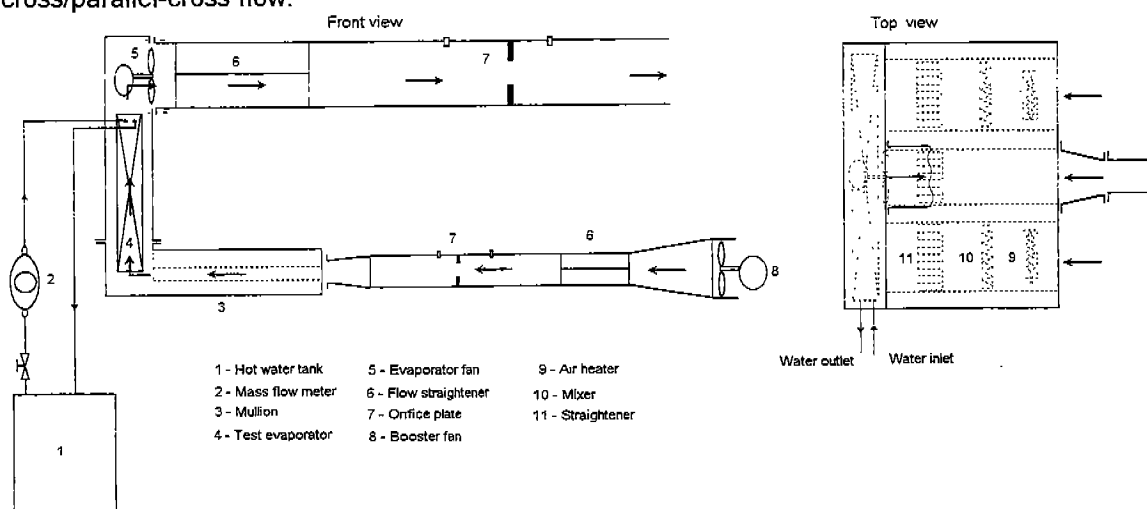| 1 - Hot water tank | 5 - Evaporator fan | 9 - Air heater |
2 - Mass flow meter	6 - Flow straightener	10 - Mixer
3 - Mullion	7 - Orifice plate	11 - Straightener
4 - Test evaporator	8 - Booster fan	

Figure 1. Experimental Setup

Instrumentation and Experimental Procedure

The parameters measured during the tests were the temperatures and the mass flow rates of both the inlet and outlet air and water streams. For temperature measurements type T thermocouples calibrated in a standard calibration bath with a digital thermometer which has an accuracy of 0.035% of the reading value are used. Air temperature in each duct is measured at the outlet section of the duct. The positions of thermocouples are centered and equally spaced across the ducts. Air temperature at the exit of the evaporator fan is measured at five points one being at the center, the others located circumferentially at mid radius. The air temperatures in front of orifices are measured to calculate the density of the air.

The water mass flow rate is measured by using a coriolis type mass flow meter with a 0.2% accuracy of measured value. Pressure drops through the orifices of the tunnels are measured with a factory calibrated micromanometer which has an accuracy of 0.25% of the pressure read. Orifices are calibrated by measuring the air velocities in the tunnels with a laser doppler anemometer system to find the orifice coefficients as a function of the Reynolds number. The pressure drop of air between the coil inlet and outlet is measured at three stations with the same micromanometer.

Data were recorded in 5 minute intervals by data loggers for a period of half an hour after steady state conditions were reached. The change in the inlet temperature of the water $\mp 0.3°C$ and of the air $\mp 0.5°C$ at most during each test. 24 experiments were conducted for uniform inlet velocity and temperature conditions. The air split ratio of the middle duct was maintained as 30% of the total air flow rate to obtain uniform velocity in each duct. A total of 44 tests were run for the nonuniform inlet conditions tests.

Table 1. Geometrical characteristics of the test coils.

	1	2	3	4
Longitudinal tube pitch (mm)	19.05	19.05	19.05	19.05
Number of rows	13	13	13	13
Transverse tube pitch (mm)	22.0	22.0	22.0	22.0
Tubes per row	2	2	2	2
Fin spacing (mm)	10	5/10/5*	5	**
Face dimensions (mm,mm)	50x535	50x535	50x535	44x535
Minimum flow area (m²)	0.0162	0.0159	0.0158	0.0149
Tube inside surface area (m²)	0.2899	0.2899	0.2899	0.2899
Tube outside surface area (m²)	0.3472	0.3441	0.3419	0.3526
Fin surface area (m²)	0.8431	1.3385	1.6817	**
Finning factor	3.4281	4.8902	5.9192	1

* varying accross the coil
** bare tube

ANALYSIS OF EXPERIMENTAL DATA

The first step in the calculation process for the average heat transfer coefficient is to find the overall conductance (UA) using the following equation,

$$Q = U\,A\,F\,\Delta T_{lm} \tag{1}$$

where ΔT_{lm} is the logarithmic mean temperature difference for a pure counter flow heat exchanger defined as:

$$\Delta T_{lm} = \frac{(T_{w,i} - T_{a,o}) - (T_{w,o} - T_{a,i})}{\ln \dfrac{(T_{w,i} - T_{a,o})}{(T_{w,o} - T_{a,i})}} \tag{2}$$

F, is the correction factor for other than a pure counter flow heat exchanger. Since the water temperature drop is controlled within 2-3 °C during the tests,and the number of tube passes in streamwise direction is 13, a pure counter flow can be assumed and so F can be taken as 1. The total heat exchange is calculated by averaging the water and the air side heat transfer rates. The difference between these values stayed within $\mp 4\%$ of the averaged heat transfer rate for most of the test runs.

The overall conductance based on the total outside surface area of the coil is defined as:

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_{t,i}} + R_c + \frac{1}{h_o(A_{t,o} + \eta_f A_f)} \tag{3}$$

Equation (3) is obtained by assuming the tube wall resistance is negligible. Here, R_c is the contact resistance between the tube outer surface and the fin base. Since the test coils are products of the same manufacturer a constant value is used in the data reduction procedure.

The water side heat transfer coefficient, h_i was determined by the Dittus-Boelter correlation given for turbulent flow in tubes [4]. The fin efficiency, η_f was obtained by a finite element analysis of the evaporator slit fin using a commercial software. Computational grid on a part of the fin sheet is shown in Figure 2. A governing nondimensional parameter for the fin geometry considered in this study is obtained from the general heat transfer equations and is defined as:

$$m_f = \sqrt{\frac{2\, h_o\, d_o^2}{k_f\, t_f}} \tag{4}$$

The fin efficiency was calculated at 8 points in the range of $0.13 \le m_f \le 0.52$. Least square method was used to formulate the fin efficiency in terms of fin parametter as a fourth order polynomial function. [5].
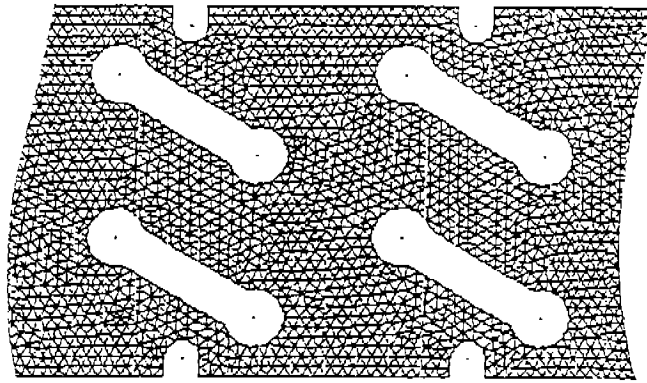

Figure 2. Computational grid generated on the fin sheet.

Heat transfer correlation for finned tube heat exchangers is generally given in the following form.[6]

$$Nu = a\, Re^b\, Pr^{1/3}\, \varepsilon^c \tag{5}$$

This equation can be rearranged in terms of j-Colburn factor,

$$j = \frac{Nu}{RePr^{1/3}} = a\, Re^{b-1}\varepsilon^c \tag{6}$$

where the Reynolds number is based on the tube outside diameter. For the determination of Re, mass flux calculated at the minimum flow area (A_{min}) is used. The geometrical parameters of the evaporator are represented by the finning factor defined below:

$$\varepsilon = \frac{A_o}{A_{to}} \tag{7}$$

A multiple regression analysis was made to calculate the constants a,b and c in equation (6) by using the 24 experimental data points of the uniform inlet tests. The air-side heat transfer correlation which is determined in the ranges of $300 \le Re \le 1000$ and $1 \le \varepsilon \le 6$ is given below:

$$j = 0.138\, Re^{-0.281}\, \varepsilon^{-0.407} \tag{8}$$

468

The j factors of data points reduced from the measured values and the values predicted by the correlation are shown in Figure 3. All of the data points are within the $\mp$ 10% of the values predicted by the correlation.

The air-side friction factor, f , was calculated from the pressure drop equation [5],

$$\Delta p = \frac{G_{max}^2}{2\,\rho_i}\left[\left(1+\sigma^2\right)\left(\frac{\rho_i}{\rho_o}-1\right)+f\,\frac{A_o}{A_{min}}\frac{\rho_i}{\rho_m}\right]$$

(9)

where A_o is the total outside surface area and A_{min} is the minimum flow area of the coil. Reduced f values of the first three coils were formulated as functions of Re and ε in a form similar to equation (6). The coefficents were found by using a multiple regression analysis. The following correlation was obtained,

$$f = 0.152\,Re^{-0.164}\,\varepsilon^{-0.331}$$

(10)

in which $300 \le Re \le 1000$ and $3.5 \le \varepsilon \le 6$. Figure 4 shows the f factors of measured values and values predicted by the correlation. 90% of the experimental data points are within the $\mp$ 10 % of the correlation.
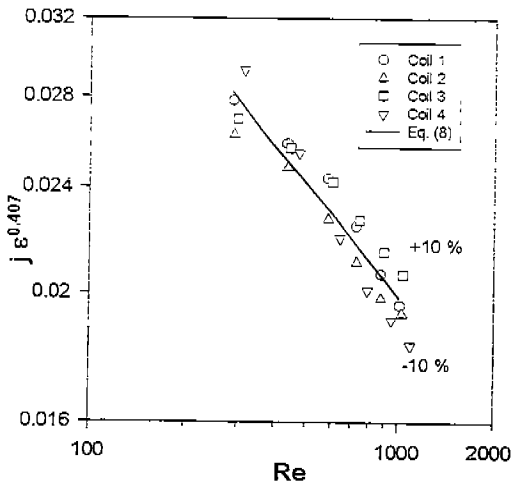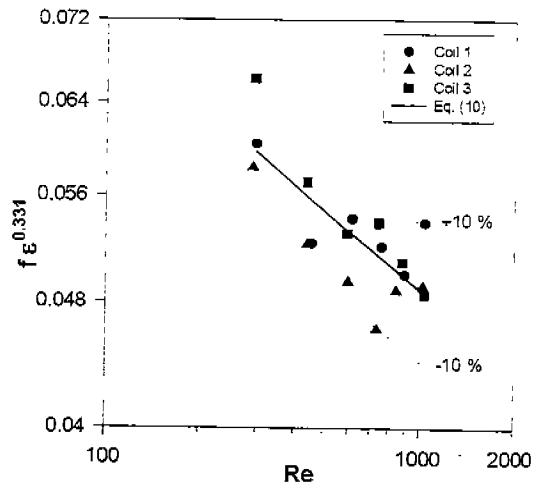

Figure 3. Colburn factor correlation.

Figure 4. Friction factor correlation.

Reduction of the Non uniform test data

The temperature distortion at the evaporator inlet is defined in terms of a nondimensional parameter,

$$\theta = \frac{T_{w,m} - T_{side}}{T_{w,m} - T_{mid}}$$

(11)

where T_{side} is the mean temperature of air flowing in the side ducts, T_{mid} is the mean temperature in the middle duct and $T_{w,m}$ is the average temperature of water flowing inside the coil tubes. According to this definition, the case having the values of $\theta = 1.0$ and the 30% air split ratio corresponds to the uniform inlet flow situation. The split ratio and θ value are perturbed from this base case to generate a matrix of nonuniform tests. This creates 11 nonuniform test runs for each coil configuration where the air flow rate in the middle duct takes the values of 30, 20 and 10% of the total air flow rate while the values of 0.25, 0.50, 0.75 and 1.0 are taken for θ.

Establishing a correlation in terms of nondimensional temperature parameter and air split ratio is not practical from the design point of view. A simple scheme was considered to characterize the effects of inlet distortion on the heat transfer parameters. Here, the objective was to use the uniform flow correlation for the

nonuniform flow situations by using a simple correction factor. For this reason, the measured heat transfer rates of nonuniform tests were compared with the predictions obtained from equation (11) by using the mass averaged temperature and velocity values for the inlet condition. Comparison of the measured and calculated heat transfer rates is shown in Figure 5 for each coil tested. This comparison clearly shows that equation (11) with averaged inlet values can be used for the nonuniform test cases.
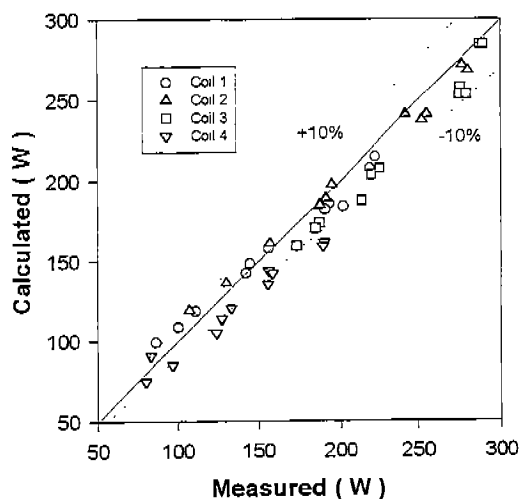

Figure 5. Comparison of the calculated and measured heat transfer rates.

RESULTS AND CONCLUSION

The heat transfer and pressure drop in the evaporator of a domestic refrigerator was investigated experimentally in this study. Correlations were developed for j and f. The repeatability of the test results was within $\mp 10\%$. A strong dependence of the heat transfer coefficients on the finning factor was observed. The convective heat transfer coefficient increases as the finning factor decreases. It was also noticed that the friction factor increases for higher fin spacing values.

ACKNOWLEDGEMENTS

REFERENCES

[1] Webb, R.L., "Air-side Heat Transfer in Finned Tube Heat Exchangers", Heat Transfer Engineering, Vol 1, No 3, 1980.
[2] Rite, R.W., "The Effect of Frosting on the Performance of Domestic Refrigerator-Freezer Finned Tube Evaporator Coils", M.Sc. Thesis, University of Illinois,1990.
[3] International Electrotechnical Commission (IEC) Standard 665, A.C Electric Ventilating Fans and Regulators for Household and Similar Purposes.
[4] Incropera, F.P., Dewitt, P.D., "Fundamentals of Heat and Mass Transfer", 3rd Edition, JohnWiley&Sons, 1990.
[5] Karataş, H., "A Theoretical and Experimental Study on a Domestic Refrigerator Evaporator", M.Sc. Thesis, Institute of Science and Technology, İstanbul Technical University,1995.
[6] Kayansayan, N., "Heat Transfer Characterization of Plate Fin-Tube Heat Exchangers" Int. J. of Refrig., Vol 17, No 1, 1994.