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Thermal Performance Investigation of Plain Finned-Tube Evaporators Used in Household Refrigerator

Evsel Bir Buzdolabında Kullanılan Düz Kanatlı Borulu Buharlaştırıcıların Isıl Performanslarının İncelenmesi

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Abstract

In this study, thermal performance of plain finned-tube evaporators in household refrigerators are investigated numerically and experimentally. The design parameters affecting on the cooling capacity are determined as air flowrate, evaporator temperature, tube alignment, number of tubes and number of fins. In the initial stage, the effects of those parameters on the cooling capacities are simulated by CoilDesigner software. The number of experiments that are necessary to obtain correlations is determined by Minitab software. The experiments are carried out in a no-frost household refrigerator in the off-mode. At the end of each experiment, air side heat transfer coefficients are calculated to generate a Nusselt (Nu) correlation as a function of the mentioned design parameters. The generated correlation of the cooling capacity can predict 95% of the experimental results within a confidence range of 15%. The correlation of the cooling capacity is converted to the Nusselt number correlation which is found to be consistent with available data in literature. The experiments reveal that the cooling capacity is dominantly affected by the evaporator temperature followed by air flow rate, tube alignment, number of tubes and the number of fins.

Keywords: *Plain Finned Tube Evaporator, Cooling Capacity, Design of Experiments, Nusselt Number Correlation, Numerical Model*

Öz

Bu çalışmada, düz kanatlı borulu buharlaştırıcıların ev tipi buzdolaplarındaki ısıl performansı sayısal ve deneysel olarak incelenmiştir. Soğutma kapasitesini etkileyen tasarım parametreleri, hava debisi, buharlaştırıcı sıcaklığı, boru düzeni, boru sayısı ve kanat sayısı olarak belirlenmiştir. İlk aşamada, bu parametrelerin soğutma kapasiteleri üzerindeki etkileri CoilDesigner yazılımı ile modellenmiştir. Isıl inceleme için gerekli olan deneylerin sayısı Minitab yazılımı tarafından simülasyon sonuçlarına göre belirlenmiştir. Deneyler ev tipi bir buzdolabında kapalı soğutma çevrimi olmaksızın yapılmıştır. Her deneyin sonunda hava tarafı ısı transfer katsayıları, söz konusu tasarım parametrelerinin bir fonksiyonu olarak ve Nusselt (Nu) korelasyonu oluşturmak için hesaplanır. Soğutma kapasitesi için oluşturulan korelasyon, deneysel sonuçların % 95'ini % 15'lik bir güven aralığında tahmin etmektedir ve literatürdeki mevcut verilerle tutarlı bulunmuştur. Deneyler, soğutma kapasitesinin baskın olarak buharlaştırıcı sıcaklığından ve ardından hava kütlesel debisinden, boru düzeninden ve kanat sayısından etkilendiğini ortaya koymuştur.

Anahtar Kelimeler: *Kanatlı Borulu Buharlaştıcı, Soğutma Kapasitesi, Deney Tasarımı, Nusselt Sayısı Korelasyonu, Sayısal Model*

1. Introduction

In the commercial refrigerators, cooling is usually carried out by passing air through a plain finned-tube type evaporator by a fan. Plain finned-tube evaporators can be in a staggered or inline alignment and they possess either continuous or discrete fins.

The most commonly used fin design for the finned-tube evaporator in refrigerators is the continuous type plain fins. These types of evaporators have wider fin pitches than general plain finned tube heat exchangers used in commercial applications (0.4 to 0.9 per cm). Their frontal areas are less than those of other general evaporator types since the interior volume of the refrigerator is a restriction. For these reasons, the thermal performances of the finned tube evaporators should be examined directly in the location at appropriate operating conditions.

In literature, evaporators are investigated mostly in specially designed test benches instead of test locations with real operating conditions. Kayansayan [1] carried out experiments to reveal the effect of surface geometries on finned tube heat exchangers. Ten different heat exchangers were investigated under different Reynolds numbers ranging from 100 to 30000 and finning factors between 11 and 23. Karatas et al. [2] conducted tests with four different heat exchangers at different air mass flow rates and temperatures. The air side heat transfer coefficients obtained from the experimental results were expressed in terms of Reynolds number and finning factor yielding the Colburn factor. It has been found that the finning factor has the most significant effect on the heat transfer coefficient, as confirmed by previous studies [1]. Wang and Chi [3, 4] experimentally examined the air-side performances of the finned-tube heat exchangers using 18 heat exchangers. A Colburn correlation was proposed depending on tube row numbers, fin spacing and tube outer diameters which can predict 88.6% of the database within the confidence range of 15%.

Kim et al. [5] found a new Colburn correlation that was obtained from the Colburn factors of 47 heat exchangers available in the literature. The suggested correlations expressed in Reynolds number, tube outer diameter, tube distances and fin spacing. The obtained correlation could predict 94% of all results with a deviation of 20%. Lee et al. [6] experimentally investigated air side heat transfer coefficients of spine finned-tube, discrete flat plate finned-tube, and continuous flat plate finned-tube evaporators used in household refrigerators. The spine finned-tube evaporator was found to have the highest air side heat transfer coefficient.

Barbosa et al. [7] experimentally investigated the thermal performances of household no frost heat exchangers based on air velocity and surface geometry. Experimental studies were carried out using a wind tunnel calorimeter and the effects of different tube rows, number of fins and fin spacings were tested. Melo et al. [8] examined the thermal performance of no frost heat exchangers in the house-hold refrigerator where the refrigerator itself was used as an experimental setup. Air side heat transfer coefficients were obtained for different air flow directions. Chen and Lai [9] studied the heat transfer coefficients of the two row and four finned- tube heat exchangers with staggered lines at different air velocities and different fin spacings in a wind tunnel.

Choi et al. [10] investigated the air side heat transfer coefficient of fins referenced to fin spacing, tube row numbers, fin alignments and vertical fin spacing using 34 samples. Paeng et al. [11] conducted both empirical and numerical studies to find the air-side heat transfer coefficient in heat exchangers with staggered tube alignment. The tube diameter was taken 10.2 mm the fin spacing was 3.5 mm and the Reynolds numbers varied between 1082 and 1649. The error rates between the correlations in the literature and the experimental results ranged from 7 to 34%, indicating the most consistent results to Kim et al. [5]. Tang and Yang [12] conducted experiments in a wind tunnel to investigate the heat transfer characteristics on a single row aluminum finned-tube heat exchanger. The Colburn factors were expressed in terms of Reynolds and Prandtl numbers.

In the studies in literature, the thermal performances of the evaporators have been usually tested in wind tunnels. Experimental studies to be carried out in real wind tunnels are more time demanding, therefore experiments can be realized in a refrigerator imposing the most realistic boundary conditions. For this reason, in the present study, the thermal performances of no frost evaporators were investigated in a household refrigerator with the actual boundary conditions rather than in a wind tunnel.

In the current parametric study, the effects of the evaporation temperature, air velocities and other design parameters such as tube alignment and number of tubes on the cooling capacity were intensely investigated to obtain air side heat transfer coefficients and Nusselt number correlations which were consistent with available data in literature. As a result, the effects of dominant governing parameters on the cooling capacities were revealed and robust correlation for the cooling capacities and Nusselt numbers were suggested for commonly used plain finned-tube evaporators. It should be pointed out that the cooling capacities and air flow rates have been normalized by the experimentally obtained cooling capacities and air flow rates of the household refrigerator.

2. Numerical Simulations

In the initial stage of the study, CoilDesigner software [13] was used to determine the parameters in experimentations. CoilDesigner is a heat exchanger simulation and optimization software first developed by University of Maryland and uses a segment-by-segment approach and in this way longitudinal conduction in tubes can be calculated. The operating conditions of the evaporator have been taken into consideration in determining the parameters to be investigated numerically. Simulation parameters such as evaporation temperature, air flow rate and geometric specifications are tabulated in Table 1.

Table 1. Simulation Parameters for one tube row

Parameters	
Air Flow Rate (-)	0.5, 1.0, 1.5, 2.0, 2.5
Evap. temp. (˚C)	-5, -15, -25
Number of tubes	12, 14, 16
Number of fins	33, 41, 63
Outer diameter (mm)	6.35, 8

In numerical simulations, ethylene-glycol (antifreeze-%60 by volume) and water mixture was chosen as the refrigerant and air was selected as the external fluid. The correlation proposed by Wang et. al [4] was used for the air side heat transfer correlation as given in Eq. 1.

$$\frac{Nu}{Pr^{1/3}} = 0.108Re^{0.71}\left(\frac{X_t}{X_l}\right)^{P1}\left(\frac{F_P}{D_C}\right)^{-1.084}\left(\frac{F_P}{D_h}\right)^{-0.786}\left(\frac{F_P}{X_t}\right)^{P2} \quad (1)$$

In Eq.(1) X_t ans X_l stand for transverse and longitudinal tube pitches, respectively. The Reynolds number Re is defined with respect to the hydraulic diameter D_h and Prandtl number Pr is constant. F_p stands for the fin pitch.

Dittus Boelter equation was selected for determining the refrigerant side heat transfer coefficient inside the tube as given in Eq. 2.

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \quad (2)$$

The comparisons in Figures 1-4 indicate the effects of design variables on the the cooling capacities, normalized by the reference design. The cases in Fig. 1 demonstrate the results of the simulations selected for inline tube alignment with 12 tubes and 8 mm outer diameter. As Figure 1 indicates, with increasing number of fins, the cooling capacities increased as well at all air flow rates.

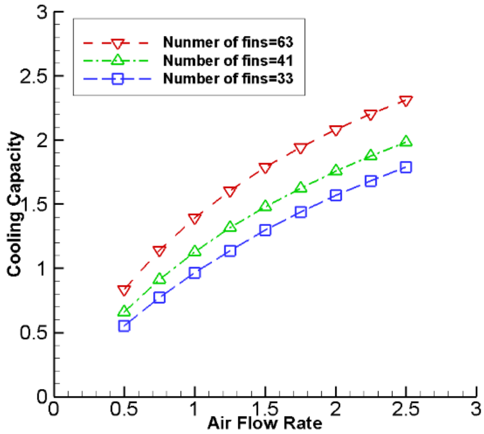

Figure 1. Effect of number of fins on cooling capacity.

Fig. 2 shows the effect of number of tubes on cooling capacities for the configuration of 33 fins, and 8 mm outer diameter with inline tube alignment. The simulations demonstrated, that the cooling capacities increased at all air flow rates if the number of tubes of the evaporator was increased.

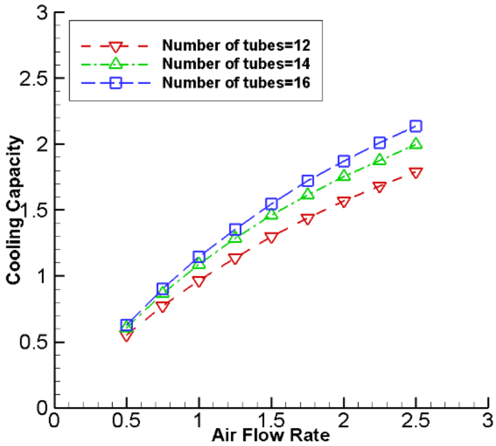

Figure 2. Effect of number of tubes on cooling capacity.

Fig. 3 shows the effect of tube alignments on cooling capacities. The simulation results have been indicated for the configuration of 33 fins and 12 tubes with 8 mm outer diameter. It was shown that cooling capacities increased if tube alignment changed from inline to staggered however, the tube's outer diameter had only negligible influence on the cooling capacities, especially in staggered alignment.

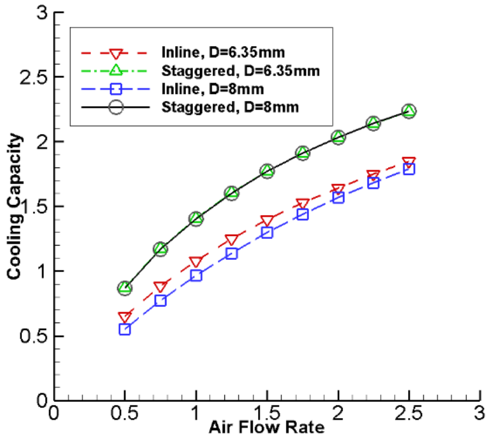

Figure 3. Effect of tube alignment on cooling capacity.

Figure 4 shows the effect of evaporator temperature on cooling capacities. If evaporator temperature was decreased from -5 °C to -15 °C, cooling capacities jumped and reached almost doubled values. It was concluded that decreasing the evaporation temperatures enhanced cooling capacities at all air flow rates.

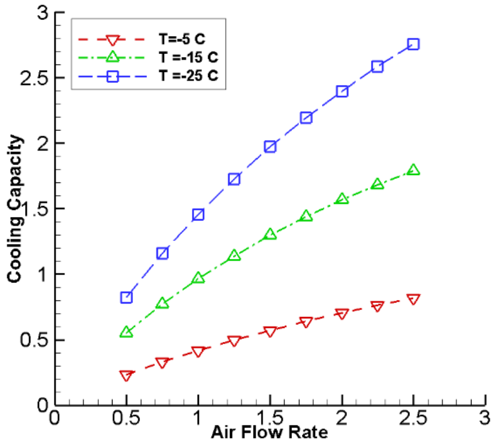

Figure 4. Effect of evaporator temperature on cooling capacity.

Numerical simulations provided an insight to understand the dominance of the parameters on the cooling capacities. It was found out that the

the evaporation temperature was the most dominant parameter on cooling capacity followed by air flow rate, tube alignment, number of tubes and number of fins. These simulations were crucial in designing the experiments and determining the necessary number of experiments.

3. Experimental Studies

3.1 Defining experimental design parameters

Using Minitab tool [14], the number of necessary experiments was determined with 5 factors and 2 level. The design parameters are tabulated in Table 2. In the experiments, eight evaporators have been used and their design specifications are tabulated in Table 3. Figure 5 shows the evaporator samples tested in experiments.

For 2 level and 5 factor experimental design, 32 experiments have to be carried out. However, if experiments might be conducted with the only selected subset, the number of experiments would be decreased. If 2 levels, 5 factors, and 3 repetitions were used in one experimental design, a total of 96 experiments would be necessary. Experiments had been performed with 1/2 fraction thus, the number of experiments was reduced to 48.

Table 2. Experimental design variables

Parameters	
Air Flow Rate (-)	0.5, 1.0
Evaporator temperature (°C)	-5, -15
Number of tube rows	1
Number of tubes	12, 16
Number of fins	33, 41
Tube outer diameter (mm)	6.35, 8

Table 3. Evaporator design specifications

Sample No	N_{tube}	N_{fin}	Tube Alignment	X_t (mm)	X_l (mm)
1	12	41	Staggered	22	20.67
2	12	33	Staggered	22	20.67
3	12	33	Inline	22	22
4	12	41	Inline	22	22
5	16	33	Staggered	22	20.67
6	16	41	Staggered	22	20.67
7	16	41	Inline	22	22
8	16	33	Inline	22	22

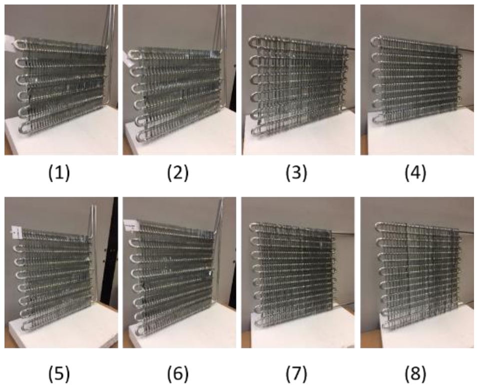

Figure 5. Evaporator samples tested in experiments

Figure 6 displays plain fin geometries for inline and staggered alignments where A represents the inline tube alignment and B represents the staggered tube alignment.

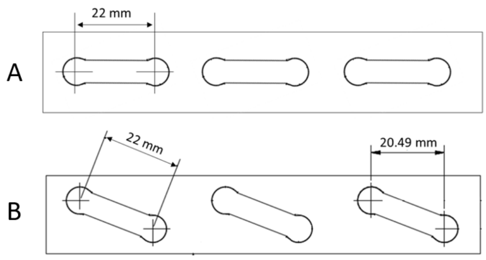

Figure 6. Plain fin geometries of the evaporators.

3.2 Experimental procedure

A refrigerator with a 310-liter fresh food compartment was adopted to a test bench. The ambient temperature was maintained at 24℃ (± 2°C) and the evaporators were used in the refrigeration cycle at off mode. In the measurements, a water bath was utilized instead of an actual cooling cycle, since the objective was to determine the air side heat transfer coefficients of the evaporators. The antifreeze-water mixture, conditioned at -5°C and -15°C in the water bath passed through the evaporator and the evaporator temperatures have been monitored. The connections between the water bath and the evaporator were insulated against heat transfer. The temperature of the antifreeze-water mixture was measured by T-type thermocouples located on the connection points between the water bath and evaporator. The mixture's flow rate was measured with a water flow sensor attached to the water line and controlled via Arduino Uno with 10% uncertainty.

Air passing through the evaporator was supplied by a fan at the original location as in the refrigerator. The air flow rate was measured with an annubar differential pressure measurement device connected to a micro-manometer placed in front of the fan. Air flow rates have been measured with 5% uncertainty. At the inlet and outlet of the evaporator, T- type thermocouples and relative humidity sensors have been used to measure temperatures and humidity, respectively. The uncertainty of T-type thermocouples was 0.2°C and the uncertainty of MEAS HTG 3500 relative humidity measurement devices was 3%. An electronic thermostat ensured that the temperatures remained at the set temperature in the fresh food compartment. An electronic thermostat controlled the cooling process by operating the fan in the desired temperature range. The temperature in the fresh food compartment was set to 5°C. The experimental setup is shown schematically in Figure 7. Uncertainties of the measurement devices in the setup are tabulated in Table 4.

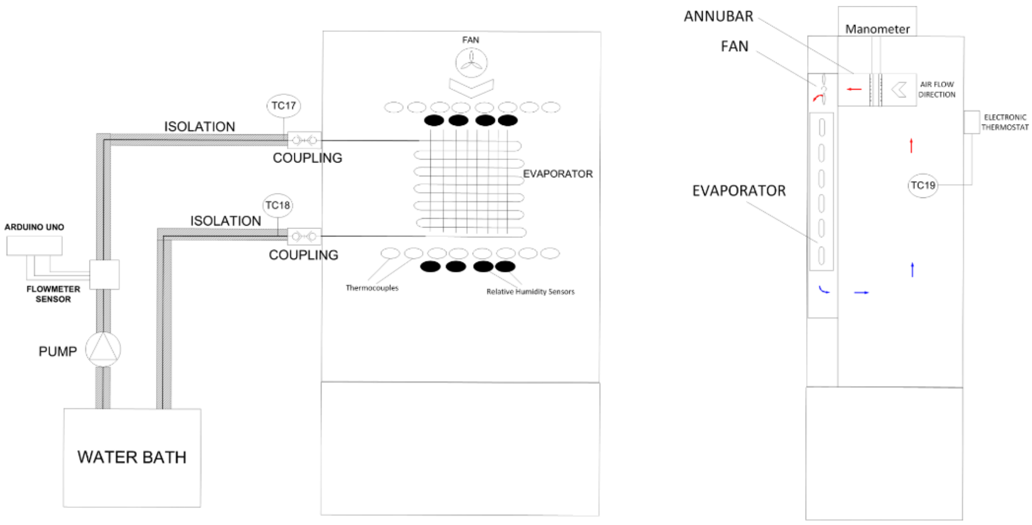

Figure 7. Sketch of experimental setup

The water bath regulated the temperatures of the evaporators at -5°C and -15°C. Temperature and relative humidity measurements have been recorded via data acquisition system every five seconds. The temperature in the fresh food cabin was observed and it fluctuated between 4°C and 5.5°C due to on and off modes of the fan during operation as shown in Figure 8. Thus, averaged air temperature and relative humidity at the inlet and outlet of the evaporators were calculated after quasi-periodic cycles have been reached.

Table 4. Measurement devices uncertainties

Measurement Device	Measuring Range	Uncertainty
Thermocouple	-200 to +300 °C	0.4 %
Relative humudity sensor	0 – 100 %	3 %
Annubar	0 – 8 l/s	5 %
Flowmeter	1 – 30 l/min	10%

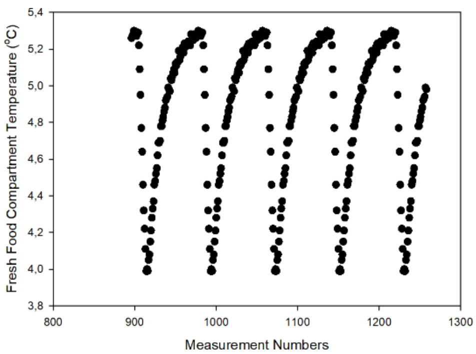

Figure 8. Temperature fluctuations in quasi-periodic state.

4. Results

4.1 Evaluation of the measurements

In order to calculate the cooling capacities, the measurements at the design points have been utilized. Air enthalpies at the inlet and outlet were obtained using the measured temperature and relative humidity. Each experiment was repeated three times hence; Table 5 shows averaged values of the measurements. Based on the enthalpy differences (air enthalpy at the inlet, h_{ai} and air enthalpy at the outlet, h_{ao}) and the mass flow rates of air ($\dot{m}_a$), cooling capacities of the heat exchangers (Q_e) have been calculated as given in Eq.(3). In Table 5, T_{ei} and T_{eo} stand for mixture's temperatures at the inlet and outlet of the evaporator, respectively. Also, V_a denotes nondimensional air flow rates such as 0.5 and 1.

$$Q_e = \dot{m}_a \left(h_{ai} - h_{ao} \right) \tag{3}$$

If the operating conditions were arranged in a way that the evaporation temperature of -15°C was combined with air flow rate equal to 1 with the number of tubes N_{tube} =16 and with the number of fins N_{fin} = 41 in staggered tube alignment, highest cooling capacity could be achieved among 16 configurations. Altering the evaporation temperature to -5℃ and reducing the air flow rate to 0.5 caused a significant decrease in the cooling capacity for the same configuration.

As Table 5 indicates, some of the measurements revealed almost identical results in terms of cooling capacities. The configuration in staggered tube alignment with N_{tube} = 12 and N_{fin} = 41 at the evaporation temperature of -5℃ and at air flow rate 1 and the same configuration at the evaporation temperature of -15℃ and at air flow rate 0.5 resulted in close cooling capacities. The measurement for N_{tube} = 12 and N_{fin} = 41 in inline tube alignment at the evaporation temperature of -5℃ and air flow rate 0.5 yielded the lowest cooling capacity.

It was found out that, the evaporation temperature together with the air flow rate should be called the most dominant factors in the increase of the cooling capacity. As a result, it was concluded that, the measurements at the lowest evaporation temperature and high air flow rates succeeded in achieving enhanced cooling capacities.

It should be pointed out that in the majority of measurements, the cases could be regarded as mean since they remained below an average of the maximum and minimum values which corresponded to Q_e = 0.738.

Table 5. Experimental results of the investigated cases.

N_{tube}	N_{fin}	Tube Alignment	T_{ei} (˚C)	T_{eo} (˚C)	V_a	Q_e
12	41	Staggered	-5.3	-4.8	1	0.485
12	33	Staggered	-15.5	-14.8	1	1.003
12	33	Inline	-5.4	-4.7	1	0.408
12	33	Inline	-15.5	-15.2	0.5	0.473
12	33	Staggered	-5.2	-5.0	0.5	0.297
16	33	Staggered	-15.4	-15.2	0.5	0.623
12	41	Inline	-5.3	-4.1	0.5	0.236
16	41	Inline	-5.2	-4.6	1	0.430
16	33	Inline	-15.5	-14.6	1	1.056
16	41	Inline	-15.6	-15.3	0.5	0.527
16	41	Staggered	-15.2	-14.5	1	1.241
16	33	Staggered	-5.3	-4.6	1	0.522
12	41	Inline	-15.3	-14.7	1	0.967
16	41	Staggered	-5.4	-5.2	0.5	0.333
12	41	Staggered	-15.4	-15.2	0.5	0.519
16	33	Inline	-5.3	-5.0	0.5	0.247

4.2 Logarithmic mean temperature approach

Cooling capacity can also be obtained from the refrigerant side with the logarithmic mean temperature approach given in Equation (4) where UA represents overall thermal conductance.

$$Q_e = UA\Delta T_{lm} \tag{4}$$

ΔT_{lm} represents logarithmic temperature differences of the refrigerant and can be calculated by Equation (5).

$$\Delta T_{lm} = \frac{(T_{ai} - T_{ei}) - (T_{ao} - T_{eo})}{ln\frac{(T_{ai} - T_{ei})}{(T_{ao} - T_{eo})}} \tag{5}$$

T_{ai} and T_{ao} denote the air inlet and air outlet temperatures, respectively. The heat transfer characteristics of each plain finned-tube evaporator were written in terms of air side heat transfer coefficients which can be derived from Equation (6).

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_h(A_p + \eta_f A_f)} + R_C \tag{6}$$

A_i, A_p, and A_f denote tube's total inner area, evaporator tube's outer area and fin's total area, respetively. Air side heat transfer coefficient is denoted by h_h. The heat transfer coefficient of antifreeze-water mixture side denoted by h_i can be found from Dittus Boelter equation and the fin efficiencies η_f can be calculated from Schmidt approximation [7]. R_c is the contacting resistance. Nu and Re for the evaporators are defined in Equations (7) and (8), respectively where Nu is based on tube's outer diameter (D_0) and Re is based on the hydraulic diamater (D_h). G in Re represents air mass flux in kg/m^2s.

$$Nu = h\,\frac{D_o}{k} \quad (7)$$

$$Re = \frac{G\,D_h}{\mu} \quad (8)$$

Nu can be expressed in terms of Re and the geometric parameters of the evaporator, thus a correlation in the form of Equation (9) was obtained which can be found in literature. As in Eq.(1) X_t ans X_l stand for transverse and longitudinal tube pitches, respectively and ϵ represents the finning factor.

$$\frac{Nu}{Pr^{0.33}} = A\,Re^{B}\,(\frac{X_t}{X_l})^{C}\epsilon^{D} \quad (9)$$

The coefficient and the exponents in Eq.(9) were solved using the nonlinear regression tool in Minitab which yields Equation (10). This Nu correlation is compared to the experimental results as shown in Fig. 9.

$$\frac{Nu}{Pr^{0.33}} = 0.000141789Re^{1.28}\,(\frac{X_t}{X_l})^{6.51}\varepsilon^{1.12} \quad (10)$$

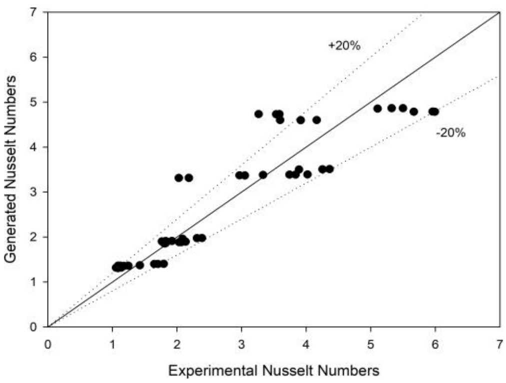

Figure 9. Comparison between experimentally obtained Nu and the correlation.

4.3 Mathematical method: ε - NTU

The validity of the generated Nu correlation was also tested by establishing the ε-NTU method. Air side outlet temperature is given in Equaiton (11) for the ε-NTU method. In Eq. (11) C_{min} and C_h stand for minimum thermal capacity and thermal capacity of hot refrigerant, respectively.

$$T_{ao} = T_{ai} - \varepsilon\,\frac{C_{min}}{C_h}(T_{ai} - T_e) \quad (11)$$

It is known from the literature that the effectiveness (ε) varies between 50% and 75% for single pass heat exchangers [15]. The ratio of the thermal capacities of cold (C_c) and hot refrigerants (C_h) is defined in Equation (12) and NTU in Equation (13) can be obtained using the overall thermal conductance.

$$\frac{C_C}{C_h} = \frac{m_c cp_c}{m_h cp_h} \quad (12)$$

$$NTU = \frac{UA}{C_{min}} \quad (13)$$

Since the thermal capacity of the antifreeze-water mixture is higher than that of air, the effectiveness of the evaporator can be calculated by Eq. (14) [15].

$$\varepsilon = 1 - e^{-NTU} \quad (14)$$

Equation (11) was iteratively solved until the convergence criteria of 10^{-4} °C was satisfied for the outlet temperature of air. Comparison of the cooling capacities obtained from Nu correlation with the current measurements and the correlation by Wang et. al. [4] is shown in Figure 10. The generated Nu correlation succeded in predicting 95% of the experimental results in a confidence range of 15% and was found sufficiently consistent with the correlations in literature.

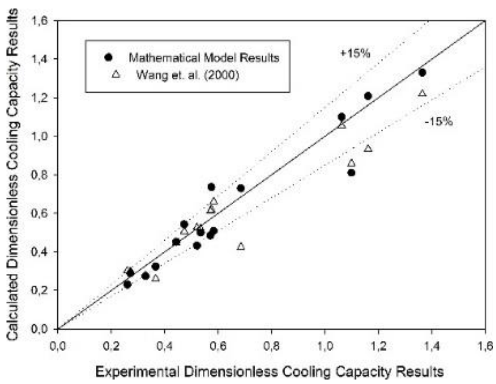

Figure 10. Comparison between experimentally obtained cooling capacity and generated correlation.

The design restirctions for the correlation in the current study and Wang's correlation are tabulated in Table 6.

Table 6. Design restrictions for the current and Wang's correlation.

Correlation	Number of Rows	Fin Pitch (F_p) (mm)	Tube Transverse Distance(X_t) (mm)	Tube Longitutional Distance (X_l) (mm)
Wang	1 -6	1.19 – 8.7	17.7 – 31.75	12.5 – 27.5
Current study	1	8 – 10	22	20.49 - 22

It should be pointed out that the generated correlation in the current study is only valid for one row plain finned tube evaporator. Compared to Wang's correlation, the ranges for the fin pitch and tube longitutional distance are in a narrower range. Because of that reason the current correlation can predict cooling capacity more accuratly than Wang's correlation. On the other hand, experiments have been carried out in the refrigerator's fresh food compartment instead of in a wind tunnel as Wang et al. [3] did, thus Nu correlation in the current study is appropriate to these design restrictions.

5. Conclusion

In the present study, single row finned-tube heat exchangers used in the fresh food compartment of household refrigerators have been subjected to parametric thermal examinations. An actual available refrigerator was used as a test setup. Preliminary simulations have been carried out to predict the effect of the design variables on enhancing the cooling capacity. Based on the selected design parameters, measurements have been conducted for a total of 16 cases with various configurations. The cooling capacity was considered as the target function to be maximized. The most promising results have been obtained for the evaporation temperature of -15°C and the normalized air flow rate equal to 1. A Nu correlation was developed from the experimentally obtained air side heat transfer coefficients and the cooling capacities have been recalculated form this correlation. The measurements and the generated correlation have been found in excellent agreement with the available correlations in literature.

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References

[1] N. Kayansayan, "Heat transfer characterization of flat plain fins and round tube heat exchangers," *Experimental thermal and fluid science*, vol. 6, no. 3, pp. 263–272, 1993.

[2] H. Karatas, E. Dirik, and T. Derbentli, "An experimental study of air-side heat transfer and friction factor correlations on domestic refrigerator finned-tube evaporator coils," *International Refrigeration and Air Conditioning Conference*, 1996.

[3] C.-C. Wang and K.-Y. Chi, "Heat transfer and friction characteristics of plain fin-and-tube heat ex- changers, part i: new experimental data," *International Journal of Heat and Mass Transfer*, vol. 43, no. 15, pp. 2681–2691, 2000.

[4] C.-C. Wang, K.-Y. Chi, and C.-J. Chang, "Heat transfer and friction characteristics of plain fin-and- tube heat exchangers, part ii: Correlation," *International Journal of Heat and mass transfer*, vol. 43, no. 15, pp. 2693–2700, 2000.

[5] N. Kim, B. Youn, and R. Webb, "Air-side heat transfer and friction correlations for plain fin-and- tube heat exchangers with staggered tube arrangements," *Journal of heat transfer*, vol. 121, no. 3, pp. 662–667, 1999.

[6] T.-H. Lee, J.-S. Lee, S.-Y. Oh, M. Lee, and K. Lee, "Comparison of air-side heat transfer coefficients of several types of evaporators of household freezer/refrigerators," *International Refrigeration and Air Conditioning Conference*, 2002.

[7] J. R. Barbosa Jr, C. Melo, C. J. Hermes, and P. J. Waltrich, "A study of the air-side heat transfer and pressure drop characteristics of tube-fin no-frostevaporators," *Applied Energy*, vol. 86, no. 9, pp. 1484– 1491, 2009.

[8] C. Melo, R. O. Piucco, and P. O. Durate, "In-situ performance evaluation of no-frost evaporators," *International Refrigeration and Air Conditioning Conference*, 2006.

[9] H.-T. Chen and J.-R. Lai, "Study of heat-transfer characteristics on the fin of two-row plate finned-tube heat exchangers," *International Journal of Heat and Mass Transfer*, vol. 55, no. 15-16, pp. 4088–4095, 2012.

[10] J. M. Choi, Y. Kim, M. Lee, and Y. Kim, "Air side heat

transfer coefficients of discrete plate finned- tube heat exchangers with large fin pitch," *Applied Thermal Engineering*, vol. 30, no. 2-3, pp. 174–180, 2010.

[11] J. G. Paeng, K. H. Kim, and Y. H. Yoon, "Experimental measurement and numerical computation of the air side convective heat transfer coefficients in a plate fin-tube heat exchanger," *Journal of mechanical science and technology*, vol. 23, no. 2, pp. 536–543, 2009.

[12] S. Tang and K.-T. Yang, "Thermal performance of a single-row fin-and-tube heat exchanger," *Journal of thermal science*, vol. 14, no. 2, pp. 172–180, 2005.

[13] "Coildesigner, heat exchanger design software," Center for Environmental Energy Engineering, 4164, Glenn L. Martin Hall, Mechanical Engineering Department, University of Maryland, College Park, MD 20742, USA, 2007.

[14] Minitab Inc. 2010. Gage Studies for Continuous Data. http://www.minitab.com/uploadedFiles/Documents/samplematerials/TrainingSampleMeasurementSystemsMTB16EN.pdf (Date of Access: 17.04.2016).

[15] T. L. Bergman, F. P. Incropera, D. P. DeWitt, and A. S. Lavine, *Fundamentals of heat and mass transfer*. John Wiley & Sons, 2011.