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NUMERICAL ASSESSMENT OF THERMOHYDRAULIC PERFORMANCE OF A DEEP FREEZE EVAPORATOR

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Abstract: Refrigerators have become indispensable for domestic and industrial purposes, with the improvement in the standard of living and manufacturing. A well-designed refrigerator provides low energy consumption and noise, have low price and large capacity of storage. Improving of the evaporator heat transfer performance is one of the most important ways for reducing the energy consumption. Therefore, this study performs determination of the influences of the fin spacing and air velocity on heat transfer coefficient, cooling capacity and pressure drop of airside of the evaporator in order to improve the energy efficiency of refrigerator. The evaporator is modeled numerically and several simulations have been done to find airside thermohydraulic performance. The airflow is assumed to be incompressible, Newtonian and laminar under the constant wall temperature boundary conditions on external surfaces of the evaporator pipes. From the numerical results, when the fin spacing is increased, the airside heat transfer coefficient is increased but heat transfer is decreased because of less total area. It has been found that as the air inlet velocity increases the heat transfer coefficient increases. The distance between fins of the evaporator is found to have considerable effect on pressure drop. The fin spacing is decreased pressure drop is increased. For provide the required cooling capacity, the most suitable fin spacing is found as 5 mm at velocity of 0,5 m/s.
The study provides a comparison between various evaporator constructions at less time with numerical analyze instead of fabricating a prototype that is an expensive application to choose the proper evaporator for the required cooling capacity in the design of the refrigeration system of a refrigerator.
Keywords: Evaporator, heat exchanger, fin spacing, heat transfer coefficient, refrigerating performance

BİR DERİN DONDURUCU BUHARLAŞTIRICISININ ISIL HİDROLİK PERFORMANSININ SAYISAL DEĞERLENDİRİLMESİ

Özet: Standart yaşam ve üretimdeki gelişme ile buzdolapları, ev tipi ve endüstriyel amaç için vazgeçilmez hale gelmiştir. İyi tasarlanmış bir buzdolabı, düşük enerji tüketimi ve az gürültü sağlar, düşük fiyat ve geniş depolama kapasitesine sahiptir. Buharlaştırıcının ısı transfer performansının arttırılması, enerji tüketiminin azaltılması için en önemli yollardan birisidir. Bu yüzden, bu çalışma buzdolabının enerji etkenliğini geliştirmek için buharlaştırıcının ısı transfer katsayısının, soğutma kapasitesinin ve hava tarafı basınç düşüşü üzerine kanat aralığı ve hava hızının etkilerinin belirlenmesini sağlar. Buharlaştırıcı sayısal olarak modellendi ve hava tarafı ısıl hidrolik performansını bulmak için çeşitli benzetimler yapıldı. Hava akışı, buharlaştırıcı borularının dış yüzeyinde sabit duvar sıcaklığı sınır şartı altında sıkıştırılamaz, Newtonian ve laminer olduğu farzedilmiştir. Sayısal sonuçlardan, kanat aralığı arttığında hava tarafı ısı transfer katsayısı artar fakat daha az toplam alan nedeniyle ısı transferi azalır. Hava giriş hızı arttığında hava tarafı ısı transfer katsayısının arttığı bulundu. Buharlaştırıcının kanatları arası mesafe, basınç düşüşü üzerine önemli bir etkiye sahip olduğu bulundu. Kanat aralığı azaldığında, basınç düşüşü artar. Gerekli soğutma kapasitesini sağlamak için, en uygun kanat aralığı, 0,5m/s hızda 5mm'dir.
Çalışma, buzdolabının soğutma sisteminin tasarımında gerekli soğutma kapasitesi için uygun buharlaştırıcı seçimi için pahalı bir uygulama olan prototip üretimi yerine sayısal analizle daha az zamanda çeşitli buharlaştırıcı yapıları arasında karşılaştırma sağlar.
Anahtar Kelimeler: Buharlaştırıcı, ısı değiştirici, kanat aralığı, ısı transfer katsayısı, soğutma performansı

NOMENCLATURE

A	total area [mm^2]
A_{min}	minimum flow area [mm^2]
c_p	specific heat [kJ/kgK]
d_o	tube outside diameter [mm]
d_i	tube inside diameter [mm]
d_h	hydraulic diameter [mm]
f	friction factor
G_{max}	mass velocity [kgm^2/s]

H	evaporator height [mm]
h_o	airside heat transfer coefficient [W/m^{2}K]
j	Colburn factor
k	thermal conductivity [W/mK]
L	total length of the evaporator [mm]
$\dot{m}$	mass flow rate[m/s]
N	tube row number
Nu	Nusselt number
Pr	Prandtl number
$\dot{Q}$	heat transfer rate [W]
Re	Reynolds number
s	fin spacing [mm]
S_l	longitudinal tube pitch [mm]
S_t	transverse tube pitch [mm]
St	Stanton number
t	fin thickness [mm]
T	temperature [K]
U	overall heat transfer coefficient [W/m^{2}K]
V	air inlet velocity [m/s]
x,y,z	cartesian coordinates

Greek Symbols

ΔP	pressure drop [Pa]
μ	dynamic viscosity [Pa.s, kg/ms]
ρ	density [kg/m^3]
σ	minimum flow area/face area

Subscripts

i	inlet
o	outlet
m	mean

INTRODUCTION

Refrigerators are extensively used to store foods to preserve its freshness and keep it safe. Because of great importance in people's life, a more efficient refrigeration system must be obtain to provide better thermal performance of refrigerator. Evaporator is the component of the refrigeration system of a refrigerator, which is working according to vapor compression system as seen in Figure 1. The thermohydraulic performance of the evaporator considerably affects efficiency of the refrigeration system capacity. Investigate the thermohydraulic performance of the evaporator is very important for people's daily life to decrease electricity consumption and for new refrigerator evaporator designs.

Heat exchangers in refrigeration and air conditioning applications can be categorized according to whether they are coils or shell and tube exchangers. Evaporator and condenser coils are used when the second fluid is air because a much longer surface area is required on the airside [21]. Evaporator coil that is a compact heat exchanger is generally limited with the heat transfer coefficient of the airside. A lot of active and passive method is developed for increase heat transfer performance of the airside with decrease heat exchanger volume and manufacturing costs because of lower heat

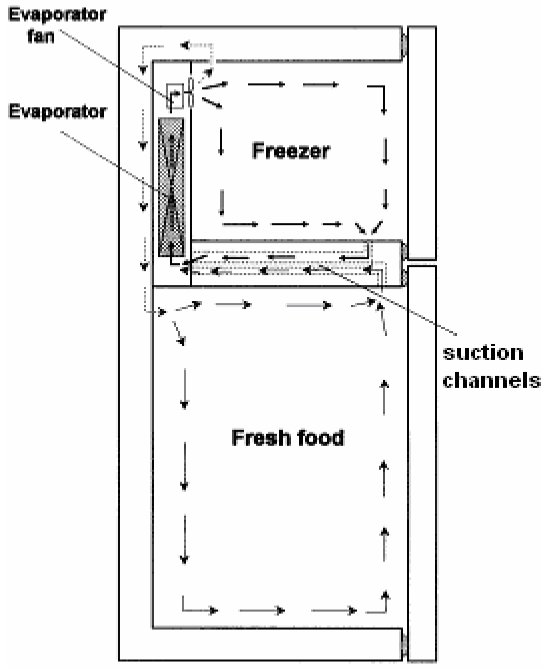


Figure 1. Refrigerator airflow system and evaporator (http://www.konveyor.com, May 2006).

transfer coefficient of the airside than the fluid side. In the active method, heat transfer can be improved by giving additional energy to the fluid but in the passive method the improvement can be obtain with changing the geometry of the evaporator (fin spacing, fin thickness, tube diameter and tube row) without giving additional energy.

Finned tube heat exchangers have been extensively investigated experimentally and numerically in recent years by many researchers. Horuz et al (1998), analyzed theoretical and experimental performance of plate fin and tube evaporators. The experimental evaporator was analyzed in account with the most common and widely used correlations together with the parameters of air velocity, fin spacing, tube diameter, evaporator temperature, refrigerant type and frost height. Erek et al. (2000) investigated the influence of the changes in fin geometry on heat transfer and pressure drop of a plate fin and tube heat exchanger, numerically. They've been modeled only one tenth segment of the fin due to the symmetry. Mon and Gross (2004), investigated the effects of fin spacing on four row finned tube heat exchanger in staggered and in-line arrangements by three dimensional numerical study. According to the results, the boundary layer developments and horseshoe vortices between the fins are found to be substantially dependent on the fin spacing to height ratio and Reynolds number. In plate finned tube heat exchangers, the effect of fin spacing on heat transfer and friction factor investigated experimentally by Rich (1973). Heat exchangers of 22 configurations with a variation of fin pitch, number of tube row, and tube alignment is tested to provide experimental data that can be used in the optimal design of flat plate finned-tube heat exchangers with large fin pitch by Kim Mand Kim (2005). It was found that the airside heat transfer coefficient decreased

with a reduction of the fin pitch and an increase of the number of tube row. As the fin pitch decreased from 15.0 to 7.5 mm over the Reynolds number range of 500-900, the reduction in the heat transfer coefficient of four row heat exchanger coil was approximately 10%. Rocha et al (1997) investigate two-dimensional heat transfer analysis is performed in one- and two-row tubes and plate fin heat exchangers (circular and elliptical sections), using experimentally determined heat transfer coefficients from a heat and mass transfer analogy. They found that a relative fin efficiency gain of up to 18% is observed in the elliptical arrangement, as compared to the circular one and the efficiency gain, combined with the relative pressure drop reduction of up to 25% observed in previous studies. The effect of fin spacing, tube rows and tube diameter on heat transfer and pressure drop of a plate fin and tube heat exchanger is determined experimentally by Wang and Chi (2000). Chen et al. (2005) present a set of equations, which can be used to predict the performance parameters of an evaporator, when there is an oblique angle between the inlet air velocity and frontal face of the evaporator. Tutar and Akkoca (2004), numerically studied the effects of governing parameters, such as fin spacing, Reynolds number, tube row number and tube arrangement on the heat transfer and flow characteristics of a plate fin and tube heat exchanger. An experimental study has been carried out to investigate the heat transfer and pressure drop characteristics of fin and tube exchangers with plate, wavy and louvered fin surfaces by Yan and Sheen (2000). Chen et al. (2007) predict the average heat transfer coefficient and fin efficiency on the vertical square fin of one circular tube plate finned tube heat exchangers for various air speeds and fin spacing. In plate finned evaporators, the heat transfer on the pipes with and without fins depend on fin spacing, length of heat exchanger and the number of tube row have been investigated experimentally by Altınışık et al.(1999). Halıcı et al.(2001), investigated experimentally the effect of the number of tube rows and air velocity on heat, mass and momentum transfer for finned tube heat exchangers which consist of aluminum fins and copper tubes. The effect of variation of the fin geometry on the egg-crate type evaporator that is forced flowed fin and tube heat exchanger have been studied by Bansal et al (2001). Byun et al.(2006), investigated airside heat transfer and pressure drop of the finned tube heat exchangers for four different fin types experimentally.
Yang et al (2006), obtained optimum fin spacing of a fin tune heat exchanger of a domestic refrigerator under frosting condition. Optimization a heat exchanger with staggered finned circular and elliptic tubes to maximize the total heat transfer rate studied numerically and experimentally by Matos et al.(2004). Airside pressure drop in plate finned tube heat exchangers established experimentally by Jacimovic et al.(2006).

In this study deep freeze evaporator of a refrigerator company that is not used the name of company due to the commercial anxious is shown in Figure 2. In literature, the studied models consist of only two fins

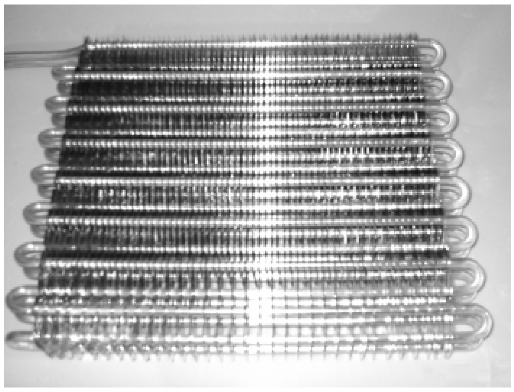


Figure 2. Finned tube evaporator of a refrigerator.

(Erek et al., 2002; Mon and Gross, 2004; Matos et al., 2004) due to the symmetry whereas whole evaporator is modeled in this study and analyzed numerically. A numerical model of the thermal system refers to a discretized representation of the system on a computer, which may be used to determine the behavior and characteristics of the system. This process of studying the behavior of the system by means of a model, rather than by fabricating a prototype, which is an expensive and time-consuming endeavor, is known as simulation. The results obtained allow us to consider many different design possibilities as well as a variety of operating conditions (Jaluria, 1998). In this study, the attention is focused on the thermo hydraulic performance of evaporator for evaluation of effects of fin spacing and air velocity.

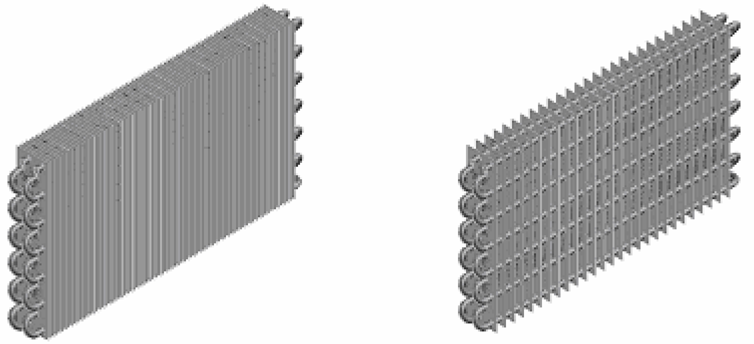
GEOMETRIC MODEL

In this study, the whole of the finned tube evaporator has been modeled as shown in Figure 3, by using Gambit software also developed by CFD code Fluent that is very easy to use and still has sufficiently powerful capabilities. The refrigerators use plate finned tube evaporators with large fin spacing. In Figure 3a, the smallest evaporator fin spacing, s = 5 mm, that is usually seen in refrigerator. But in this study 2 mm fin spacing has been modeled for compared the heat transfer characteristics.

The studied models, which are shown in Figure 4, consist of 26 horizontal parallel pipes with same diameter and length and 25 inclined U pipes and 9 configurations modeled with different fin spacing. Geometric characteristics of these configurations are given in Table 1.

MATHEMATICAL MODELING

A mathematical model is an abstract model that uses mathematical language to describe the behavior of a system. Continuity, momentum and energy equations constitute the mathematical model in this system. The mathematical model is based on the following assumptions.



(a) (b)

Figure 3. Finned tube evaporator isometric view a) s =5 mm b) s = 15 mm.

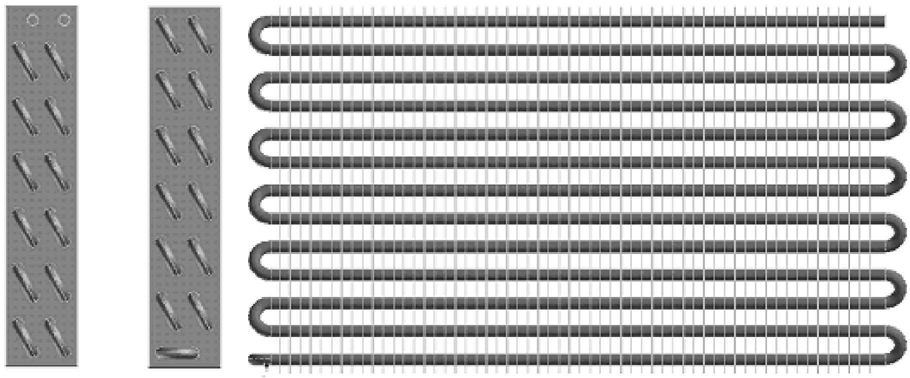


Figure 4. Front, back and left side views of the finned tube evaporator.

Table 1. Geometric characteristic of evaporator configurations.

Parameter	Value
Face dimension (mmxmm)	50x535
Tube outside diameter (mm)	8
Length of each finned tubes (mm)	485
Number of finned tubes	26
Fin spacing (mm)	2, 5, 6, 7, 8, 9, 10, 12.5, 15, bared tube
Fin thickness (mm)	0.19
Transverse tube pitch (mm)	22
Longitudinal tube pitch (mm)	19.053
Row arrangement	Staggered
Tube material	Aluminum
Fin material	Aluminum

(a) The studied model assumed that there is no frosting or condensation on the outside surface of the evaporator.

(b) The heat transfer coefficient on the airside has been uniformed.

(c) Conduction in the fin has been neglected.

(d) The airflow is treated to be three dimensional, steady, incompressible and laminar due to the low Reynolds number of the flow.

(e) Thermo physical properties of air have been assumed to be constant.

According to these assumptions, governing equations are written in Cartesian coordinates as follows:

Continuity

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

Momentum

$$\rho \frac{D\mathbf{V}}{Dt} = -\nabla P + \mu \nabla^2 \mathbf{V} + \rho \mathbf{G} \tag{2}$$

Energy

$$\rho c_p \frac{DT}{Dt} = k \nabla^2 T \tag{3}$$

Reynolds number is given by the equation,

$$\mathrm{Re} = \frac{v d_h}{\mu / \rho} \tag{4}$$

where v is velocity, μ is dynamic viscosity and ρ is density of air. Hydraulic diameter, d_h, is expressed as,

$$d_h = 4\frac{LA_{\min}}{A} \tag{5}$$

where L is total length of the evaporator, $A_{\min}$ and A minimum flow area and total area, respectively. The airside heat transfer coefficient was calculated by using Colburn factor,

$$j = St\,\mathrm{Pr}^{2/3} = \left(\frac{h_o}{\rho v c_p}\right)\left(\frac{\mu c_p}{k}\right)^{2/3} \tag{6}$$

A correlation by Gray and Webb [19] for a four-row plate finned tube heat exchangers as follows:

$$j_4 = 0.14\,\mathrm{Re}^{-0.328}\left(\frac{S_t}{S_l}\right)^{-0.502}\left(\frac{s}{D}\right)^{0.0312} \tag{7}$$

Here, S_t is transverse tube pitch and S_l is longitudinal tube pitch. If the row number is less than four, the equation (7) is modifying by a factor as follows:

$$\frac{j_N}{j_4} = 0.992\left[2.24\,\mathrm{Re}^{-0.092}\left(\frac{N}{4}\right)^{-0.031}\right]^{0.607(4-N)} \tag{8}$$

where N is total fin number of the evaporator. Total heat transfer rate, $\dot{Q}$, is defined as follows:

$$\dot{Q} = \dot{m}c_p\left(T_i - T_o\right) \tag{9}$$

where T_i and T_o, inlet and outlet temperature of the air, c_p is specific heat at constant pressure, $\dot{m}$ is mass flow rate and is given as

$$\dot{m} = v\rho A_{\min} \tag{10}$$

The overall heat transfer coefficient can be obtained from the following equation:

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{\ln\left(d_o/d_i\right)}{2\pi kL} + \frac{1}{h_o A_o} \tag{11}$$

The pressure drop is calculated from following equation proved by Kays and London (1984).

$$\Delta P = \frac{G_{\max}^2}{2\rho_i}\left[\left(1+\sigma^2\right)\left(\frac{\rho_i}{\rho_o}-1\right) + f\frac{A_o}{A_{\min}}\frac{\rho_i}{\rho_m}\right] \tag{12}$$

f, is the friction factor given as follows:

$$f = 0{,}1243\,\mathrm{Re}_{dh}^{-0{,}2095} \tag{13}$$

NUMERICAL MODELING

Numerical models are based on the mathematical model and allow one to obtain, using a computer, quantitative results on the behavior of a system for different operating conditions and design parameters (Jaluria, 1998). In this study, finite volume method, by means of commercial CFD software program Fluent; have been used to solve the governing equations numerically. The simulations are carried out for low Reynolds numbers on the airside due to the relatively small air velocities and the choice of the model summarized in Table2.

Table 2. Model characteristics.

Model	Settings
Space	3D
Time	Transient
Viscous	Laminar
Wall treatment	No slip condition
Heat transfer	Neglected

The mesh is based on 3D unstructured tetrahedral elements provided by pre-processor software Gambit for all evaporator configurations. The solution scheme for the governing mathematical equations is given at Table 3.

Table 3. Solution scheme.

Variable	Scheme
Pressure	Standart
Pressure-Velocity Coupling	SIMPLE algorithm
Momentum	First Order Upwind
Energy	First Order Upwind

For the specified system, the velocity and the temperature conditions on the surfaces have to described due to the defined problem includes fluid flow and heat transfer. Air enters from the bottom of the evaporator, so the velocity inlet boundary condition is defined for this surface and air inlet temperature is given as 246 K. The outlet boundary condition is applied to the top of the evaporator. Due to the no slip condition, the velocity along the evaporator pipes and fins surfaces was zero. Temperature was constant at 253 K on the external surface of the evaporator pipes since phase change in the pipes.

The properties of the aluminum that is material of pipes and fins of the evaporator and air are summarized in Table 4.

RESULTS

The effects of fin spacing and air velocity on the airside performance of the finned tube evaporator have been investigated numerically. The heat transfer coefficient, pressure drop, friction factor and heat transfer that called performance parameters are summarized in Table

5 and the results are presented graphically in this section.

Table 4. Properties of the materials (FLUENT).

	Air	Aluminum
Density (kg/m^3)	1,408	2719
Thermal conductivity (W/mK)	0,02227	202,4
Specific heat (J/kgK)	1,005324	871
Molecular weight (kg/kgmol)	28,966	-
Viscosity (kgm/s)	$1,5177.10^{-5}$	-

Figure 5 demonstrates the effect of fin spacing and air inlet velocity on the airside heat transfer coefficient of the evaporator. While the fin spacing is increasing 2 to 15 mm, airside heat transfer coefficient is increasing, too. The airside heat transfer coefficient of the finless evaporator is given at the Table 5. It is observed that the airside heat transfer coefficient of a barred tube bundle is greater than a finned tube evaporator. At the same graphic, the increase of air velocity is to cause increasing the airside heat transfer coefficient. Therefore, finned tube evaporator of the refrigerator can be designed smaller size.

Figure 6 present the effect of varying the fin spacing according to the heat transfer on the airside. Although increase of the heat transfer coefficient on the airside, heat transfer is considerably decreased because of less total area when fin spacing is increased. For larger fin spacing, heat transfer is substantially occurred around the tube, the effect of the fins can be neglected. In highest velocities, heat transfer is significantly increased due to the more turbulence. It can be seen in Table 5, at the smallest number of fins, N = 27, heat transfer is 28, 39594 W whereas at the N=197 the heat transfer increases to about 145, 73190 W. Results are in good agreement with the experimental results of the Horuz et al. (1998) and Kim and Kim (2005).

In Figure 7, variations of the airside pressure drop at the different air velocities are given according to the fin spacing. One can observe that as fin spacing decreases, since may create problem for free movement of air though the evaporator, pressure drop through the evaporator increases which corresponds to the increase in the number of fins. At the same graphic as expected, increase of the separation of the flow around the tube and vortices causes an increase in the pressure drop with the rising air inlet velocity. Results are in good agreement with the data from the Jacimovic et al (2006).

Table 5. Performance parameters of evaporator configurations.

Performance Parameters	Fin Spacing (mm)	Fin Numbers	Velocity (m/s)		
			0,1	0,3	0,5
Heat transfer Coefficient	Finless	0	9,619	18,224	23,995
	2	197	7,882	14,970	19,867
	5	80	8,521	15,739	21,701
	6	66	8,869	15,824	21,853
	7	57	8,976	16,141	22,134
	8	49	9,050	16,350	22,238
	9	45	9,114	16,638	22,353
	10	40	9,133	16,773	22,428
	12,5	32	9,190	16,955	22,640
	15	27	9,283	17,081	22,803
Pressure Drop	Finless	0	0,084	0,493	1,202
	2	197	0,416	1,854	3,854
	5	80	0,187	0,826	1,771
	6	66	0,167	0,773	1,705
	7	57	0,154	0,728	1,623
	8	49	0,145	0,682	1,524
	9	45	0,139	0,662	1,487
	10	40	0,134	0,659	1,494
	12,5	32	0,121	0,611	1,401
	15	27	0,115	0,593	1,375
Friction Factor	Finless	0	0,642	0,057	0,050
	2	197	0,027	0,024	0,025
	5	80	0,040	0,031	0,028
	6	66	0,040	0,032	0,029
	7	57	0,041	0,033	0,029
	8	49	0,043	0,033	0,031
	9	45	0,044	0,034	0,031
	10	40	0,045	0,036	0,032
	12,5	32	0,045	0,037	0,033
	15	27	0,046	0,038	0,033
Heat Transfer	Finless	0	12,179	13,342	15,136
	2	197	145,732	167,658	194,909
	5	80	65,433	78,518	79,670
	6	66	56,909	66,622	67,291
	7	57	51,201	58,382	58,156
	8	49	45,424	52,650	50,597
	9	45	42,003	47,640	46,870
	10	40	36,962	42,574	41,952
	12,5	32	31,507	36,489	35,726
	15	27	28,396	35,188	31,996

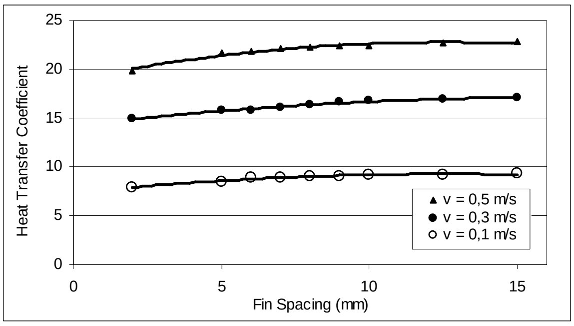


Figure 5. The variation of the heat transfer coefficient of the airside with respect to the fin spacing at different velocities.

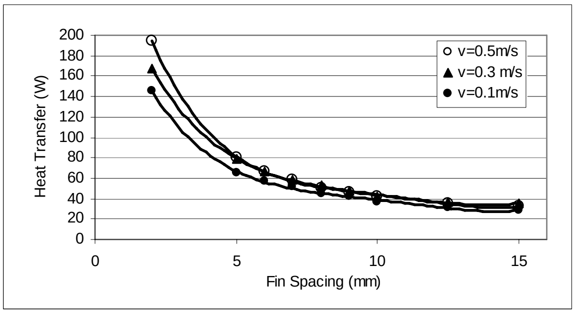


Figure 6. The variation of the heat transfer with respect to the fin spacing at different velocities.

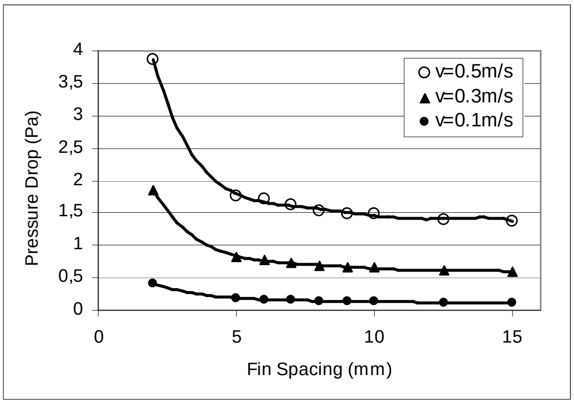


Figure 7. The variation of the pressure drop of the airside with respect to the fin spacing at different velocities.

The effect of fin spacing on the friction factor at different velocities is shown in Figure 8. The friction factor increases for larger values of the fin spacing of the evaporator and on a barred tube bundle, maximum values of the friction factor were also noticed in Table 5.

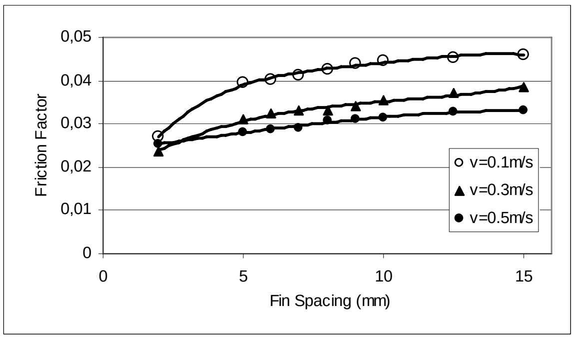


Figure 8. The variation of the friction factor with respect to the fin spacing at different velocities.

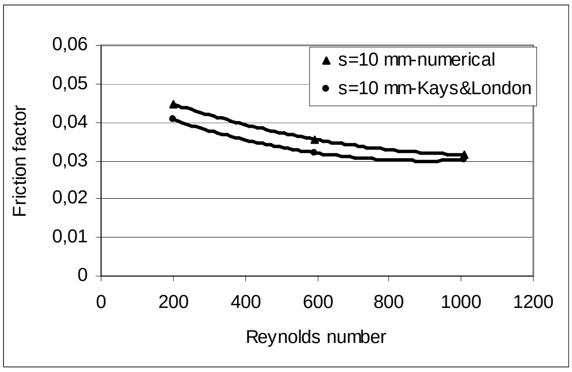


Figure 9. The variation of the friction factor with respect to the fin spacing in comparison with the correlation given by Kays and London at different Reynolds numbers.

Figure 9 gives the alteration of the friction factor according to Reynolds number at 10 mm fin spacing. When Reynolds number is increased 200 to 1000, friction factor is fairly decreased. To verify validness of the model has 10 mm fin spacing, the numerical results are compared with correlation given for finned tube heat exchanger by Kays and London (1984). In this comparison the difference between the results is become smaller when the Reynolds number is approximated 1000.

Table 6. Cooling capacities.

	Fin spacing (mm)	Air velocity (m/s)		
		0,1	0,3	0,5
Cooling Capacity (W)	2	147,412	170,466	197,125
	5	68,255	80,128	83,698
	6	60,689	67,967	71,006
	7	55,114	61,076	63,497
	8	48,592	54,681	56,088
	9	45,481	51,220	53,823
	10	40,872	46,096	48,754
	12,5	36,024	40,658	42,096
	15	32,985	38,583	39,96
	Finless	16,717	18,349	23,766

The variation of the cooling capacity respect to the fin spacing at different velocities is given at Table 6. While the fin spacing increases, cooling area increases on account of more finned surface hence cooling capacity of the evaporator linearly decreases. It has seen that cooling capacity is higher when velocity increased however augmentation on the cooling capacity is becomes less at high velocities.

CONCLUSION

In this study, the investigation of the effects of fin spacing and air velocity to heat transfer coefficient, pressure drop and friction factor in the airside of the finned tube deep freeze evaporator has been performed numerically.

Numerical results are summarized that fin spacing increases, heat transfer rate and pressure drop decreases but heat transfer coefficient increases and when the air inlet velocity increases, heat transfer coefficient and pressure drop increases. It was found that, the most suitable fin spacing for provide the required cooling capacity is found as 5 mm at velocity of 0,5 m/s.

By means of numerical analyze, different finned tube evaporator models have been compared in shorter time than the other methods in this study. The results analyzed numerically are in good agreement with previous experimental and theoretical studies.

Designers will continue to investigate heat transfer on both refrigerant and airsides of the evaporator and they'll determine the complex conjugate heat transfer on the finned tube deep freeze evaporator with sufficient accuracy.

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