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INVESTIGATION OF THERMAL PERFORMANCE OF FINNED TUBE EVAPORATOR UNDER DRY CONDITIONS

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Abstract: In this study, analysis of thermal performance of finned tube evaporator under dry conditions has numerically been done. The heat convection coefficient in air and refrigerant sides were calculated by considering their varying properties. In addition, the surface temperature of fin and tubes along the fin were calculated. A numerical model was developed in order to calculate the overall thermal permeability (UA) in transient regime. The numerical model was solved by using Matlab R2010 program. The overall thermal permeability (UA) was examined under different air inlet temperatures and air velocity. In addition, variation of the convection heat transfer coefficient with respect to quality at the refrigerant side was examined.
Keywords: finned tube, Evaporator, Heat transfer, Dehumidify, Modeling, Transient state, Steady state.

KANATLI BORULU EVAPORATÖRÜN KURU ŞARTLAR ALTINDA ISIL PERFORMANSININ İNCELENMESİ

Özet: Bu çalışmada kanatlı borulu bir evaporatörün ısıl performansının analizi kuru şartlar için sayısal olarak yapılmıştır. Havanın ve soğutucu akışkanın değişen özelliklerine göre hava ve soğutucu akışkan tarafındaki taşınım katsayısı hesaplanmıştır. Ayrıca kanat boyunca yüzey sıcaklıkları ve boru yüzey sıcaklıkları hesaplanmıştır. Geçici rejimde evaporatörün toplam ısıl geçirgenliğini (UA) hesaplamak için bir sayısal model oluşturulmuş ve sayısal model Matlab R2010 bilgisayar programında çözülmüştür. Toplam ısıl geçirgenliği, havanın farklı giriş sıcaklıkları ve hızlarında incelenmiştir.Ayrıca kuruluk derecesine göre soğutucu akışkan tarafındaki taşınım katsayısının değişimi incelenmiştir.
Anahtar Kelimeler: Kanatlı boru, Evaparatör, Isı transferi, Nemini almak, Modelleme, Geçici hal, Kararlı hal.

NOMENCLATURE

A	area [m^2]
B_0	boiling number
c_p	specific heat [J /kg K]
D_h	hydraulic diameter [m]
h	convection heat transfer coefficient [W/m^2K]
h_{fg}	enthalpy of evaporation [J kg^{-1}]
h_L	liquid to liquid convection heat transfer coefficient [W/m^2K]
h_{TP}	two phase heat transfer coefficient [W/m^2K]
h_r	local evaporation heat transfer coefficient [W/m^2K]
k	thermal conductivity [W/mK]
L_{fin}	length of the fin [m]
M_a	mass per meter of air [kg/m]
m	mass [kg]
$\dot{m}$	mass flow rate [kg /s]
m_w	mass flow from tube wall [kg]
Nu	Nusselt number [=hL/k]
q	heat transfer [W]
q''	heat flux [W /m ^{2}]
Pr	Prandtl number [=$c_p\mu/k$]
r	radius [m]
Re	Reynolds number [=$u\rho d/\mu$]
w_{fin}	fin width [m]
T	temperature [^oC]
U	The overall heat transfer coefficient [W/ m ^{2}K]
$U_{t,r}$	coefficient of total heat transfer from tube to refrigerator [W /m ^{2}K]
UA	The overall heat transfer coefficient of evaporator [W/ K]
X	degree of dryness [kg /kg]
x,y,z	coordinate [m]

Greek letters

η	efficiency
ρ	density [kg /m^3]
μ	dynamic viscosity [Ns/m^2]

Subscripts

a	air
fin	fin
i	inside

o outside
r refrigerant
s surface
T total
t tube
tab tab
w wall

INTRODUCTION

Evaporators are devices that enable heat to be drawn from an area to be cooled through evaporation of a refrigerating fluid. In the no-frost refrigerators that work on the principle of air circulation cooling, finned tube tube evaporators are used. Heat exchangers which operate at below freezing temperature are subjected to frost deposition and progressive buildup of frost. The presence of frost leads to the degradation of heat exchanger's performance in terms of lower heat transfer and higher pressure drop (Tso *et al*., 2006).

There are very few studies in literature on the non anticipated frost formation under dry conditions. Mc Quiston (McQuiston, 1978) developed the equations for heat transfer and friction factor used in finned tube heat exchangers for both dry and wet surfaces. Wang et al. (Wang *et al*.,1997) investigated the effects of fin slope, number of fin rows and relative humidity on performance of heat transfer taking place under dry conditions. Mirth and Ramadhyani (Mirth and Ramadhyani, 1993) carried out a study involving investigation of mass and heat transfer properties of wavy finned- heat exchangers. Pirompugd et al. (Pirompugd *et al*., 2005, Pirompugd *et al*., 2006) studied a tube-by-tube reduction method for simultaneous heat and mass transfer characteristics for plain fin-and-tube heat exchangers in dehumidifying conditions. The researchers presented a new method and they concluded that the results depend on fin slope and relative humidity.

One of the parameters that characterizes heat transfer in finned tube heat exchangers is the total heat transfer coefficient. Şeker et al. (Seker *et al*.,2004), investigated the variation of total heat transfer coefficient of finned tube heat exchangers under frost conditions at different air inlet temperatures.

In this study, evaporator performance was numerically investigated under dry conditions where no frost formation exists.

The effects of air inlet temperature and its velocity on total thermal permeability were studied. Temperature distribution over the fin surface was given. In addition; coefficient of heat transfer for local evaporation of the refrigerant R 22, and the quality based on Shah (Kakaç, 1998) principle were also studied.

NUMERICAL MODELING AND CALCULATION PROCEDURE

A numerical model in transient state was established for calculating the fin-tube evaporator's UA value under dry conditions. In forming the numerical model, thermodynamics first law, energy balance equations, Fourier heat transfer and Newton's law of cooling as well as some empirical expressions were used. In the formed model, the evaporator's total heat permabilities for different air inlet temperatures and air velocities were calculated under dry conditions. In the model, partial differential equations were solved with the Finite Differences Method. Flow area was divided with nodal points into equal intervals as shown in Figure 1. The solution was achieved by forming centers of control volumes for these nodal points and then applying the Finite Differences Method.

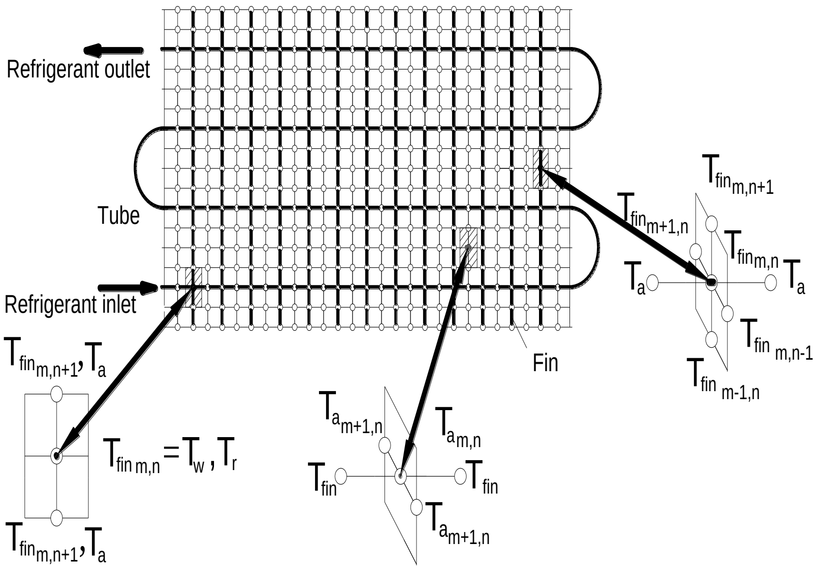

Figure 1. Cell division of flow area for numerical solution.

The numerical algorithm prepared for the numerical model in the analysis of performance of finned tube evaporator under dry conditions was given in the flow chart in Figure 2. Solution for the numerical algorithm shown in Fig. 2 was achieved with the Gauss-Seidel iteration method. The coil was divided into 583 control volumes and computation was started with estimated air conditions and refrigerant's quality in every cell. First the temperature on refrigerant side was calculated. In the next iteration, the fin surface temperatures and tube surface temperatures were calculated with the help of boundary conditions. Real air temperatures were used based on the calculated fin and tube surface temperatures; then compatibility of these values with respect to previous values was checked. If the values do not cope, then the refrigerant and surface temperatures were recalculated by using real air temperatures. The cycle was repeated for a time interval of 1 minute.

In the formed numerical model the following assumptions were considered:

1. The model is at a quası-steady state. The numerical model was studied within a finite time interval. This situation provides possibility to a steady state system.

2. Due to the fact that heat transfer is dominant along the fin, axial heat transfer on the pipe surface is negligible.

3. The surface temperature on the tubing arrangement is regarded as variable.

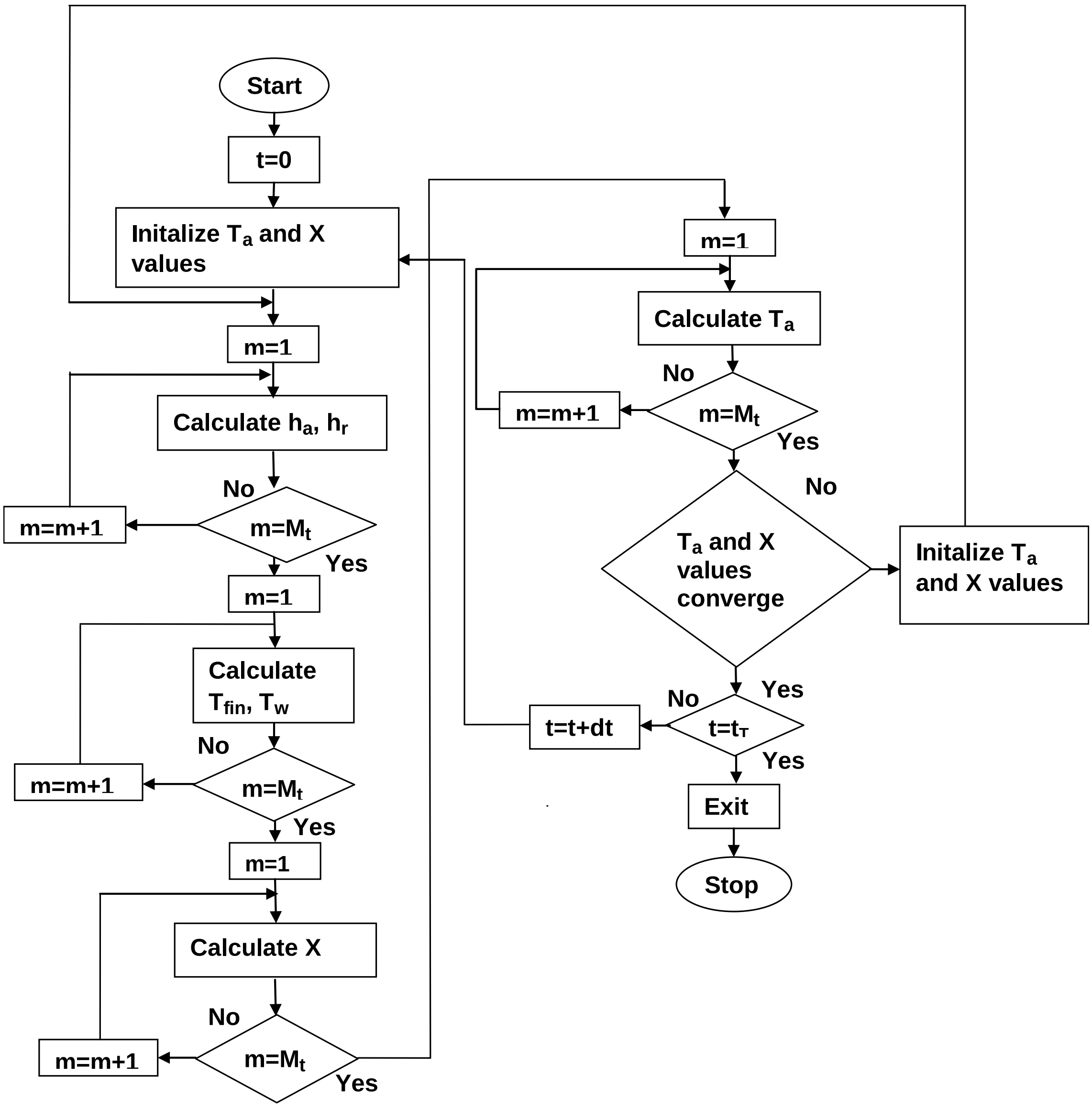

Figure 2. Flow chart for numerical algorithm.

Energy Balance on Fin Side

Energy balance on fin side can be given by;

$$m_{fin}\, c_{p,fin}\, \frac{\partial T_{fine}}{\partial t} = k_{fin}\, A_{tab}\, \frac{\partial^2 T_{fine}}{\partial x^2}\, w_{fin}$$

$$+ k_{fin}\, A_{tab}\, \frac{\partial^2 T_{fin}}{\partial y^2}\, L_{fin} + 2\, A_{fin}\, h_{fin} (T_a - T_{fin}) \qquad (1)$$

where m_{fin} and $c_{p,fin}$ are fin mass and specific heat respectively.

Energy Balance on Tube Side

Energy balance on tube side can be given in the following form;

$$\mathrm{m_w} \,.\, \mathrm{c_{p.w}} \cdot \frac{\partial \mathrm{T_w}}{\partial \mathrm{t}} = q_{\mathrm{t}} + q_{\mathrm{fine}} - U_{tr}\, A_i\, (T_w - T_r) \qquad (2)$$

where $c_{p,w}$, m_w , q_{fine} and $U_{t,r}$ are, respectively specific heat of tube, mass of the tube, heat transferred from fin to tube and the total heat transfer coefficient from tube to the refrigerant. U_{tr} can be found from equation (3), U_{tr} is written by considering the tube inner cross section.

$$U_{t,r} = \frac{1}{\dfrac{A_i \ln(r_0 / r_i)}{2\pi k_t L} + \dfrac{1}{h_r}} \qquad (3)$$

Here q_t is the heat transfer rate from air to tube and is given by:

$$q_t = h_{tube}\, A_0 (T_a - T_{tube}) \qquad (4)$$

where h_{tube} indicates the coefficient of convection heat transfer on the tube side .

Energy Balance on Air side

Energy balance on air side can be given with equation (5) .

$$\mathrm{M_a c_{p.a}}\, \frac{\partial \mathrm{T_a}}{\partial \mathrm{t}} + \dot{m}_a\, c_{p,a}\, \frac{\partial \mathrm{T_a}}{\partial y} = h_a\, \frac{A_{fine}}{L_{fine}} (T_a - T_{fine}) \qquad (5)$$

where M_a and $c_{p,a}$ are the air mass per unit length and specific heat of air respectively.

Calculating Quality

The quality can be calculated with equation (6), as follows,

$$\mathrm{m_r} \,.\, \mathrm{C_{p.r}} \cdot \frac{\partial \mathrm{T_r}}{\partial \mathrm{t}} + \dot{m}_r\, h_{fg} \cdot \frac{\partial \mathrm{X}}{\partial x} = A_i \cdot \frac{U_{t,r}}{L} (T_w - T_r) \qquad (6)$$

where m_r, $c_{p,r}$ and h_{fg} are respectively, mass of the refrigerant, specific heat and and latent heat of evaporation.

Coefficient of convection on refrigerant side

Shah equation, which contains four dimensionless parameters, can be applied in to the core, convection and split boiling zones. The equation is given (Kakaç, 1998), as below

$$\Psi = \frac{h_{TP}}{h_L} \qquad (7)$$

where h_{TP} is two phase heat transfer coefficient, h_L is liquid-liquid convection heat transfer coefficient for liquid phases. The Ψ value was calculated with respect to Ns dimensionless parameter value.

For Ns > 1,

$$\Psi_{cb} = \frac{1.8}{N_s^{0.8}} \qquad (8)$$

$$\Psi_{nb} = 230 B_0^{0.5} \qquad B_0 > 0.3\mathrm{x}10^{-4}$$

$$\Psi_{nb} = 1 + 46 B_0^{0.5} \qquad B_0 < 0.3\mathrm{x}10^{-4} \qquad (9)$$

B_0 is boiling number. Ψ is the largest as compared to Ψnb, and Ψcb.

For 0.1< Ns< 1;

$$\Psi_{bs} = F B_0^{0.5} \exp(2.74 N_s^{-0.1}) \qquad (10)$$

Ψ is the largest as compared to Ψbs, and Ψcb.

As for Ns ≤ 0.1;

$$\Psi_{bs} = F B_0^{0.5} \exp(2.74 N_s^{-0.15}) \qquad (11)$$

Ψ is the largest as compared to Ψbs, and Ψcb.

Coefficient of convection on air side

The empirical formulas obtained by Yang, Lee and Song were experimentally given in equation (12) and (13). In this study, these expressions for coefficient of convection on air side were used by considering the tube and fin geometries (Yang *et al.*, 2006).

$$\overline{Nu}_{fin} = \frac{\bar{h}_{fin} L}{k_a} = 0.204\, \mathrm{Re}_L^{0.657}\, \mathrm{Pr}^{1.334} \qquad (12)$$

$$\overline{Nu}_{tube} = \frac{\bar{h}_{tube} D_h}{k_a} = 0.146\,\mathrm{Re}_L^{0.917}\,\mathrm{Pr}^{2.844} \qquad (13)$$

where h_{fin} and k_a are the convection heat transfer coefficient and thermal conductivity for the fin and air sides respectively. Here h_{tube} and D_h respectively, indicate the coefficients of convection heat transfer on the tube side and the hydraulic diameter.

The overall heat transfer coefficient

The overall heat transfer coefficient of the evaporator is written as

$$U = \frac{1}{\dfrac{1}{\eta_T h_{tube}} + \dfrac{1}{h_r \dfrac{A_i}{A_T}}} \qquad (14)$$

CONCLUSION

In this study, the numerical model was solved by using the Mat lab computer program. As seen in Figure 3, as the air inlet temperature increases, the total heat transferability decreases. This result is consistent with Yan (Yan *et al.*,2003). As the air temperature drops Reynolds number increases. And this causes an increase in the coefficient of convection on the air side. As the coefficient of convection on the air side increases, the UA value increases too. The studies in literature have shown that with time, formation of frost decreases the UA value (Lenic *et al.*, 2009, Yan *et al.*,2003, Yang *et al.*, 2006, Tso *et al.*,2006,). Under dry conditions, this value increases with time.

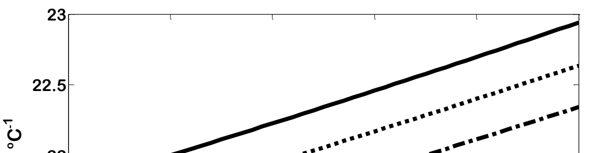

Figure 3. Variation of total thermal permeability at different air inlet temperatures for $v_{a,i}$=0.7 m/s.

This stuation has shown that frost formation increases thermal resistance, lowers the heat amount the refrigerating fluid absorbs and increases energy consumption as a result of frost formation in between the fins. High air velocities cause corresponding high heat transfer ratio and as expected, this leads to an increase in UA value (Fig.4). This results complies with the results of Yan (Yan *et al.*,2003) and Seker (Seker *et al.*,2004). In this study, R 22 refrigerant was used and evaporation heat transfer coefficient was investigated based on Shah Principle, it was found that the quality increased with the heat transfer coefficient (Fig. 5).

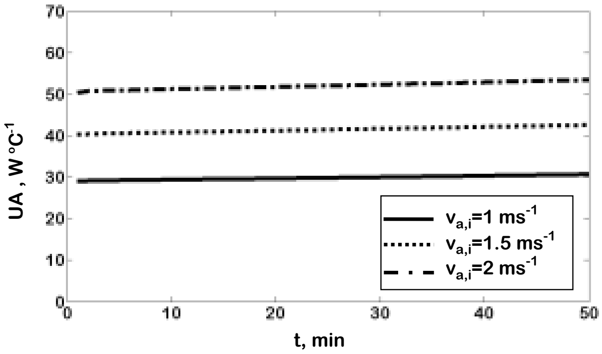

Figure 4. Variation of total thermal permeability at different air velocities for $T_{a,i}$=5^{o}C.

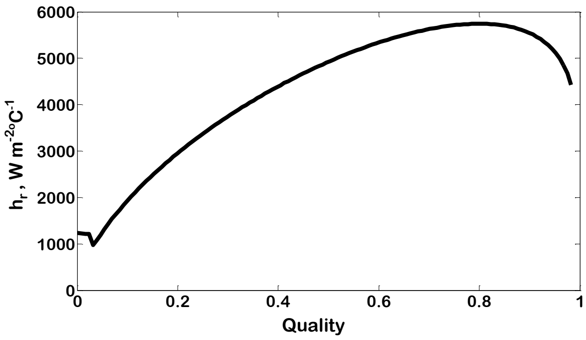

Figure 5. Variation of evaporation heat transfer coefficient with the quality.

Temperatures of the finned surfaces were studied in two dimensional basis, and the temperature variation along the finned surface shown in Fig. 6. As seen in the figure, temperature takes its minimum value at the fin root while at the middle of the fin the temperature rises to its maximum value. The temperature, then increases based on the increase in the quality of the refrigerant throughout the fin.

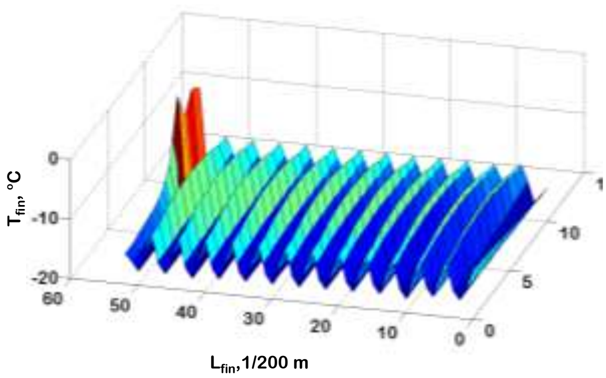

Figure 6. Temperature variation along fin surface for $T_{a,i}$=10^{o}C and $v_{a,i}$=1m/s.

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