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Performance of a Plate Fin and Tube Dehumidifying Coil Using Entropy Generation Method

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Abstract

The objective of the present work is to apply the concept of entropy generation method to evaluate the performance of a plate fin and tube dehumidifying coil. The present study provides an analytical method to model the working and performance of this coil as an independent part from the system without modeling the other parts that the system consisted of under wet and dry surface conditions for variable operating parameters. A comparison was made between the traditional effectiveness of the coil and the entropy generation model to evaluate the performance of the coil for variable conditions. The study found that the entropy generation algorithm is more efficient and more sensitive to parameters change than traditional effectiveness to describe the performance of a dehumidifying coil depending on the design and operating parameters. The combination of operating parameters of the coil such as air velocity, air wet bulb temperature, air dry bulb temperature, air relative humidity and heat load capacity affected the performance of the coil and the variable operating parameters lead to variable performance, so the air velocity should be varied according to the state of entering air and the total space load during operation time to compensate the loss in performance and to save power consumed.

Keywords: Plate fin and tube; dehumidifying coil; entropy generation

اداء ملف ازالة الترطيب نوع انبوب- زعنفه مستويه بأستخدام طريقة تولد الأنتروبيه

الخلاصه

الهدف من الدراسه هو تطبيق مفهوم طريقة تولـد الأنتروبيه (Entropy Generation) لأيجـاد اداء ملف ازالة الترطيب نوع انبوب- زعنفه مستويه. الدراسه الحاليه تقدم طريقه تحليليه لمماثلة عمـل وأداء هذا الملف كوحده مستقله عن المنظومه بدون مماثلة الأجزاء الأخرى التي تتكون منها المنظومه في حالـة جفاف او رطوبة سطح الملف ولعدة متغيرات تشغيليه. تمت مقارنة الفعاليه التقليديه للملف مع نموذج تولـد الأنتروبيه لوصف اداء الملف في ظروف متغيره. الدراسه وجدت ان طريقة تولد الأنتروبيه اكثـر كفـاءه وأكثر حساسيه للمتغيرات من الفعاليه التقليديه لوصف اداء ملف ازالة الترطيب اعتمادا علـى المتغيـرات التشغيليه والتصميميه. مجموعه المتغيرات التشغيليه للملف مثل سرعه الهواء , درجة الحـراره الرطبـه , درجة الحراره الجافه , الرطوبه النسبيه والحمل الحراري تؤثر على اداء الملف وهذه العوامـل التشـغيليه

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المتغيره تؤدي الى تغير الأداء لذلك فأن سرعة الهواء يجب ان تتغير طبقا الى حالة الهواء الداخل والحمـــل الحراري خلال وقت التشغيل لتعويض النقصان بألأداء ولتوفير الطاقه المستهلكه.

Introduction

The plate fin and tube dehumidifying coil mainly consists of plain aluminum fins with round copper tubes. Air is flowing between fins and the other fluid passing inside tubes according to specified circuiting arrangement as shown in Figure 1. The fluid inside tubes may be chilled water or an evaporated refrigerant. Mass production of dehumidifying coils in industrial, commercial and residence applications encouraged researchers to find new and efficient designs. The performance evaluation of these coils could be used as a tool to enhance the efficiency of existing coils and to design new coils. The coil effectiveness is a traditional method to find the performance of the coil as an independent part from the system. In many cases the dehumidifying coil is modeled as a part of refrigeration system and the coefficient of performance (COP) of the cycle is calculated only as a measure of efficiency without considering the efficiency of each element of the system. Chwalowski et al. **[1]** compared the capacity of an evaporator coil predicted by three computer modeling algorithms and manufacturer's catalog with experimental data. Flat and V-shaped coils used in an air conditioning system were tested at different saturation temperatures with various face angles of the flat coil. Oskarsson et al. **[2]** suggested three models to simulate the working and

performance of a plate fin and tube evaporator under dry, wet and frosted conditions of the air side. The results of simulation using finite element, three regions and parametric models were compared with experimental data for a single and six rows evaporator. Domanski et al. **[3]** presented a comparable evaluation of a number of refrigerants as working fluids in an optimized finned tube evaporator. EVAP-COND **[4]** simulation package was used to find the performance of the evaporator. A wide range of circuiting arrangements was considered and the impact of the evaporator performance on the cycle's coefficient of performance (COP) was analyzed. Byun et al. **[5]** study the performance of a finned tube evaporator using tube by tube analysis scheme. Fin shape, heat exchanger type, inner tube surface and type of refrigerant were considered and examined under variable operating parameters such as air velocity, inlet air temperature and relative humidity. The objective of the present work is to apply the concept of entropy generation method to find the performance of a plate fin and tube dehumidifying coil as an independent part from the system. The entropy is generated due to heat exchange and due to pressure drop comes from fluid flow. Increasing of entropy generation or irreversibility in the system leads to higher loss of work, so the minimization of entropy generated enables the system performs better.

Theory

The modeling of a plate fin and tube evaporator depends on the heat balance between wall surface and entering air taking into account the heat and mass transferred. Heat and mass balance between entering and exiting air are considered also. The performance of the evaporator is predicted depending on the concepts of the first law and second law of thermodynamics.

Modeling of a plate fin and tube dehumidifying coil

Assuming that the total heat load of the evaporator (Q_T) is transferred from air to fins and bare tubes surfaces,

$$Q_T = (h_T A_t + h_T A_f \eta_f)(H_m - H_w) \quad ....(1)$$

Total heat load of the evaporator consists of sensible and latent parts as follows,

$$Q_T = Q_{sen} + Q_{lat} \quad ....(2)$$

$$Q_{sen} = (h_{sen} A_t + h_{sen} A_f \eta_f)\Delta T_m \quad ......(3)$$

$$Q_{sen} = m^{\bullet} Cp(T_{di} - T_{do})$$

$$T_{do} = T_{di} - \frac{Q_{sen}}{m^{\bullet} Cp} \quad .....(4)$$

$$Q_{lat} = m^{\bullet} H_{fg}(W_i - W_o)$$

$$W_o = W_i - \frac{Q_{lat}}{m^{\bullet} H_{fg}} \quad ....(5)$$

where,

$$A_t = \pi . D . L . N_t . N_r .[1 - F_t . F_d]$$

$$A_f = 2 F_d . L . N_t . N_r . \left[X_a X_b - \frac{\pi . D^2}{4} \right]$$

$$H_m = \frac{H_i + H_o}{2}$$

Assuming counter flow heat exchanger, the mean temperature difference between wall and outside air can be calculated as,

$$\Delta T_m = \frac{(T_{di} - T_w) - (T_{do} - T_w)}{\ln\left[\frac{T_{di} - T_w}{T_{do} - T_w}\right]}$$

$$\Delta T_m = \frac{T_{di} - T_{do}}{\ln\left[\frac{T_{di} - T_w}{T_{do} - T_w}\right]} \quad ......(6)$$

McQuiston [6] presented one of the well known correlations to predict air side heat transfer coefficient taking into account the effect of number of rows as follows,

$$h_{sen} = \frac{j_{n,sen} . Cp . G_{max}}{Pr^{2/3}} ,$$

$$h_T = \frac{j_{n,T} . G_{max}}{Sc^{2/3}} \quad ..... (7)$$

where,

$$\frac{j_n}{j_4} = \frac{1 - 1280 \cdot N_r \cdot Re_b^{-1.2}}{1 - 5120 \cdot Re_b^{-1.2}}$$

$$j4_{sen} = 1.325*10^{-6} + 0.2675 \cdot jp . j_s \quad \textbf{[2]}$$

$$j4_T = 1.325*10^{-6} + 0.2675 \cdot jp . j_T$$

$$jp = Re_D^{-0.4} . \left(\frac{A_o}{A_t}\right)^{-0.15}$$

$$j_s = 0.84 + 4*10^{-5}.Re_S^{1.25}$$
$$j_T = (0.95 + 4*10^{-5}.Re_S^{1.25}).Fs^2$$
and

$$Re_b = \frac{G_{max} \cdot X_b}{\mu}, \qquad G_{max} = \frac{\rho_{in} \cdot U}{\sigma},$$

$$\sigma = \frac{A_{min}}{A_{face}}, \qquad Re_D = \frac{G_{max} \cdot D}{\mu},$$

$$Re_S = \frac{G_{max} \cdot S}{\mu}, \quad Fs = \frac{1}{1 - F_t F_d}$$

$$A_{face} = Hi.L = N_t.X_a.L$$
$$A_{min} = N_t.L.[(X - D)(1 - F_t.F_d)]$$

$$X = X_a \quad \text{or} \quad X = \sqrt{\left(\frac{X_a}{2}\right)^2 + X_b^2},$$

According to which is less **[7]**.

When dew point temperature of entering air is less than wall temperature, no mass transfer occurs (Q_{lat} =0) and the heat energy will be transferred sensibly (dry surface evaporator), hence j_T and j_S are equal to 1.

Assuming staggered tube arrangement and hexagonal fin layout, the fin efficiency (η_f) can be calculated as **[8],**

$$\eta_f = \frac{\tanh(m \cdot r \cdot \varphi)}{m \cdot r \cdot \varphi} \qquad .....(8)$$

where,

$$m = \left[\left(\frac{2.h_{sen}}{k_f.F_t}\right)\left(1 + \frac{C.H_{fg}}{Cp}\right)\right]$$

$$C = \frac{C_1 + C_2}{2}$$

$$C_1 = \frac{W_w - W_{in}}{T_w - T_{di}},$$

$$C_2 = \frac{W_w - W_{out}}{T_w - T_{do}}$$

$$\varphi = \left(\frac{R}{r} - 1\right) \cdot \left(1 + 0.35 \cdot \ln\left(\frac{R}{r}\right)\right)$$

$$\frac{R}{r} = 1.27\, \psi\, (\beta - 0.3)^{1/2}$$

$$\psi = \frac{M}{r}, \qquad \beta = \frac{l}{M} \geq 1,$$

$$l = 0.5\left[\sqrt{\frac{X_a^2}{4} + X_b^2}\right]$$

M is defined as $X_a/2$ or X_b depending on which is less. For dry surface evaporator (no mass transfer), the parameter C equals to zero.

Performance of a plate fin and tube dehumidifying coil

To evaluate the performance of a plate fin and tube dehumidifying coil by using EGM method, inside tube and outside tube irreversibility should be included. The effective variables for in-tube side that must be included are inside tube diameter, tube circuiting arrangement and the type of fluid. This type of heat exchangers has a complex tube circuiting and no rules available to specify a general method for all tube circuiting arrangements, besides the designer has the ability to change the tube circuiting according to pressure drop limitations after the inside tube diameter and the type of fluid are specified. So the present work deals with the air side region only because of its higher resistance to heat flow than inside tube region and it contains most design variables.

Khan et al. [9] proposed an approach to model the tube bank in cross flow. This model was modified for the present analysis as follows,

For outside fluid (air) of the coil shown in Figure 1, conservation of mass leads to

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m} \quad (9)$$

For the steady state flow condition for the air outside tubes of the coil assuming no change in potential and kinetic energies and no work done during the process, the first law of thermodynamics will be reduced to,

$$Q = \dot{m}(H_o - H_i) \quad ....(10)$$

From the second law of thermodynamics of steady state flow condition for air outside tubes,

$$\dot{m}(s_o - s_i) \geq \frac{Q}{T_w} \quad \text{(Irreversible process)} \quad ....(11)$$

$$\dot{m}(s_o - s_i) = \frac{Q}{T_w} + \dot{S}_{gen} \quad ....(12)$$

$$\dot{S}_{gen} = \dot{m}(s_o - s_i) - \frac{Q}{T_w} \quad ....(13)$$

$$dh = T_a ds + \frac{dp}{\rho} \quad .....(14)$$

$$H_o - H_i = T_a(s_o - s_i) + \frac{p_o - p_i}{\rho} = T_a(s_o - s_i) - \frac{\Delta p}{\rho} \quad ....(15)$$

Substituting Equation 15 into Equation 10 and simplifying it, we get,

$$s_o - s_i = \frac{Q}{\dot{m}.T_a} + \frac{\Delta p}{\rho.T_a} \quad ..... (16)$$

Substituting Equation 16 into Equation 13, we get,

$$\dot{S}_{gen} = \frac{Q(T_w - T_a)}{T_w T_a} + \frac{\dot{m}.\Delta p}{\rho.T_a} \quad ....(17)$$

$$Q = -Q_T$$

$$\dot{S}_{gen} = \frac{Q_T(T_a - T_w)}{T_w T_a} + \frac{\dot{m}.\Delta p}{\rho.T_a} \quad ...(18)$$

Equation 18 consists of two parts, the first represents the entropy generation due to heat transfer and the second is the entropy generation comes from pressure drop. The pressure drop of the air side of a plate fin and tube dehumidifying coil can be calculated as follows neglecting the entrance and exit losses **[8]**,

$$\Delta p = \frac{G_{max}^2}{2\rho_{in}} \left[\left(\frac{\rho_i}{\rho_o} - 1\right)(\sigma^2 + 1) + f \frac{A_o}{A_{min}} \frac{\rho_i}{\rho_m} \right] ....(19)$$

where,

$$A_o = A_t + A_f$$

Friction factor can be calculated as **[10],**

$$f = f_f \frac{A_f}{A_o} + f_t\left(1 - \frac{A_f}{A_o}\right)(1 - F_t.F_d) \quad ....(20)$$

where,

$$f_f = 1.455.\ Re_D^{-0.656}\ (X_a/X_b)^{-0.347}\ (S_f/D)^{-0.134}\ (X_a/D)^{1.23}$$

$$S_f = (1/F_d) - F_t$$

$$f_t = \frac{\pi}{4}\left[0.25 + \frac{0.118}{\left[(X_a/D) - 1\right]^{1.08}} . Re_D\right] * \left[(X_a/D) - 1\right]$$

Substituting Equation 19 into Equation 18 and simplifying it, we get,

$$S^{\bullet}_{gen} = \left[\frac{Q_T(T_a - T_w)}{T_a T_w}\right] + \frac{m^{\bullet} . U^2}{2.T_a .\sigma^2} \left[\left(\frac{\rho_i}{\rho_o} - 1\right)\left(\sigma^2 + 1\right) + f\,\frac{A_o}{A_{min}}\frac{\rho_i}{\rho_m}\right] \quad .....(21)$$

The traditional efficiency or effectiveness (ε) of a plate fin and tube dehumidifying coil may be calculated as follows,

$$\varepsilon = \frac{\text{actual heat energy transferred}}{\text{maximum heat energy transferred}}$$

$$\varepsilon = \frac{m^{\bullet}(H_i - H_o)}{m^{\bullet}(H_i - H_w)}$$

$$\varepsilon = \frac{(H_i - H_o)}{(H_i - H_w)} \quad .....(22)$$

Calculations procedure

The present modeling and performance procedure of a dehumidifying coil were programmed using Visual Basic 6.0 language with the equations of thermodynamic and thermo-physical properties of moist air presented by Ashrae Fundamentals **[12]**. The design parameters of the present coil are listed in Table 1 which has been used as an evaporator in a window type air conditioner. Heat load of the coil was predicted by EVAP-COND **[4]** software under assumed conditions and Figure 2 shows the execution of the coil simulation. Figure 3 shows a flow chart of the present coil modeling under dry and wet conditions. At first the wall temperature is assumed and compared with dew point temperature of air. If the dew point temperature is less than wall temperature then the surface will be dry, out air temperature will be assumed and corrected until convergence occurs. Else the heat and mass will be transferred and the surface of the coil will be wet, moisture content and temperature of out air will be assumed and corrected until convergence occurs. The total heat load will be calculated and compared with the actual value. The assumed value of wall temperature is corrected using binary search method until convergence occurs between actual and calculated values of heat load, then the entropy generation rate and effectiveness of the coil are calculated using Equations 21 and 22 respectively.

Results

The present work provides a mathematical approach to evaluate the performance of a dehumidifying coil using entropy generation concept and compared it with traditional effectiveness of heat exchangers. Operating parameters such air velocity, air wet bulb temperature, air dry bulb temperature, air relative humidity and heat load are varied under specified ranges, ambient temperature (reference temperature) was fixed during calculations (T_a=35 C^o). Figure 4 compares the coil effectiveness (ε) and entropy generation rate ($\dot{S}_{gen}$) according to inlet air velocity (U), $\dot{S}_{gen}$ decreases with U and reaches minimum (optimum) at U=2.75, then starts to increase beyond this value but ε decreases with U and the optimum occurs at minimum value of U. The effect of the heat load (Q) is shown in Figure 5, when Q increases $\dot{S}_{gen}$ increases also because of increasing heat irreversibility part but ε doesn't be affected. Figure 6 shows the effect of the inlet air dry bulb temperature (T_{di}), when T_{di} increases at constant wet bulb temperature, $\dot{S}_{gen}$ and ε affects slightly, that may be due to fixed heat load capacity and unchanged pressure drop. Figure 7 represents the effect of air wet temperature (T_{wi}), when T_{wi} increases $\dot{S}_{gen}$ decreases due to both decreasing the sensible part and increasing the latent part of heat capacity but ε remains constant. It is clear from Figures 4, 5, 6 and 7 that $\dot{S}_{gen}$ has more sensitivity than ε for changing operating parameters that may be because of neglecting the effect of pressure drop in calculating ε and it depends on enthalpy change only, so $\dot{S}_{gen}$ is more practical than ε to describe the performance of the coil. Figures 8, 9, 10, 11 and 12 describe the performance of the present dehumidifying coil according to various operating parameters. As mentioned previously that maximum performance occurs at minimum entropy generation point, so we can trace and fix the optimum operating parameters for the coil to produce a higher performance. Increasing of heat load will decrease the performance and increase optimum air velocity (both undesirable) as shown in Figure 8 because of increasing the heat irreversibility part. The performance increases and optimum air velocity decreases (both desirable) with increasing wet bulb temperature (air closer to saturation) as shown in Figure 9 due to increasing latent part and decreasing the sensible part of heat load capacity. Increasing dry bulb temperature with constant wet bulb temperature affects the performance slightly as described in Figure 10 (minimum air velocity still constant). When the dry bulb temperature is being close to wet bulb temperature, the performance will be higher as shown in Figure 11. The effect of relative humidity and heat load capacity of air is presented in Figure 12, when relative humidity increases performance increases also with minimum heat load capacity.

Conclusions

The entropy generation algorithm can be used efficiently to describe the performance of the dehumidifying coil as an independent part of the system depending on the design and operating parameters without considering the other parts of the system such as compressor, condenser and expansion valve, it may be useful to evaluate the efficiency of an existing coil and to design a new efficient one because of including the effect of pressure drop and heat load capacity while the traditional efficiency or effectiveness depends on enthalpy change only. It's found that the combination of operating parameters of the coil such as air velocity, air wet bulb temperature, air dry bulb temperature, air relative humidity and heat load capacity affected the performance and we can conclude from the results that optimum air velocity increases with decreasing heat load capacity and with decreasing the sensible part of heat load. Most of air conditioning devices are designed and they are working at constant air velocity for all operating parameters conditions, but it is known that at the beginning of operation the space heat load is maximum and the space air is dry (higher sensible load), after a period of operation time the heat load decreases and the space air being more saturated, hence the speed of the entering air should be varied according to the state of air and according to the total space load to save power consumed because the optimum or economic air velocity will be changed according to operating parameters.

Nomenclature

A	Area	m^2
Cp	Specific heat	$J\ kg^{-1}K^{-1}$
D	Outer diameter of tube	m
D_v	Diffusion coefficient	$m^2 s^{-1}$
f	Friction factor	-
F_d	Fins density	fins m^{-1} or fins $inch^{-1}$
F_t	Fin thickness	m
G	Mass velocity	$kg\ m^{-2} s^{-1}$
Hi	Height of the heat exchanger	m
H	Enthalpy	$J\ kg^{-1}$
H_{fg}	Latent heat of evaporation	$J\ kg^{-1}$
h	Convective heat transfer coefficient	$W\ m^{-2} K^{-1}$
h_T	Total (dehumidifying) heat transfer coefficient.	$Kg\ m^{-2} s^{-1}$
j_n	Colburn j-factor for n rows	-
j_4	Colburn j-factor for 4 rows	-
k	Thermal conductivity.	$W\ m^{-1} K^{-1}$
L	Length of heat exchanger	m
$m^\bullet$	Mass flow rate	$kg\ s^{-1}$

N Number of tubes, rows -

p Pressure Pa

Pr Prandtl number $\left(\frac{\mu . Cp}{k}\right)$ -

Q Heat load W

r Tube radius m

R Equivalent radius m

Re_D Reynolds number based on D -

Re_b Reynolds number based on X_b -

Re_S Reynolds number based on S -

S Fins spacing m

s Entropy J kg $^{-1}$

Sc Schmidt number $\left(\frac{\mu}{\rho . D_v}\right)$ -

$\dot{S}_{gen}$ Entropy generation rate W K $^{-1}$

T Temperature K

U Air face velocity m s $^{-1}$

W moisture content kg_{water} / kg_{air}

X_a Transverse tube spacing m

X_b Longitudinal tube spacing m

Greek symbols

η Efficiency -

μ Dynamic viscosity pa.s

Δ Difference -

ρ Density kg m $^{-3}$

ε Effectiveness %

Φ Relative humidity %

Subscripts

a Ambient

b based on X_b

D based on tube diameter

di dry bulb in

do dry bulb out

dp dew point

f fins

face face of the heat exchanger

i input of heat exchanger

lat latent

m mean

min minimum flow area

max based on minimum flow area

o output of heat exchanger

r rows

sen sensible

t tubes

w wall

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Table (1) Details of the base design heat exchanger

D (mm)	X_a (mm)	X_b (mm)	L (mm)	F_d (fins/inch)	N_t	N_r	F_t (mm)	U (m/s)	Q (w)	T_{di} (C^o)	Φ_i %
9.53	25.4	19.06	365	12	16	3	0.114	2.5*	5990**	24	50

*Ashrae Systems and Equipment [12] recommended that air face velocity of dehumidifying coils should be between (2-2.5) m/s.
**Found from Evap-Cond software at assumed condition (see Fig. 2).

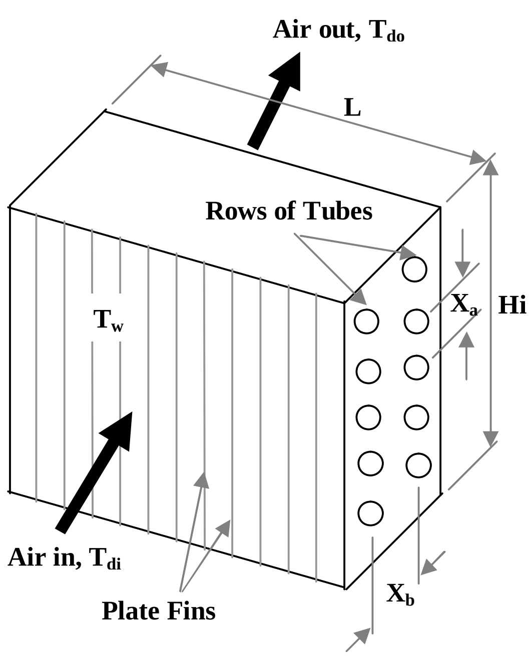

Figure (1) Plate fin and tube heat exchanger

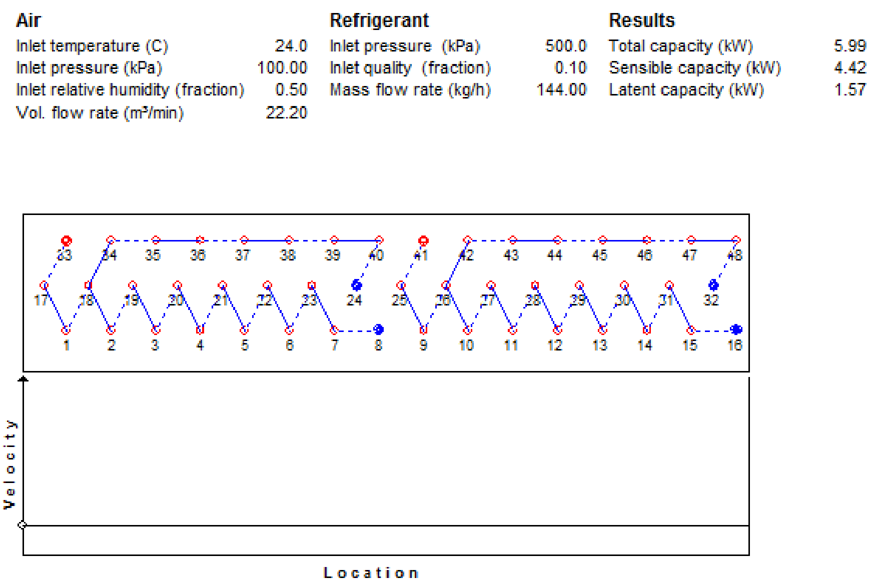

Figure (2) The simulation of the present coil using Evap-Cond software.

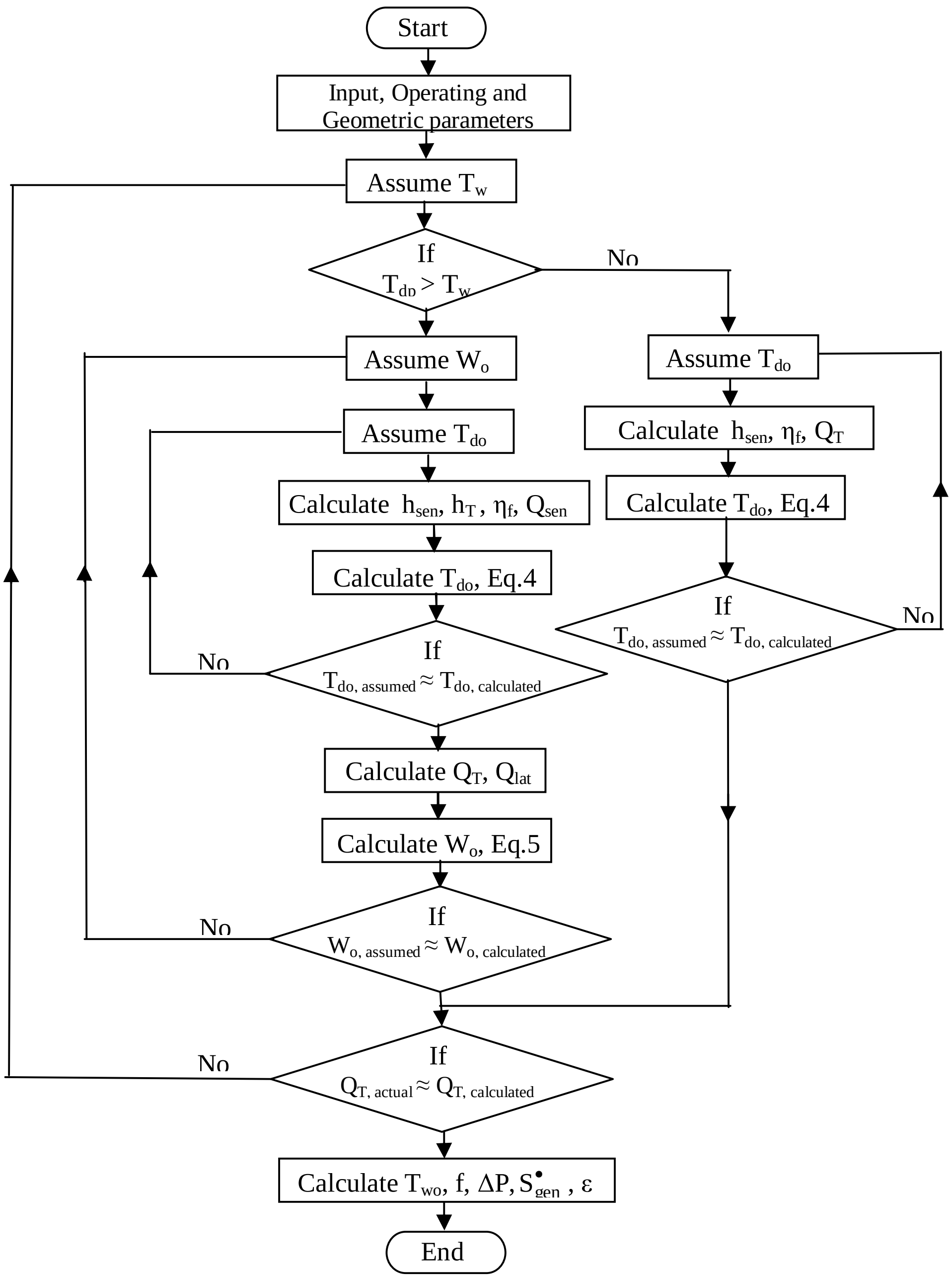

Figure (3) The flow chart of the coil modeling

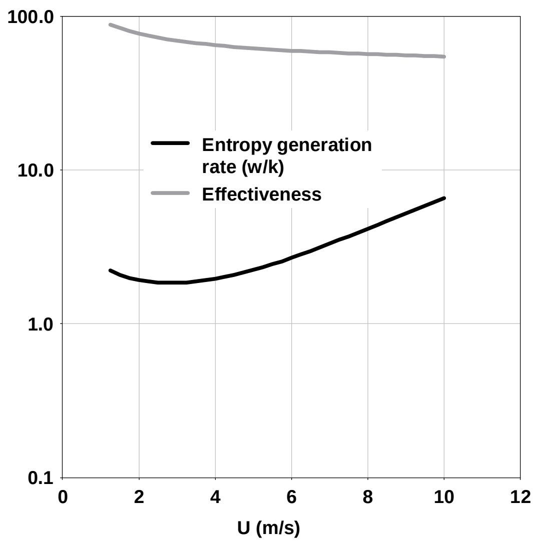

Figure 4. Entropy generation rate and effectiveness according to air velocity

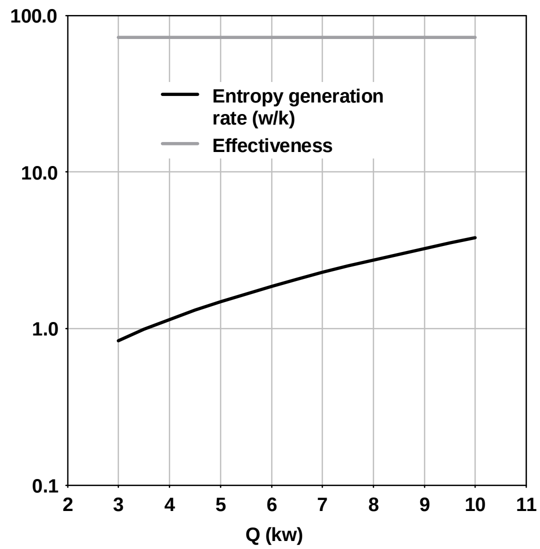

Figure 5. Entropy generation rate and effectiveness according to heat load

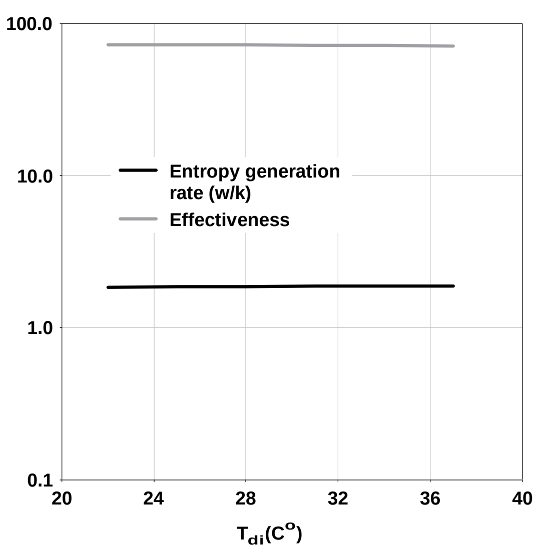

Figure 6. Entropy generation rate and Effectiveness According to dry bulb temperature

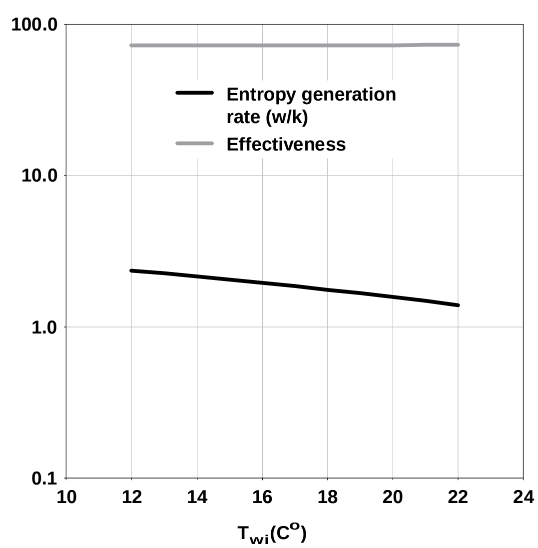

Figure 7. Entropy generation rate and effectiveness according to wet bulb temperature

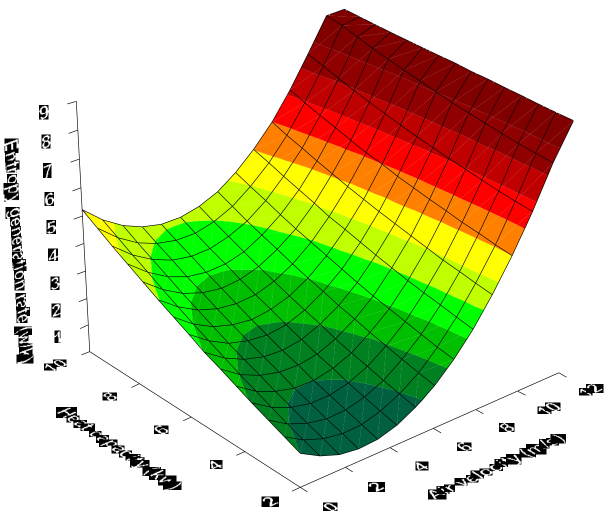

Figure (8) The relation between U, $\dot{S}_{gen}$ and Q

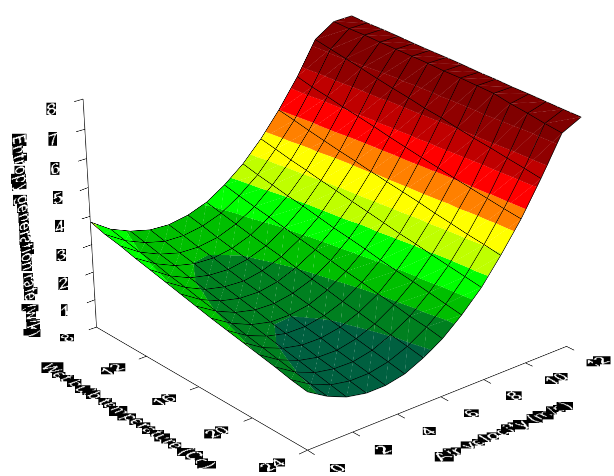

Figure (9) The relation between U, $\dot{S}_{gen}$ and T_{wi}

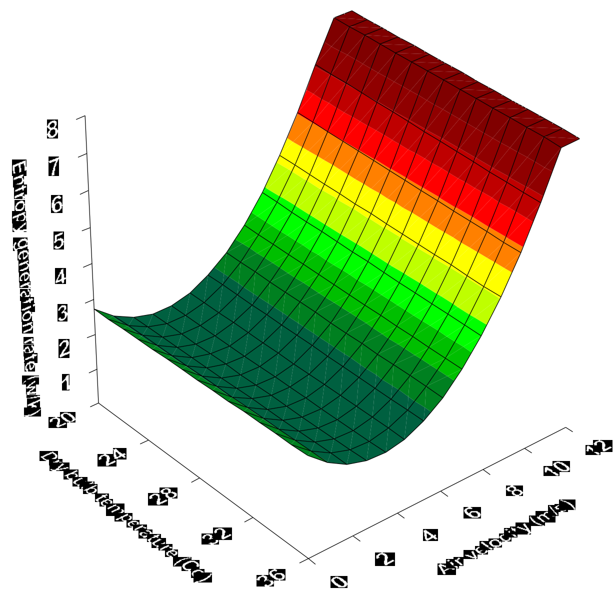

Figure (10) The relation between U, $\dot{S}_{gen}$ and T_{di}

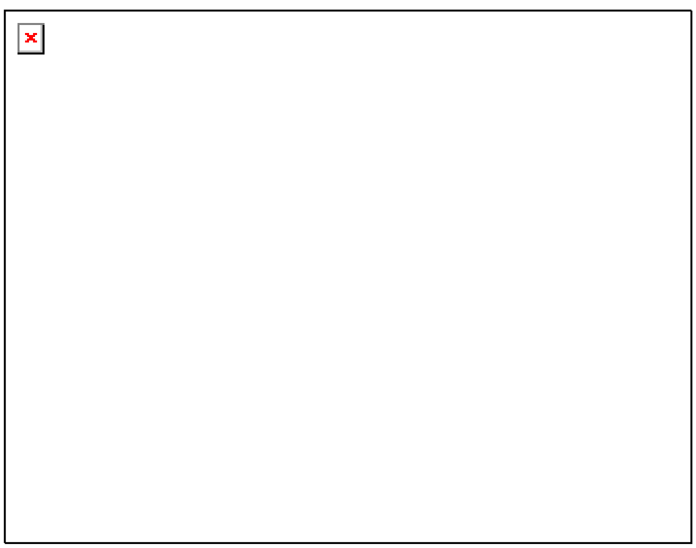

Figure (11) The relation between T_{wi} , $\dot{S}_{gen}$ and T_{di}

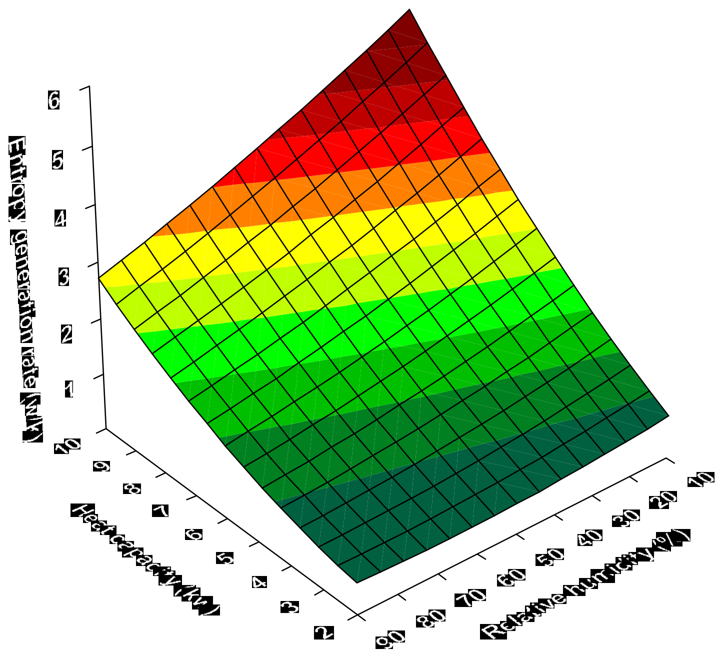

Figure (12) The relation between Φ, $\dot{S}_{gen}$ and T_{di}