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Heat transfer performance analysis of dehumidifying fin and tube heat exchangers

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ABSTRACT

In that study, effects of dehumidifying of air on heat transfer performance of fin and tube heat exchanger under different operating conditions are investigated. Detailed literature survey is done and appropriate correlations are discussed for fully wet conditions in heat transfer process. For thermal analysis, ε-NTU method is used. Due to the dehumidifying process, temperature potential cannot be used and enthalpy potential must be introduced. For defining heat capacity rate, fictitious enthalpy is preferred for most of the researchers. However, it is not clear that at which temperature, this enthalpy should be calculated. At that point, different cases are formed and most accurate approach is determined. Since experimental data in a wide range is not accessible, a reliable and certificated simulation program, "Coils for Windows-Luvata", is used to compare obtained results. It is shown that estimation of heat capacity rate of the air is the dominant factor in determining heat transfer capacity of the exchanger under dehumidifying conditions. Using mean temperature of inlet and exit of cooling water to define heat capacity rate of the air gives the best results. Most accurate results are obtained from Wang et. al. (2000) heat transfer coefficient correlation.

1. Introduction

Fin and tube heat exchangers are used extensively in refrigeration and air conditioning applications. Generally, air flows between fins and different type of fluids such as water, R22, R134a etc. circulated in tubes. Based on application, air can be heated or cooled. For cooling process of air, if the surface temperature of the coil is less than the dew point temperature of the air, water vapor condensed and covers the some part of the fin and tubes. For dry cooling process, no moisture will condense out of the air stream and heat transfer from air to the cold fluid is only in sensible form. For wet cooling process, latent heat must be also taken into account. As mentioned before, condensed water change the flow pattern of air and film thickness increased the thermal resistance. For that reasons, thermal analysis of dehumidifying fin and tube heat exchangers are more complicated than the dry cooling process. For example, ε-NTU method is not iterative for dry cooling heat transfer process. By using inlet conditions of streams, exit conditions and heat capacity of the coil can be determined. Based on the accuracy of the correlation used in analysis, accurate results can be obtained. However, ε-NTU method is iterative for wet cooling heat transfer process. Since latent and sensible heat transfer occurs simultaneously, temperature potential approach cannot be applied and enthalpy potential approach is used. Adding to that, for practical engineering applications, this method and other similar methods like LMTD or LMED, heat transfer analysis with mass transfer due to condensation of vapor can be done in a macroscopic view. For that reason distinction between latent and sensible heat cannot be shown clearly. To solve this problem, fictitious air enthalpy is defined. The fictitious air enthalpy is the saturated air enthalpy evaluated at a certain temperature.

In that study, different cases for fictitious enthalpy is formed and heat capacity rate of the heat exchanger calculated. These cases include different temperatures for fictitious air enthalpy. Average of cooling water inlet and exit temperatures, average of air inlet and exit temperatures, average of cooling water and air inlet temperatures are used to calculate fictitious enthalpy by using the curve of air saturated enthalpy in psychometric chart. Secondly, heat transfer coefficient correlations for air side analyzed and deviations of these correlations discussed in detail. All thermal calculations are done for different geometries under the same operating conditions. Since experimental data in a wide application range is not available, a simulation program, Coils for Windows, produced by Luvata Söderköping AB Company is used.

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Nomenclature

A	: Area (m^2)	NTU	: number of transfer unit
A_1	: Primary Area (m^2)	Q	: Heat capacity (W)
A_2	: Secondary area (m^2)	Pr	: Prandtl number
$A_{T,air}$	: Total surface area (m^2)	Re	: Reynolds number
A_o	: Minimum air flow area (m^2)	Re_D	: Reynolds number defined based on tube outside diameter
b'	: slope of the saturated air enthalpy curve	Re_o	: Reynolds number defined based on minimum air flow area
C	: Heat capacity rate (W/K)	UA	: Overall heat transfer coefficient (W/m^2K)
c_p	: Specific heat constant (J/kgK)	T	: Temperature (ºC)
d_h	: Hydraulic diameter (mm)	V_o	: maximum air velocity (m/s)
d_i	: Inside tube diameter (mm)	y	: film thickness
d_o	: Outside tube diameter (mm)	Z	: Depth of the heat exchanger (mm)
h	: heat transfer coefficient (W/m^2K)	ΔL	: fin spacing (mm)
H	: height of the heat exchanger (mm)	ΔH	: tube spacing in transverse the direction (mm)
i	: enthalpy (kj/kg)	ΔZ	: tube spacing in the longitudinal direction (mm)
k	: heat conduction coefficient (W/mK)	δ	: fin thickness (mm)
L	: width of the heat exchanger (mm)	ε	: efficiency
l'	: Equivalent fin height (mm)	ρ	: density (kg/m^3)
$\dot{m}$	: Mass flow rate (kg/s)	η_f	: fin efficiency
N	: number of rows		
N_f	: number of fins		
N_k	: number of columns		

Subscripts

a	: air	e	:exit
dh	: dehumidifying	sat	: saturated
i	:inlet	w	:water

2. Literature Survey

Jacobi et. al. (2001) summarized experimental and theoretical studies about air-side heat transfer and pressure-drop performance of serpentine-fin, flat-tube heat exchangers. This research provides a careful assessment of the serpentine-fin, flat-tube geometry for HVAC&R applications. Seshimo et. al. (1989) measured thermal performance of single-row plate-fin-tube heat exchanger experimentally. It was found that heat transfer increased 20% under wet condition when compared with dry condition. The main reason of this enhancement was defined as the changing of the fin geometry due to condensed vapor. Condensed vapor also change the air flow passage and that cause an increase in pressure drop. McQuiston (1975) researched fin efficiency under wet condition. Based on heat and mass transfer analysis, it is found that fin efficiency decreases by 7~8% under dehumidifying process. Abu Madi et. al. (1998) investigated heat transfer performance of round tube and plate finned heat exchangers under dry condition. It was found fin thickness and number of tube rows has little effects on friction. For smaller fin thicknesses, higher heat transfer rate were obtained as expected. Rich (1975) investigated the effect of the number of tube rows on heat transfer performance of smooth plate fin-and-tube heat exchangers. At high Reynolds number, higher heat transfer rates were obtained as number of tube rows increased. However, at low Reynolds number, heat transfer rate decreased. Pacheco-Vega et. al. (2001) discussed heat transfer coefficient correlations defined for air side of the heat exchanger in detail. They analyzed McQuiston (1978) correlation. McQuiston (1978) presented experimental data for five plate fin-and-tube heat exchangers, and developed a heat transfer coefficient correlation for both dry (Equation 1) and wet (Equation 2) surfaces.

$$j = 0.0014 + 0.2618Re_D^{-0.4}\left(\frac{A_{T,air}}{A_1}\right)^{-0.15} \quad (1)$$

$$j = 0.0014 + 0.2618Re_D^{-0.4}\left(\frac{A_{T,air}}{A_1}\right)^{-0.15}(0.84 + 4x10^{-5}Re_o^{1.25}) \quad (2)$$

Wang et. al. (2000) developed an air side correlation for plain fin-and-tube heat exchangers in wet conditions by using a total of 31 samples with different geometries. For the Colburn j factor is given as follows ($300<Re_D<5000$):

$$j = 19.36Re_D^{j1}\left(\frac{\Delta L}{D}\right)^{1.352}\left(\frac{\Delta H}{\Delta Z}\right)^{0.6795}N^{-1.291} \quad (3)$$

$$j_1 = 0.3745 - 1.554\left(\frac{\Delta L}{D}\right)^{0.24}\left(\frac{\Delta H}{\Delta Z}\right)^{0.12}N^{-0.19} \quad (4)$$

It was stated that the proposed j factor gives a mean deviation of 6.33% when compared with experimental data. Theerakulpisut and Priprem (1998) used correlation developed by Myers (1967) which is a ratio of j factor defined for wet coil and dry coil. The value of j factor for dry coil was calculated by the relation proposed by McQuiston (1978)

$$j_4 = (1.0673V_o^{0.101})\left[0.0014 + 0.2618Re_D^{-0.4}\left(\frac{A_{T,air}}{A_1}\right)^{-0.15}\right] \quad (5)$$

In literature, there are many experimental studies and correlations. In this paper, most widely used correlations are preferred.

3. Theory

In that study, plate fin and tube heat exchanger with in-line tube arrangement is analyzed. Schematic representation of the exchanger is given in Figure 1.

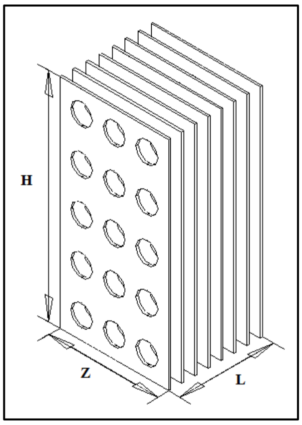

Figure 1. Schematic representation of heat exchanger

Four different heat exchangers are used in thermal analysis. Total heat transfer surface area and dimensions of the exchangers are given in Table 1.

Table 1. Dimensions of the heat exchangers

	Coil 1	Coil 2	Coil 3	Coil 4
$A_{t,air}$ (m^2)	29	37	64	105
L (mm)	1440	970	1880	2320
Z (mm)	220	280	280	220
H (mm)	370	470	600	836
ΔL (mm)	4	4	6	4
d_i (mm)	12.7	12.7	12.7	12.7
t (mm)	0.35	0.35	0.35	0.35

3.1. Heat Transfer Area of the Heat Exchanger

The first step of the thermal analysis is the determination of the heat transfer area of the heat exchanger both for cooling water and humid air. Heat transfer area of the air consists of two main parts. First one is the surface of the tube and named as primary area and given in Equation 6.

$$A_1 = \left(\pi d_o - \pi d_o \delta \frac{1}{\Delta L}\right) N_k N L \quad (6)$$

Secondary area is given in Equation 7 and defined as the fin surface area. Total surface area that air contacts is given in Equation 8.

$$A_2 = \left(HZ - \left(\frac{\pi d_o^2}{4} N_k N\right)\right)\left(\frac{2L}{\Delta L}\right) \quad (7)$$

$$A_{T,air} = A_1 + A_2 \quad (8)$$

Beside the total heat transfer area, cross sectional flow area of the air is important to determine the hydraulic diameter and Reynolds number and given in Equation 9.

$$A_o = HL - \delta H N_f - d_o L N \tag{9}$$

Generally, inlet conditions of heat exchangers are known and it is desired to calculate the heat transfer capacity and outlet conditions of the heat exchanger. To solve this problem ε-NTU method is used.

3.2. Heat Capacity Rate

As mentioned before, ε-NTU method is not iterative for dry cooling process of air, heat capacity of the air and water can be easily determined by using Equation 10 and Equation 11.

$$C_w = \dot{m}_w c_{p,w} \tag{10}$$
$$C_a = \dot{m}_a c_{p,a} \tag{11}$$

Since all heat transfer is in sensible form and temperature potential can be used, dry cooling process heat capacity rates can be determined easily. However, in dehumidifying process, enthalpy potential of air is used and heat capacity rate must be determined based on this approach. In that study, the slope of the saturated air curve in psychometric chart at water inlet temperature is used to define the heat capacity rate of the air. Different type of definitions are available for this approach, for simplicity of the calculations since water inlet temperature is known, heat capacity rate of the air is determined by using Equation 12.

$$C_a = \dot{m}_a b' \tag{12}$$
$$b' = \left[\frac{di_{sat}}{dT}\right] \tag{13}$$

The most important point in determination of the b’ in Equation 13 is at which temperature it is going to be evaluated. Theerakulpisut and Priprem (1998) used a polynomial for the saturation enthalpy of air as a function of dry bulb temperature.

$$i_{sat,a} = 9.3839 + 1.71137T + 0.0222T^2 + 0.00063T^3 \tag{14}$$

The slope of this polynomial at any given dry bulb temperature is given in Equation 15.

$$\frac{di_{sat,a}}{dT} = 1.71137 + 0.0444T + 0.00189T^2 \tag{15}$$

By using Equation 15, b’ is determined at mean water temperature at inlet and exit. For dry cooling process, total heat capacity of the heat exchanger can be determined by using Equation 16.

$$\dot{Q} = \varepsilon C_{min}\left(T_{w,i} - T_{a,i}\right) \tag{16}$$

For dehumidifying process, total heat capacity is summation of latent and sensible heat of the air and calculated by using Equation 17. Dehumidifying process is given in Figure 2.

$$\dot{Q} = \dot{m}_a (i_i - i_e) \tag{17}$$

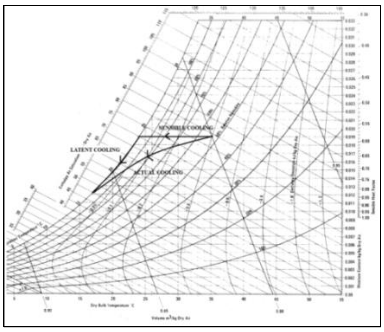

Figure 2. Dehumidification process on psychometric chart

To complete the thermal analysis of heat exchanger overall heat transfer coefficient must be determined. This coefficient is depending on air and water sides heat transfer coefficient. For dry cooling process, overall heat transfer coefficient can be determined by using Equation 18.

$$\frac{1}{UA}=\frac{1}{h_w A_w}+\frac{ln\left({}^{d_o}/{}_{d_i}\right)}{2\pi k L N_k N}+\frac{1}{h_{a,dry}(\eta_f A_2+A_1)} \quad (18)$$

For dehumidifying process, generally, air side heat transfer coefficient is modified based on saturated air enthalpy curve and calculated by using Equation 19.

$$h_{a,wet}=\frac{1}{\frac{c_{p,a}}{b' h_{a,dry}}+\frac{y_{dh}}{k_{dh}}} \quad (19)$$

This modified heat transfer coefficient is replaced by dry cooling process heat transfer coefficient in Equation 18. Modified heat transfer coefficient includes the thermal resistance due to condensed liquid film. In most practical engineering application, it is difficult to calculate film thickness. Threlkeld (1972) has demonstrated that thermal resistance of film can be ignored. For practical applications, film thickness can be assumed as 0.1016 mm. As mentioned before heat capacity rate of dehumidifying air is determined by using b'. The next step is the determination of the fin efficiency. For dry cooling process fin efficiency can be calculated by using Equation 20 and Equation 21.

$$\eta_f=\frac{tanh(ml')}{ml'} \quad (20)$$

$$m=\sqrt{\frac{2h_{a,dry}}{k_{fin}\Delta L}} \quad (21)$$

For wet fin efficiency, the same equations are used but heat transfer coefficient is replaced by the one obtained in Equation 19. For water side heat transfer coefficient, widely used Gnielenski (1976) correlation is preferred.

3.3. ε-NTU Relations

The effectiveness of the heat exchanger is defined as the ratio of the actual amount of heat transferred to the maximum possible amount of heat transferred. In general it is function of the number of transfer units NTU, the heat capacity rate ratio C_r , and the flow arrangement of the hot and cold fluids. In that study, a counter flow heat exchanger is considered and ε -NTU relationship is given in Equation 22.

$$\varepsilon=\frac{1-exp\left(-NTU(1-C_r)\right)}{1-C_r exp\left(-NTU(1-C_r)\right)} \quad (22)$$

Number of transfer unit is determined by using minimum heat capacity rate and overall heat transfer coefficient.

$$NTU=\frac{UA}{C_{min}} \quad (23)$$

C_r is the ratio of these heat capacities and defined as the ratio of minimum heat capacity rate to maximum heat capacity rate.

$$C_r=\frac{C_{min}}{C_{max}} \quad (24)$$

4. Results

In that study, heat transfer performances of four fin and tube heat exchangers with different heat transfer surface area are investigated under fully wet dehumidifying conditions. 40 different operating conditions introduced to the reliable coil design program, 'Coils for Windows, Version 9.1.0.0 - Luvata'. Operating conditions are selected similar for each heat exchanger. Obtained results are compared with the results of the theoretical model. Three different cases are introduced to determine the fictitious enthalpy and heat capacity rate of the air. The difference between the cases is the temperature used to estimate fictitious enthalpy. Deviation of these cases analyzed. Finally, heat transfer coefficient correlations are investigated. Obtained results from McQuiston (1978), Wang et. al. (2000) and Myers (1967) heat transfer correlations are compared with each other and effects of these correlations on estimating heat transfer capacity of the heat exchanger discussed.

1. Effects of Temperature on Fictitious Enthalpy and Heat Capacity Rate

As mentioned before, derivative of fictitious enthalpy defined at a certain temperature is used for dehumidifying process of air to define heat capacity rate. Three different cases are formed and for each case total heat transfer capacity of the heat exchanger determined. For each case, heat capacity rate of air is defined for different temperatures like average of water inlet and exit temperatures, average of air inlet and exit temperatures, etc. Details of these cases are given in Table 2.

Table 2. Temperatures defined for fictitious enthalpy

Case 1	$C_a = \dot{m}_a \left[\frac{\mathrm{di_{sat}}}{\mathrm{dT}}\right]_{\left(\frac{T_{w,i}+T_{w,o}}{2}\right)}$
Case 2	$C_a = \dot{m}_a \left[\frac{\mathrm{di_{sat}}}{\mathrm{dT}}\right]_{\left(\frac{T_{w,i}+T_{h,i}}{2}\right)}$
Case 3	$C_a = \dot{m}_a \left[\frac{\mathrm{di_{sat}}}{\mathrm{dT}}\right]_{\left(\frac{T_{a,i}+T_{a,o}}{2}\right)}$

When these average temperatures are plotted on psychometric chart, it is seen that minimum value of these temperatures is for average of the water inlet and exit temperatures (Case 1) and the maximum value is the average of air inlet and exit temperature (Case 2). Based on this approach, comparison of the calculated heat transfer capacity with simulation results for model 1 and 2 are given in Figure 3 and Figure 4, respectively.

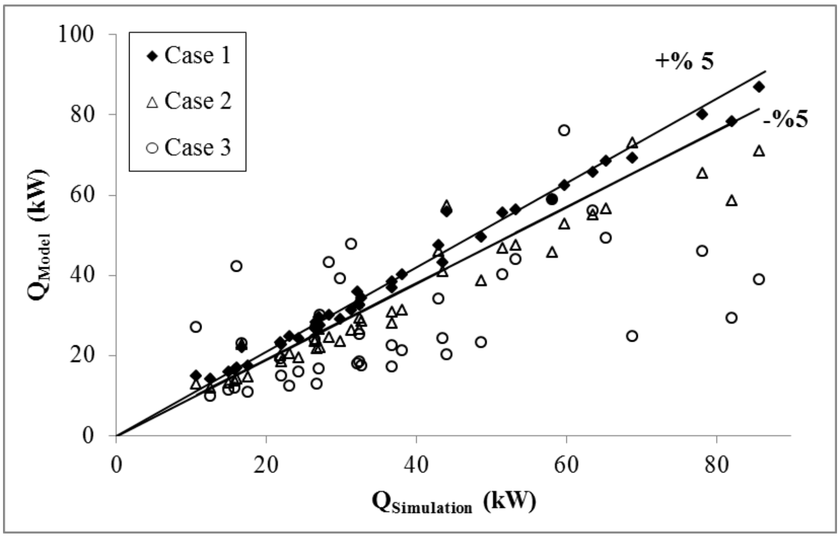

Figure 3. Heat capacities obtained from model 1 for different cases

In Figure 3 most of the results obtained from the model using case 1 are in the range of ±%5 deviations. To better clarify the comparison of the results obtained from each case, absolute mean deviation of theoretical models results from simulation results are given in Table 3.

Table 3. Comparison of cases

	Absolute Mean Deviation (%)
Case 1	5.8
Case 2	15.4
Case 3	24.4

It is seen that most accurate results are obtained from case 1 with an absolute mean deviation of 5.8 % where average of water inlet and exit temperatures is used. When different temperatures are used for heat capacity rates, deviations increased dramatically. It is also observed that mean temperature used for defining fictitious enthalpy and heat capacity rate is minimum at case 1 and maximum at case 3. As this mean temperature decreases, deviation of the models also decreases.

As mentioned before, heat capacity rate of the exchanger is defined based on effectiveness. Effectiveness is function of heat capacity rate and overall heat transfer coefficient. Using different temperatures on saturation enthalpy curve of air changes the heat capacity rate of the air and that is the main reason of the increase in deviation.

2. Effects of ε-NTU Relation on Heat Transfer

Effectiveness of a heat exchanger is defined for a given heat exchanger of any flow arrangement as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate. Since heat transfer process includes latent and sensible heats, maximum possible heat transfer is calculated by using heat capacity rates. For that reason, ε-NTU relation must be analyzed in detail. In figure 4, deviation of cases heat transfer capacity results from simulation results are given.

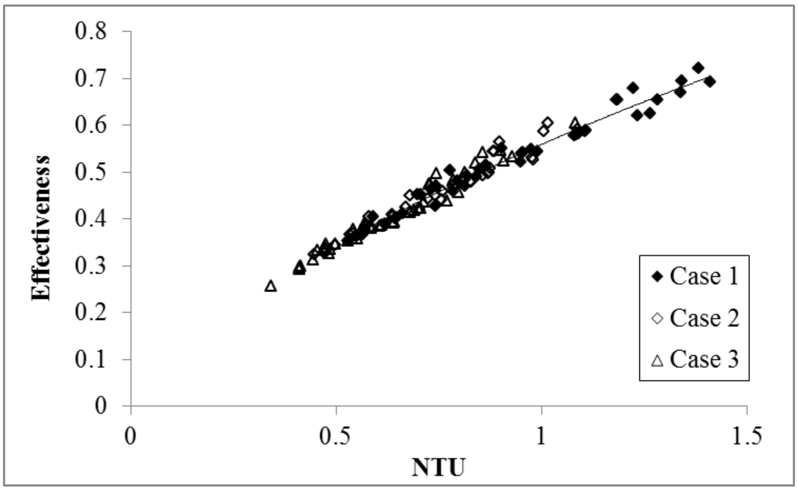

Figure 4. ε-NTU relations for model 1

Effectiveness of the heat exchanger is calculated similar for each cases and number of transfer unit calculated in a wide range for case 1. It can be clearly stated that estimating heat capacity rate of the air under dehumidifying conditions is more effective than the ε-NTU relation.

Beside that condensate rate of the vapor in humid air is calculated for each cases. In Figure 5, comparison of condensation rate calculated by model with coil design program is given. Similar to heat capacity rates, most accurate result is obtained from Case 1.

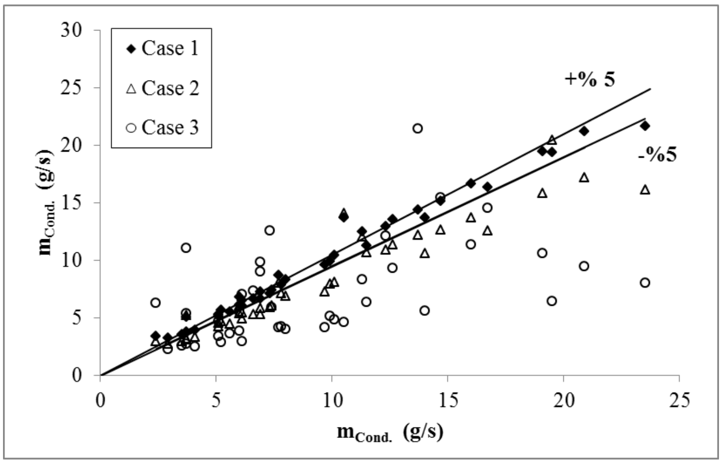

Figure 5. Comparison of condensate rate obtained from the model with simulation results

Condensation rate depends on exit condition of the air. Model calculates the dry bulb temperature and relative humidity at the exit of the heat exchanger. By using these two data, moisture content is determined and condensation rate is calculated.

3. Effects of Heat Transfer Coefficient Correlations

Another important point that must be emphasized is the effect of air side heat coefficient on heat transfer performance of heat exchanger. In that study, McQuiston (1978), Wang et. al. (2000), Myers (1967) heat transfer correlations are discussed. For all calculations, fictitious air enthalpy and heat capacity rate of air are determined by using average temperature of the inlet and outlet of the cooling water as given before in case 1. Comparison of air side heat transfer coefficients for different air side Reynolds number is in figure 8. For low Reynolds numbers ($Re_{a,D}$<1500), McQuistion (1978) correlations for dry and wet conditions and Myers (1967) correlation gave same results. For this range Wang et. al. (2000) correlation estimate heat transfer coefficient 30 % higher than the other correlations.

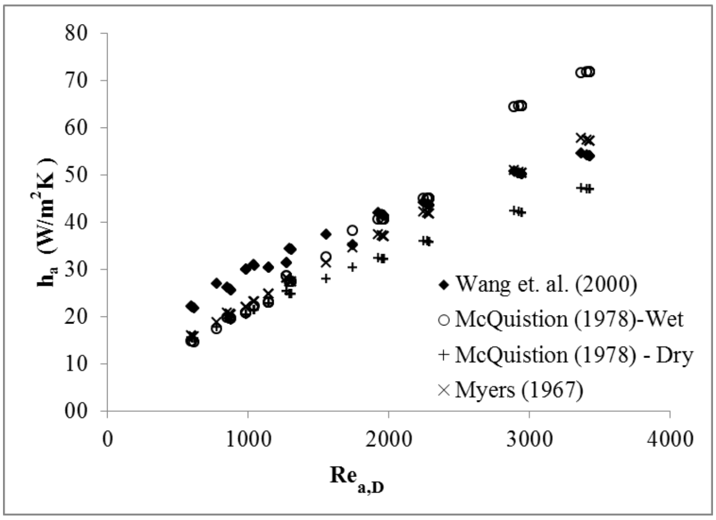

Figure 6. Comparison of heat transfer coefficients obtained from correlations

As the air side Reynolds number increased, trend of the obtained results from each correlation differ. Firstly, McQuistion (1978) correlation for wet condition estimates the highest heat transfer coefficient. The lowest heat transfer coefficients are obtained from McQuistion (1978) correlation for dry condition as expected. When compared with wet condition heat transfer coefficient correlations, that correlation estimates heat transfer coefficient 20 % lower than Wang et. al. (2000), Myers (1967) and 40 % lower than McQuistion (1978) correlation.

According to those establishing, McQuiston (1978) and Wang et. al. (2000) correlations are used for calculating heat transfer capacity of heat exchanger and deviation from simulation results is given in figure 9. Heat capacity rate of the air is calculated as in case 1.

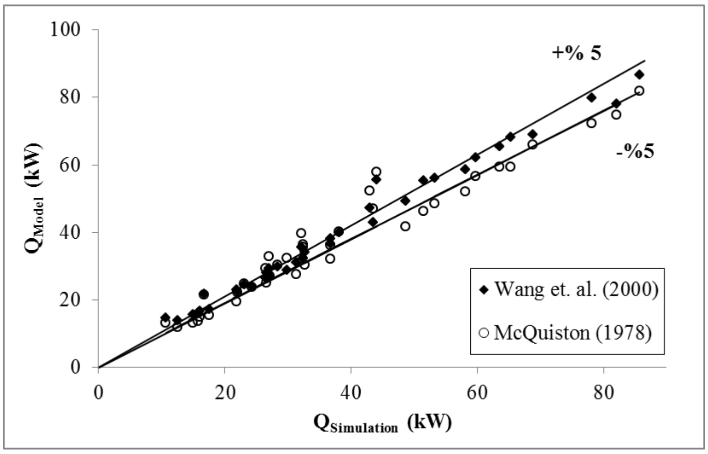

Figure 7. Comparison of the heat capacity rates based on Wang et. al. (2000) and McQuiston (1978) correlations

There is an evident difference between heat transfer coefficient correlations. However, that difference doesn't directly affects the calculated heat capacity results. Heat transfer capacity of the heat exchanger is calculated with 5.8 % absolute mean deviation by using Wang et. al. (2000) correlation and 10.4 % absolute mean deviation by using McQuiston (1978) correlation. Theoretical model using Wang et. al. (2000) correlation is very successful to estimate the simulation results.

Conclusion

In that study, heat transfer performance of fin and tube heat exchangers under dehumidifying conditions is investigated. Heat exchangers with different heat transfer surface area are used in analysis. Three different cases are formed to define fictitious enthalpy and heat capacity rate. For each case, total heat transfer capacity of the heat exchanger determined. Best results are obtained for case 1 where fictitious enthalpy and heat capacity rate are defined by using average of water inlet and exit temperatures. When different temperatures are used for heat capacity rates, deviations increased dramatically. As the temperature used for determining air side heat capacity rate decreases, deviation of the model also decreases.

Effect of air side heat coefficient on heat transfer performance of heat exchanger is also discussed in that study. McQuiston (1978), Wang et. al. (2000), Myers (1967) heat transfer correlations are discussed in detail. At low Reynolds numbers, Wang et. al. (2000) correlation has the highest heat transfer coefficients. For high Reynolds numbers, Wang et. al. (2000) and Myers (1967) correlations have similar values. Highest heat transfer coefficient values are obtained from McQuiston (1978) correlation.

Finally, heat transfer capacities of the heat exchangers are calculated by using McQuiston (1978) and Wang et. al. (2000) correlations. By using first correlation, results deviates 10.8 %. For the second correlation, this deviation is 5.8 %.

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