Experimental Study of Air-Side Performance of A Flat Fin-and-Tube Heat Exchanger Under Dry and Wet Conditions

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ABSTRACT

In this study, experiments were conducted on a fin-and-tube heat exchanger under both wet and dry conditions. Throughout the experiments, the air inlet temperature to the heat exchanger remained constant. Tests were carried out at relative humidities of 30%, 70%, 80%, and 90% for both dry and wet conditions. Additionally, experiments were conducted at five different air velocities for each relative humidity. Since the same heat exchanger was used for all test conditions, the geometric parameters of the heat exchanger remained unchanged during the experiments. The aim of this study is to investigate the performance of a finned-tube heat exchanger under different tube arrangements in both dry and wet conditions, a topic that has not been extensively covered in the literature. The experimental data were analyzed theoretically, first comparing the heat transfer coefficient and then the Colburn j-factor. These analyses were compared with some commonly used correlations in the literature. Based on the results obtained from the experiments, a comparison was made regarding the air-side performance of the fin-and-tube heat exchanger. The pressure drop values on the air side under wet conditions showed similar results at the same Reynolds number, despite varying relative humidities. There was a difference of 3% to 10% in the Reynolds number under wet conditions. When comparing the pressure drop values on the air-side between dry and wet conditions at the same Reynolds number, there was an average difference of 71%. The Colburn j-factor and heat transfer coefficient results are consistent with each other. The heat transfer coefficient values under dry and wet conditions were approximately 2.5 times higher at low relative humidities, increasing up to 3.1 times at 90% relative humidity.

1. INTRODUCTION

Fin-and-tube heat exchangers are affected by numerous parameters that influence heat transfer performance. Tube arrangement, fin geometry and tube thickness, tube diameter, and tube material are among the most significant factors. The inlet conditions on the air-side of the heat exchanger also play a crucial role in heat transfer performance. There are many studies in the literature examining the effect of flow inside the tubes on heat transfer and investigating the impact of air-side conditions. As a result of these studies, there is a greater consensus in the literature regarding the influence of flow inside the tubes on heat transfer compared to the effect of air-side conditions. Research on the air side is still ongoing. Initial studies on flat fin geometry were conducted who examined dry conditions in a 31.75x27.5 mm geometry by Albert (1957). F. McQuiston (1980) investigated a total of 14 different heat exchangers with the same tube diameter, analyzing the effects of row and tube counts and concluding that heat transfer coefficient is independent of fin spacing, on the other hand, conducted studies on various fin spacing 25.4x22 mm geometry with D_0=9.96 mm and N=4. He claimed that his proposed correlation had an accuracy of 35%. *Wang et al.*(2000) proposed correlations for both dry and wet conditions for a flat fin model. They also conducted studies on different fin models and materials. *Wang et al.* (2002) This study compared experimental studies of a 4-row fin-and-tube heat exchanger under dry and wet conditions at different relative humidity and different air velocities with the correlations in the literature.

2. THEORITICAL ANALYSIS OF FIN-AND-TUBE HEAT EXCHANGER

Heat capacity calculations in heat exchangers are based on three important parameters: temperature difference, heat exchanger surface area and overall heat transfer coefficient values. The overall heat transfer coefficient, denoted by U_0, is a based on logarithmic temperature difference and represents one of the most significant parameters of heat transfer performance in heat exchangers. The overall heat transfer coefficient is calculated by determining the thermal resistances. In addition to thermal resistances, contact resistance and fouling resistance are also included in the calculation of the total heat transfer coefficient. The formation of very small gaps at the contact points creates a resistance, which is referred to as contact resistance. Contact with the external environment or the gradual roughening of the surface over time results in the accumulation of pollution, which in turn creates a resistance to heat transfer between the medium and the surface. This phenomenon is known as fouling resistance.

2.1 Overall Heat Transfer Coefficient for Wet Conditions

In the wet condition tests carried out in this study, it is assumed that the fin surface of the heat exchanger is completely wet. This means that even the highest temperature in the fin-and-tube heat exchanger is lower than the dew point temperature of the air. In this case, condensation occurs on the finned surface of the heat exchanger. In case of condensation, heat transfer and mass transfer occur simultaneously. There are two ways in which condensation can occur: The first one is liquid film condensation and second is droplet condensation on the surface. In droplet condensation, the condensed water vapour particles remain as droplets, rather than forming a film layer on the surface. These water droplets that later merge can show a flow like a film layer. The heat transfer equation written for wet conditions given below as Equation (1) was originally proposed by Threlkeld (1970) for simplifying the calculation of the total heat transfer rate in a wet cooling coil. The overall heat transfer coefficient is based upon the enthalpy difference, where $U_{0,w}$ is written instead of U_0.Unlike the equation written for dry conditions, Δi_{lm} is used instead of ΔT_{lm}, logarithmic temperature difference.

$$Q_{avg} = U_{0,w} A_0 \Delta i_{lm} F \tag{1}$$

The logarithmic enthalpy difference calculation is defined according to Bump (1963) and Meyers (1967) as follows. $i_{a,m}$ represents the mean air enthalpy. $i_{r,m}$ saturated air temperature at the mean water temperature.

$$\Delta i_m = i_{a,m} - i_{r,m} \tag{2}$$

$$i_{a,m} = i_{a,in} + \frac{i_{a,in} - i_{a,out}}{\ln\left(\dfrac{i_{a,in} - i_{r,out}}{i_{a,out} - i_{r,in}}\right)} - \frac{(i_{a,in} - i_{a,out})(i_{a,in} - i_{r,out})}{(i_{a,in} - i_{r,out})(i_{a,out} - i_{r,in})} \qquad i_{r,m} = i_{r,out} + \frac{i_{r,out} - i_{r,in}}{\ln\left(\dfrac{i_{a,in} - i_{r,out}}{i_{a,out} - i_{r,in}}\right)} - \frac{(i_{r,out} - i_{r,in})(i_{a,in} - i_{r,out})}{(i_{a,in} - i_{r,out})(i_{a,out} - i_{r,in})} \tag{3}$$

The F factor is the correction factor used for cross-flow heat exchangers. For four rows and above, the correction factor is taken as "1" since the number of rows will be considered as a counter. The total heat transfer coefficient is related to the thermal resistances and is expressed in equation (4).

$$\frac{1}{U_{0,w} A_0} = \frac{1}{R_t} = \frac{b'_r}{h_i A_i} + \frac{b'_p \ln(\frac{D_o}{D_i})}{2\pi k_{tube} L} + \frac{1}{h_{o,w}\left[\dfrac{A_{p,o}}{b'_{wf,p,out}} + \dfrac{A_f \, \eta_{f,wet}}{b'_{wf}}\right]} \tag{4}$$

To find the value of $U_{0,w}$, must first determine the values of b'_r, b'_p, $b'_{wf,p,out}$, b'_{wf} which are shown in equations (5) and (6), respectively.

$$b'_r = \frac{\left(i_{r,p,in,m} - i_{r,m}\right)}{\left(T_{r,p,in,m} - T_{r,m}\right)} \qquad b'_p = \frac{\left(i_{r,p,out,m} - i_{r,p,in,m}\right)}{\left(T_{r,p,out,m} - T_{r,p,in,m}\right)} \tag{5}$$

$$b'_{wf,f} = \frac{i_{h,a}(T_{o,s}+1{,}1) - i_{h,a}(T_{o,s}-1{,}1)}{\left(T_{o,s}+1-T_{o,s}-1\right)} \qquad b'_{wf,f} = \frac{i_{h,a}(T_s+1{,}1) - i_{h,a}(T_s-1{,}1)}{\left(T_s+1-T_s-1\right)} \tag{6}$$

The Gnielinski correlation is used to find the heat transfer coefficient inside the tube. This correlation has separate correlations for laminar flow, transition region and turbulent region. The correlation in the transition region is used, but since the Reynolds number is above 5000, it gives almost the same result as the correlation used for turbulent flow. The Gnielinski correlation in the transition region is defined as follows. (VDI Heat Atlas, 2011)

$$Nu = \frac{\frac{f}{8}(\mathrm{Re}-1000)\,\mathrm{Pr}(T)}{1+12.7\frac{f}{8}^{0.5}\left(\mathrm{Pr}(T)^{2/3}-1\right)} \qquad f = (0.790\ln(\mathrm{Re}) - 1.64)^{-2} \tag{7}$$

$$h_i = Nu\frac{k_w}{D_i} \tag{8}$$

The Reynolds number in the correlation is calculated by taking the tube inner diameter as the characteristic length.

$$\mathrm{Re} = \rho V\frac{D_i}{\mu} \tag{9}$$

2.2 Fin Efficiency and Total Surface Efficiency

Fins are used in heat exchangers to increase the heat transfer through a surface. Fin efficiency is one of the important parameters in heat performance. Fin efficiency is defined as the ratio of the heat flow from the fin to the maximum heat that can be discharged from the fin under those conditions. The highest temperature difference for convection is the temperature difference between the temperature at the point where the fin contacts the tube, called the base temperature, and the fluid

$$\eta_f = \frac{q_f}{q_{\max}} = \frac{q_f}{hA_f\theta_b} \tag{10}$$

This study will analyse the straight fin type. The efficiency calculation for the fin will be based on the formula given for the flat fin model in the VDI Heat Atlas. (VDI Heat Atlas, 2011)

$$\eta_f = \frac{\tanh(X)}{X} \tag{11}$$

$$X = \varphi\frac{D_o}{2}\sqrt{\frac{2h_o}{k_{tube}\delta}} \tag{12}$$

$$\varphi' = 1.28\frac{b_f}{d_o}\sqrt{\frac{l_f}{b_f} - 0.2} \qquad \varphi = (\varphi' - 1)(1 + 0.35\ln(\varphi')) \tag{13}$$

Once the fin surface efficiency has been determined, the total surface efficiency can be calculated. The expression for the fin efficiency represents the thermal behaviour of a single fin, whereas the total surface efficiency describes the thermal behaviour of a fin array placed on a tubular array.

$$\eta_0 = 1 - \frac{A_f}{A_0}\left(1 - \eta_f\right) \tag{14}$$

The studies for fin efficiency and surface efficiency were applied for both dry and wet conditions. In the case of calculating fin efficiency and surface efficiency in wet conditions, the air side heat transfer coefficient calculated in the wet condition is used.

3. EXPERIMENTAL STUDY

3.1 Experimental Apparatus

This study was carried out in the test laboratory of R&D Center. There are two different rooms in the test laboratory. One of these rooms is called conditioning room and the other is called calorimetric room. The tests in this study were carried out in the conditioning room. Figure 1 shows a schematic picture of the conditioning room and the calorimetric room. On the right side is the calorimetric room where products with fans, such as evaporators and condensers, are tested. On the left side is the conditioning room where test of units without fans are carried out. The place numbered 1 shown in the conditioning room is the front side of the duct where the product is placed and represents the product to be tested. Representative image number 2 indicates the mixing room. The place number 3 is the room where the air flow rate is measured. Image number 4 represents the air handling unit

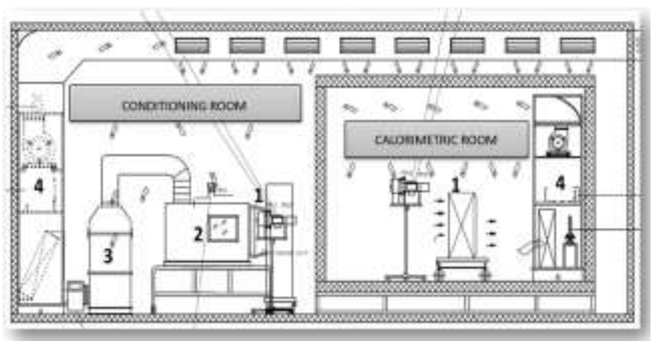

Figure 1: Experimental System Schematic Diagram

Air temperature, relative humidity, air flow rate, fluid temperature and fluid flow rate are measured during the tests carried out in the conditioning room. Before the air enters the heat exchanger, an air sampler is used to measure the air temperature and relative humidity by collecting air from 28 different points. Air temperature is measured with PT100 sensor. The relative humidity is measured by the Galltec® humidity sensor both at the air inlet and at the air outlet. On the fluid side, temperature measurement is made with the PT100 at the inletof the heat exchanger. In order to see the temperature change of the fluid, PT100 sensor is also used at the outlet. The air flow rate is adjusted with the help of the nozzles in the conditioning room. 4 nozzles with different diameters of Ø90 mm, Ø110 mm, Ø150 mm, Ø180 mm respectively creates a reverse pressure when opened and closed according to the desired air flow rate and adjust the air flow rate. The water flow rate on the fluid side is controlled by a YOKOGAWA ® brand digital flowmeter. Pressure losses for water side and air side are measured with the help of a differential pressure sensor. For the air side, the pressure loss value on the air side of the heat exchanger is found with the differential pressure sensor placed at the inlet and outlet zones of the heat exchanger. Similarly, on the water side, the pressure difference on the fluid side is obtained by placing a differential pressure sensor in the water inlet and outlet zone.

Figure 2: Experimental System Layout

Figure 3: Experimental System Layout-2

3.2 Experimental Uncertainty Analysis

In this study, the uncertainty calculations were initially conducted for the primary parameters that were directly measured from the experiments. Subsequently, the same methodology was employed to assess the uncertainty associated with the derived parameters, namely capacity, heat transfer coefficient and Colburn j-factor. Table 1. shows the uncertainty results.

Table 1: Uncertanity Analysis Results

Primary Parameters		**Derived Parameters**		
Parameters	Uncertainty	Parameters	Min. Uncertanity	Max. Uncertanity
ΔP	%1.5	Q_{avg}	3.03	7.24
$T_{w,i}$	0.1 ºC	Colburn j-factor	5.27	12.26
$T_{w,o}$	0.1 ºC	h_{air}	5.27	12.26
T_a	0.1 ºC			
m_a	%1.5			
m_w	%1			

3.3 Test Conditions

The conditions of 20 different experiments are presented in Table 2. The air inlet temperature, water inlet temperature and water flow rate were held constant. The relative humidity and air flow rate were variable parameters. The test conditions were selected from the conditions typically used in a representative air handling unit.

Table 2: Test Conditions

Air Inlet Temperature	20 ºC
Inlet Relative Humidty(%)	30-90-80-70
Air Side Volumetric Flow (m^3/h)	2000-2500-3000-3500-4000
Fluid Mass Flow Rate (kg/h)	3000
Fluid Inlet Temperature	8 ºC

3.4 Fin-and Tube Heat Exchanger Construction for Experiment

The fin-and-tube heat exchanger used in this study is constructed from copper tube and aluminum fins. The dimensions of the manufactured coil have been designed to align with the constraints of the duct in the conditioning room. The test conditions of the tested coil have been configured to align with the maximum capacity of the conditioning room. The following table provides information about the geometrical properties of the tested coil.

Table 3: Structural and geometric parameters of test unit

Tube Arrangement	31.75 x 28 mm	Fin Type	Flat Fin
Tube Count in a Row	28	Tube Diameter	12 mm
Row Count	4	Fin Thickness	0.12 mm
Number of Circuits	14	Tube Thickness	0.32 mm
Finned Length	750 mm	Tube Material	Copper
Fin Pitch	2.5 mm	Fin Material	Aluminium

4. EXPERIMENTAL RESULTS

The results of the experimental studies conducted in both dry and wet conditions are presented in Table 4 below. The experiments carried out at a relative humidity of 30% in a dry ambient conditions and condensation was not observed. However, condensation was observed in the experimental studies at relative humidities of 70%, 80% and 90%.

Table 4: Experimental Results

Test Number	Relative Humidity(%)	Air Volumetric Flow (m^3/h)	Air Outlet Temperature(°C)	Water Outlet Temperature(°C)	Heat Transfer Capacity (kW)	Air Side Pressure Drop(Pa)
1	30	2000	10.34	9.9	6.82	9.4
2	30	2500	11.03	10.28	7.83	12.9
3	30	3000	11.6	10.55	8.81	16.79
4	30	3500	12.09	10.77	9.63	21.52
5	30	4000	12.55	10.99	10.35	26.8
6	70	2000	11.31	10.8	9.65	15.48
7	70	2500	11.87	11.07	10.59	21.32
8	70	3000	12.47	11.35	11.37	29.03
9	70	3500	12.87	11.49	11.9	36.73
10	70	4000	13.36	11.75	12.37	45.64
11	80	2000	11.8	11.3	11.3	14.7
12	80	2500	12.5	11.7	12.6	20.9
13	80	3000	13.1	11.9	13.4	29.4
14	80	3500	13.6	12.2	14.2	36.1
15	80	4000	14	12.4	14.7	45.8
16	90	2000	12.3	11.8	12.9	16.2
17	90	2500	13.1	12.2	14.2	23
18	90	3000	13.6	12.5	15.2	30.3
19	90	3500	14.1	12.7	16	38.1
20	90	4000	14.5	12.9	16.5	47.2

The performance in this study is analysed by comparing the Colburn j-factor and Reynolds number parameters. To obtain the Colburn factor, the heat transfer coefficient is calculated for the air side. The forms of Equation (1) valid for dry and wet conditions are used. Here A_0 is the total external surface area. When dry and wet condition tests are performed, ΔT_m and Δi_m expressions are obtained from the test results in order to examine the total heat transfer performance. Q_{avg} expression is taken as the average capacity in the experimental results. The average capacity expression in the test results is the average of the fluid side capacity and air side capacity obtained in the test outputs. If the data obtained from the experimental results are substituted in the equation given to analyse the total heat transfer performance $U_{0,w}$ and U_0 is obtained

1. The coil surface was assumed to be free of contamination and pollution resistance was not considered.

2. It was assumed that the system was in a steady state.

3. The contact resistance between the tube and the fin inside the coil was not included.

4. The heat transfer coefficient in the tube was assumed to be constant.

5. The thermal conductivity of the tube and fin materials was assumed to be constant as a function of temperature.

Once the airside heat transfer coefficient has been determined, the Colburn-j factor can be defined. The heat transfer coefficient and pressure drop performances in heat exchangers have been reported with the Colburn j-factor at different Reynolds numbers. The Colburn j-factor is calculated in accordance with equation (15).

$$j = St\,\mathrm{Pr}^{2/3} \tag{15}$$

The Stanton number, as defined in Equation 4.13, is a dimensionless expression that represents the ratio of the heat transferred to a fluid to the thermal capacity of the fluid. It also provides insight into the thermal performance of a physical state with known flow dynamics. The Stanton number, as a dimensionless expression, is given in Equation (16).

$$St = \frac{Nu}{\mathrm{Re}\,\mathrm{Pr}} \tag{16}$$

In order to calculate the heat transfer coefficient, it is necessary to expand the expressions in the St number, taking into account the Colburn j-factor;

$$j = \frac{Nu}{\mathrm{Re}\,\mathrm{Pr}^{1/3}} \qquad j = \frac{h_0}{G_a \dfrac{Cp_a}{\mathrm{Pr}_a^{\;2/3}}} \tag{17}$$

4.1 Experimental Results for Dry Conditions

Similar studies are also available in the literature. Therefore, the Colburn j-factor and heat transfer coefficient obtained from the experimental results were compared with the correlations in the literature and frequently used Mcquiston, Wang *et al*. (2000), Rich's correlation and VDI Heat Atlas. These comparisons were made for 5 different Reynolds numbers. The hydraulic diameter in the reference Reynolds number calculation is the inner tube diameter in all correlations. Figure 4 and Figure 5 demonstrate the variation of the heat transfer coefficient and Colburn j-factor with Reynolds number for dry conditions.

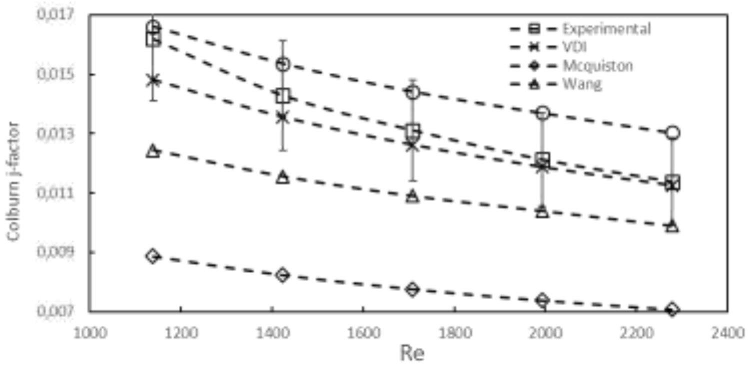

Figure 4: Colburn J-factor and Reynolds number graph obtained from correlations with experimental study

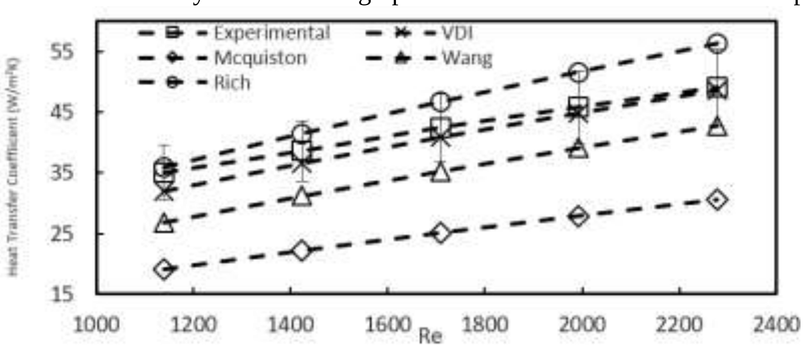

Figure 5: Heat Transfer Coefficient and Reynolds number graph obtained from correlations with experimental study

4.2 Experimental Results for Wet Conditions

We examined the behaviour of heat transfer coefficient and air side pressure loss at different relative humidities to understand how condensation forms. Our experimental studies showed that condensation occurs as droplet condensation at all relative humidities. We observed changes in the size and quantity of these droplets as the air flow rate changed at different relative humidities. We used a conventional method to measure the amount of condensed water in wet coil tests. We calculated how much water condensed in the coil and 5 minutes later poured it into a bottle. We then calculated the mass of the condensed water and converted it into hour units. Figure 6 shows the visuals of the condensation formed on the fin surface of the tests performed at different relative humidities. The tests were performed at different relative humidities, but the condensation formed was always droplet condensation. There was never any film condensation. Figure 6.a represents the test performed at 90% RH, where the water droplets of the condensing air are larger and clearer than at other RHs because the condensation is very high. Similarly, the condensed water droplets in the tests performed at 80% RH are larger than those formed at 70% RH and smaller than those formed at 90% RH. As expected, the least condensation occurred at 70% RH.

Figure 6.a: 90%RH

Figure 6.b: 80%RH

Figure 6.c: 70%

Figure 7 and Figure 8 demonstrate the variation of the heat transfer coefficient and Colburn J-factor with Reynolds number for wet conditions.

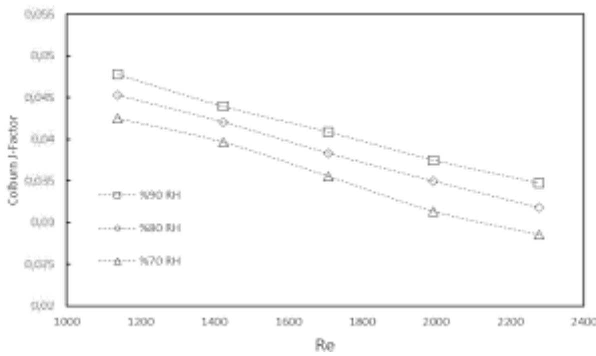

Figure7: Colburn J-factor and Re number graph obtained from correlations with experimental study for wet conditions

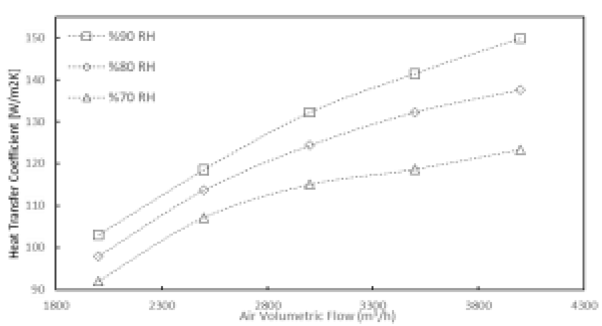

Figure 8: . Heat Transfer Coefficient and Air Volumetric Flow graph obtained from correlations with experimental study for wet condition

5. CONCLUSIONS

This study tested a fin-and-tube heat exchanger with a 4-row and 31.75 x 28 mm tube arrangement at different relative humidities. The experimental studies were carried out for dry conditions without condensation and for wet conditions with condensation. For dry conditions, comparisons were made with other correlations in the literature. Given the lack of suitable geometry for wet conditions in the literature, no comparison could be made with the existing correlations. We have obtained results regarding the thermal performance of the fin-and-tube heat exchanger. These results are discussed in terms of air side pressure loss, heat transfer coefficient and Colburn j-factor.

- In dry conditions, as the air velocity increases, the thermal capacity and air side pressure loss value increase. When the air velocity is doubled, the air side pressure loss value increases by 2.85 times.
- In dry conditions, the fin efficiency decreases as the Reynolds number increases. When the Reynolds number is doubled, the fin efficiency decreases by 6.2%.
- Colburn j-factor decreases when the Reynolds number is increased. The heat transfer coefficient forms a completely opposite behaviour from the Colburn j-factor. The heat transfer coefficient also increases as the Reynolds number increases. When the Reynolds number is doubled, the Colburn j-factor decreases by 30.5% while the heat transfer coefficient value increases by 35.5%.
- The heat transfer coefficient and Colburn j-factor values obtained from the experimental results are most similar to the correlation in the VDI Heat Atlas in the literature. While there was a 10% difference at low Reynolds numbers, this decreased to 1% when the Reynolds number increased. In the Rich's correlation (D.Rich,1975), while the results were quite similar at low Reynolds numbers, it was observed that when the Reynolds number increased, the results diverged from the results.
- It was commented that the geometry in this study was not similar to Mcquiston and Wang *et al*. (2000). It is interpreted that the correlations are not similar due to the difference in geometry. In wet conditions, Colburn j-factor and heat transfer coefficient increased as the relative humidity increased. As the Reynolds number increases, the Colburn j-factor decreases while the heat transfer coefficient increases.
- At the same Reynolds number and different relative humidities in wet conditions, the results vary between 3% and 10%.
- When the air side pressure loss values of dry and wet conditions are compared at the same Reynolds number, there is an average of 71% difference with a minimum of 56% and a maximum of 80% difference.
- Heat transfer coefficient and Colburn j-factor values are compatible with each other. While the heat transfer coefficient value in wet and dry conditions is approximately 2.5 times higher at low relative humidity, this ratio increases up to 3.1 times at 90% relative humidity.

NOMENCLATURE

A_0	External Surface Area	(m^2)
A_f	Fin Area	(m^2)
A_i	Tube Inner Area	(m^2)
Cp	Specific Heat	(kJ/kg)
D_o	Tube Outer Diameter	(mm)
D_i	Tube Inner Diameter	(mm)
f	Darcy Friction Factor	(-)
F	Correction Factor	(-)
G	Mass Flux	(kg/m^2s)
h_i	Heat Transfer Coefficient for Tubes	(W/m^2K)
h_o	Air Side Heat Transfer Coefficient	(W/m^2K)
i	Entalphy	(kJ/kg)
k_{tube}	Thermal Conductivity of tube	(W/mK)

k_w Thermal Conductivity of water (W/mK)
L Finned Length (mm)
Nu Nussel Number (-)
$\Pr$ Prandtl Number (-)
Re Reynolds Number (-)
R_t Total thermal resistance (W/ m^2 K)$^{-1}$
St Stanton Number (-)
T Temperature (^{o}C)
U_0 Overall Heat Transfer Coefficient (W/ m^2 K)
$U_{0,w}$ Overall Heat Transfer Coefficient For Wet (W/ m^2 K)
b'_r Slope of the air saturation curved at the mean water temperature (kJ/kgK
b'_p Slope between the outside and inside tube surface temperatures (kJ/kgK)

Greek Symbols

Q_{avg} Average Capacity (kW)
η_0 Surface Efficiency (-)
η_f Fin Efficiency (-)
Δi_{lm} Logaritmic Mean Enthalphy (kJ)
φ Adjusment Diameter (mm)
δ Thermal Conductivity of tube (W/mK)
θ_b Temperature Difference (-)
ΔT_{lm} Logaritmic Mean Temperature (^{o}C)
ΔP Pressure Difference (Pa)

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