Condensate Accumulation
Effects on the Air-Side Thermal
Performance of Slit-Fin Surfaces

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ACRC CR-26

January 2000

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ABSTRACT

Fin-and-tube heat exchangers are commonly used in air-conditioning systems and improving the efficiency of this component is very important. For a fin-and-tube heat exchanger used as an evaporator, moisture from the air can condense on the heat exchanger when its surface temperature is below the dew point. The moisture on heat exchanger surface usually increases the air-side pressure drop across the heat exchanger and affects the overall thermal-hydraulic performance significantly. The purpose of this study is to investigate condensate accumulation effects on air-side heat transfer performance, particularly for the slit-fin-and-tube heat exchanger. Real-time and steady-state mass of retained condensate are measured in order to characterize retention behavior. Plain-fin-and-tube and slit-fin-and-tube heat exchangers with varying fin spacing, numbers of tube rows and surface coatings are tested under dry and wet conditions. Using dimensional analysis and the experimental data, correlations to predict the heat exchanger performance under dry and wet conditions are developed for a slit-fin-and-tube heat exchanger. The correlations predict most of the experimental data within 20%. A simple model to predict the mass of retained condensate for the uncoated, slit-fin-and-tube heat exchanger is developed and compared to the experimental data, and the predictions agree with the experimental data relatively well.
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NOMENCLATURE

Roman Symbols:

a  constant used in B.1
A  area (m$^2$)
A$_{\text{drop}}$ area of a droplet (cm$^2$)
A$_{\text{fr}}$ frontal area (m$^2$)
b  constant used in B.1
c  constant used in B.1
C  coated (see Table 2.1), constant used in B.1
Cp specific heat at constant pressure (kJ/kg-K)
D  diameter (m)
D$_{\text{coll}}$ collar outside diameter (m)
D$_{\text{AB}}$ binary mass diffusion coefficient (m$^2$/s)
D$_{\text{drop}}$ diameter of droplet (m), Equation 4.3
D$_h$ hydraulic diameter (m)
E  Eckert number
$\Delta$P$_{\text{HX}}$ heat exchanger differential pressure (kPa)
EB  energy balance
f  friction factor
f$_s$ fin spacing (m)
g  gravitational acceleration (9.81 m/s$^2$)
G mass velocity based on minimum free flow area (kg/m$^2$-s)
h  enthalphy (kJ/kg)
h air-side heat transfer coefficient (W/m$^2$-K)
h$_l$ coolant-side heat transfer coefficient (W/m$^2$-K)
h$_s$ height of slit used in A.3.3.
H' parameter defined in A.3.2
H$_f$ height of heat exchanger
HX heat exchanger
\( j \) sensible \( j \) factor

\( \kappa \) thermal conductivity (W/m-K)

\( L \) dimension used in A.3.3. fundamental dimension of length (see Appendix B)

\( l \) length of heat exchanger in direction of air-flow

\( l_s \) width of condensate bridge (m) See Figure 4.8

\( l_s \) breadth of a slit in the direction of air flow (m) See Table 4.1

\( L_f \) length of fin (m)

\( L_{\text{max}} \) maximum length of fin-tube bridge neglecting air-flow forces (m) See Figure 4.8

\( m \) parameter defined in A.3.2

\( M \) dimension used in figure A.3.3. fundamental dimension of mass (see Appendix B)

\( n_s \) number of slits in an enhanced zone

\( N_{\text{tr}} \) number of tube row in the direction of air-flow

\( N_t \) total number of tubes

\( Nu \) Nusselt number

\( P \) Plain (see Table 2.1)

\( Pr \) Prandtl number

\( Q \) heat transfer rate (KW)

\( r_i \) inner radius of circular fin (m)

\( R \) thermal resistance (K/W)

\( R_c \) coolant flow rate (pulse/s)

\( S \) slit (see Table 2.1)

\( S_s \) width of an enhanced zone (m) Table 4.1

\( St \) Stanton number

\( S_T \) transverse tube spacing (m) (see Figure B.1)

\( S_L \) longitudinal tube spacing (m) (see Figure B.1)

\( T \) temperature (\(^\circ\)C), fundamental dimension of temperature (see Appendix B)

\( t \) time (s), fundamental dimension of time (see Appendix B)

\( UC \) uncoated (see Table 2.1)

\( V \) velocity (m/s)

\( V_{\text{bridge}} \) volume of bridge (m\(^3\)), Equation 4.4

\( V_{\text{drop}} \) volume of droplet (m\(^3\)), Equation 4.3
$W_d$ heat exchanger finned width (m)

**Greek symbols:**

- $\alpha$ angle of inclination (radians)
- $\delta$ fin thickness (m)
- $\delta_a$ gap distance between fins, see Table 4.1
- $\varepsilon$ $A_{to}/A_{to}$, see Table 4.1
- $\Phi$ fin efficiency
- $\phi$ relative humidity
- $\phi_s$ ratio of an enhanced zone area to a cell area, Table 4.1
- $\gamma$ surface tension (mN/m)
- $\eta$ surface effectiveness
- $\Pi$ dimensionless group used in Appendix B
- $\mu$ dynamic viscosity (N-s/m$^2$)
- $\theta$ contact angle (radians)
- $\theta_A$ advancing contact angle (radians)
- $\theta_M$ mean contact angle (radians) $\theta_M=\frac{\theta_A+\theta_R}{2}$
- $\theta_R$ receding contact angle (radians)
- $\rho$ density (kg/m$^3$)
- $\sigma$ contraction ratio ($A_{min}/A_{fr}$)

**Subscripts:**

- $\text{air}$ air
- $\text{atm}$ atmospheric pressure (atm)
- $\text{ave}$ average
- $c$ coolant
- $\text{cali}$ calibrated
- $\text{coll}$ collar
- $\text{dp}$ dewpoint
- $f$ fin
if inner fin
in inlet
it inside tube
l liquid
mair mean air
min minimum
max maximum
out outlet
ot outside tube
read reading
s slit
t tube
tot total
to tube outside
CHAPTER 1 – INTRODUCTION

1.1 Introduction

Air-conditioning systems are used all throughout the world to provide human comfort, and their use consumes substantial energy resources [1]. Therefore, improving the efficiency of this system has attracted significant research attention. Fin-and-tube heat exchangers are commonly used in air-conditioning systems and their performance has been widely studied. When the fin-and-tube heat exchanger is used as an evaporator, moisture from the air can condense on the heat exchanger because the surface temperature is usually below the dew point of the air. Condensation formed on the heat exchanger surfaces can take either dropwise or filmwise shapes, and it accumulates until removed by gravitational or air-flow forces. The retained condensate usually increases the air-side pressure drop across the heat exchanger and affects the overall thermal-hydraulic performance significantly. Although there has been research to characterize the effects of retained condensate on heat exchanger performance, these effects depend on geometry and operating conditions, and the effect of retained condensate on fin-and-tube heat exchanger performance is not well understood.

The goal of the current study is to investigate condensate accumulation effects on air-side heat transfer performance, particularly for the slit-fin-and-tube heat exchanger. This work will be used to develop correlations to predict the heat exchanger performance under dry and wet conditions, and for a model to predict the quantity of retained condensate on the slit-fin-and-tube heat exchanger surface as a function of geometry, contact angle, and air-side Reynolds number. A wind tunnel was constructed to measure real-time and steady-state quantities of retained condensate as well as heat transfer performance. Flow directions of the air and the coolant were perpendicular to each other, allowing counter-cross-flow tube circuiting for each heat exchanger tested. The heat transfer performance results under dry and wet conditions were compared to results in extant literature. A simple model to predict the mass of retained condensate for slit-fin-and-tube heat exchanger was developed and compared to the experimental data. Further studies on designing methods and guidelines to improve evaporator performance are outlined.
1.2 Literature Review

The literature review consists of the early studies on the enhanced sensible heat transfer performance and the studies of fin-and-tube heat exchangers. Previous studies on modeling condensate retention are also discussed here.

1.2.1 Enhanced Sensible Heat Transfer Performance

Bryan [2] tested a bare tube surface coil under dehumidifying conditions. Actual surface temperatures of the tube coils were measured experimentally by using each coil as a resistance thermometer in a Kelvin bridge circuit, which read the tube resistance to 0.5 μΩ. The resistance-to-temperature ratio was 4 μΩ per degree F. He also calculated the tube surface temperature based on the Lewis relation, assuming Le = 1. The actual tube surface temperature was found to be lower than the calculated tube surface temperature. Bryan explained that these differences were due to the exponential effect of the air velocity variation in the Lewis relation. A small change in the Lewis relation results in a large change of the tube surface temperature for a given heat transfer performance as the air progresses over the tubes. He calculated the heat and mass transfer coefficients directly with the actual tube surface temperature of the coils and reported that the heat transfer coefficient and the friction factor under wet conditions were higher than those under dry conditions. In 1962, Bryan [3] presented experimental data for heat and mass transfer on dehumidifying extended-surface coils. Six rows of individual integral-fin copper tubes with 9 fins per inch were tested under dry and wet conditions. By the method used in his previous experiments for bare-tube coils, the heat and mass transfer coefficients were calculated. Bryan concluded that the heat transfer coefficient was greater in dry conditions, whereas, the bare coil experimental data showed the opposite result. The difference was caused by a decreased fin-coil surface effectiveness due to the condensation on fin surfaces.

Bettanini [4] reported an enhanced sensible performance under condensing conditions for vertical flat plates. The experiment was conducted under dry and wet conditions at three different air velocities. He observed that the air-to-surface heat transfer coefficient for wet conditions was higher than under dry conditions. Bettanini explained this result as being due to the condensation which roughened the flat plate surface. He reported that the mass transfer affected the heat transfer and relative coefficients. The enhancement experiment was conducted by simulating the
condensation with solid gypsum drops. Bettanini also investigated smooth-plate heat transfer performance and determined a direct relationship between the surface roughness and the heat transfer coefficient. According to him, this study confirmed that the effects of the increased effective roughness were due to the condensation.

Guillory and McQuiston [5] and McQuiston [6] studied developing flow between horizontal flat plates. The experimental $j$ factors for dry and wet conditions were compared to analytical predictions, and the experimental data were higher than the predicted data. They found a heat transfer enhancement of about 30% for the wet surface condition, and explained it by the roughening of the walls caused by condensate deposition. McQuiston observed an enhancement of the friction factor by 25% under wet conditions. These studies showed that the effect of condensation on the heat and mass transfer coefficients was significant in a developing flow field.

Tree and Helmer [7] tested two parallel-plate heat exchangers. The experimental data provided by Tree and Helmer agreed with Guillory and McQuiston’s analytical prediction better than the experimental data provided by Guillory and McQuiston. Tree and Helmer found no effect of condensation on sensible heat transfer and pressure drop characteristics for developing laminar flow, but there was a small effect on fully developed laminar flow between horizontal flat plates. Under dehumidifying conditions, both the sensible heat transfer and the pressure drop were higher than those under dry conditions.

1.2.2 Fin-and-Tube Heat Exchangers

Fin-and-tube heat exchangers are widely used in air-conditioning and refrigeration applications. In order to increase the air-side heat transfer coefficient, several heat exchanger surface designs for fin-and-tube heat exchanger have been adopted. In this section, plain-fin-and-tube, slit-fin-and-tube, and other types of fin-and-tube heat exchangers which are extensively used in air-conditioning and refrigeration industries are reviewed.

A. Plain-Fin-and-Tube Heat Exchanger

In 1973, Rich [8] studied the effect of fin spacing on heat transfer performance for multi-row, plain-fin-and-tube heat exchangers. The heat exchanger was made of copper fins that were smooth and continuous with no collars, and four staggered tube rows in a multi-pass cross-flow
circuit. When the number of fins was low, he found that the friction factor varied only slightly with the Reynolds number, but as the fin spacing decreased, the friction factor variation increased progressively with the Reynolds number. Rich assumed that the overall pressure drop might be comprised of contributions from both the tubes and the fins. He concluded that the heat transfer coefficient and the friction factor were independent of fin spacing within the range of 3 to 21 fins per inch, and 3 to 14 fins per inch respectively, at a given mass velocity.

McQuiston [9] observed the heat and mass transfer performance of five plain-fin-and-tube heat exchangers with 8, 10, 12 and 14 fins per inch in dry and wet conditions. The fins for the heat exchanger were made of plain flat aluminum with collars while the tubes were made of copper. Data were obtained for three different surface conditions for each heat exchanger: dry surface, wet surface with filmwise condensation, and wet surface with dropwise condensation. McQuiston found that the effect of moisture on the surface became more significant at higher Reynolds numbers, and the $j$ factor for the dropwise condensation was larger than that for the filmwise condensation. He also found that with the wider fin spacing, the effect of condensate in the filmwise condition became minor, while dropwise condensation increased the friction factor significantly. He concluded that the heat transfer performance strongly depended on the fin spacing and mode of condensation.

In 1978, McQuiston [10] developed the generalized correlations of $j$ and $f$ factors for plain-fin-and-tube heat exchangers with staggered tube arrangement. Five four-row, plain-fin-and-tube heat exchangers with fin spacings of 4, 8, 10, 12, and 14 fins per inch were used to develop the correlations. McQuiston took into account all of the flow and geometric variables for his correlations. The air-flow direction was downward through the heat exchanger. To determine the tube row effect, he developed an empirical correlation using Rich's data [11]. The correlations were developed for dry conditions, wet conditions with film-type condensation, and wet conditions with drop-type condensation. Uncertainties for the $j$ and $f$ factor correlations were $\pm 10\%$ and $\pm 35\%$, respectively.

Seshimo et al. [12] measured the air-side performance of a single-row plain-fin-and-tube heat exchanger under dehumidifying conditions. They treated the aluminum fin surface by boiling it in a 0.5% aqueous ammonia solution. During the experiments, the entire fin surface was covered with a uniform condensate film. Seshimo et al. concluded that the heat transfer and the
pressure drop were about 20% and 30–40% greater, respectively, due to the condensation effect. Since the shape of the heat exchanger geometry was changed due to the presence of the condensate, Seshimo et al. concluded that the heat transfer coefficient under wet conditions can be calculated in the same way as it is under dry conditions if the equivalent thickness of the condensate film is accounted for the fin thickness. The equivalent thickness of the condensate can be calculated from the increase in the air-side pressure drop.

Kundu and Das [13] studied the optimum fin thickness for plain-fin-and-tube heat exchangers having either in-lined or staggered tube arrangements. The heat transfer rate from the fin was calculated for each dimensionless fin volume. They found that, for a fixed fin volume, the rate increased with the plate thickness, reached a maximum, and then fell gradually. As the fin volume increased, the optimum plate thickness, which yield the maximum heat transfer, was also found. Kundu and Das concluded that the plain-fin-and-tube heat exchangers with a staggered tube arrangement had a greater heat transfer rate by the fin surface than that of the in-lined tube arrangement.

Wang et al. [14] observed the effects of the number of tube rows, fin spacing, and the inlet condition for plain-fin-and-tube heat exchanger under dehumidifying conditions. Nine plain-fin-and-tube heat exchangers were tested with fin spacings of 1.82 to 3.2 mm, and 2, 4 and 6 tube rows. They derived and calculated the circular wet fin efficiency based on the approximation proposed by Threlkeld [15]. The performance of the heat transfer and the friction factor were measured at relative humidities of 50 and 90%, and they concluded that the inlet air temperature did not affect the sensible $j$ factors. The friction factors of a fully wet surface did not depend on the fin spacing, the number of tube rows, and the inlet air condition. The friction factor in wet conditions was greater than in dry conditions. The degradation of the sensible heat transfer coefficients in wet conditions was observed at a low Reynolds number while a small increase was observed at a higher Reynolds number. Wang et al. developed correlations for the $j$ and $f$ factors under dehumidifying conditions based on the number of tube rows and fin spacing. The correlations predicted their experimental data for $j$ and $f$ factors $\pm 10$ percent.

In 1998, Chuah et al. [16] studied the performance of a three-row, plain-fin-and-tube heat exchanger in dehumidifying conditions. The experimental data for heat and mass transfers at various air velocities and water flow rates were measured and shown to have the same trends as
the heat and mass transfer data analyzed in other published correlations. The dehumidification capacity was overpredicted by as much as 1.5 to 2 times when measured at the maximum air velocity, but this discrepancy was explained by the differences in the number of tube rows used. Chuah et al. concluded that as the water flow velocity increased, the dehumidification capacity increased; however, when the air velocity increased at a given water flow velocity, the dehumidification capacity decreased except at the highest water flow velocity of 1.07 m/s. They suggested further studies on the revised correlations to account for the effect of the number of tube row in order to predict the performance of a three-row, plain-fin-and-tube heat exchanger.

B. Slit-Fin-and-Tube Heat Exchanger

The slit-fin-and-tube heat exchanger was developed in the early 1970’s by Hitachi, Ltd. It aimed to improve the air-side heat transfer coefficient by minimizing the thickness of the boundary layer at the end of the fin edge without increasing the pressure drop. Slits were raised at equal intervals from the plain-fin surface to create new boundary layers for each slit, as shown in Figure 1.1. Newly developed boundary layers then maintained the high air-side heat transfer coefficient through the fin.

Hosoda et al. [17] presented the principles and characteristics of the slit-fin-and-tube heat exchanger by comparing it to a corrugated-fin type heat exchanger. The performances in dry and wet conditions were evaluated. The heat transfer coefficient on the air-side was calculated from the temperature difference between the air and tube surfaces, created by hot water flowing in the tubes, and varying the air velocity. Hosoda et al. found that at a frontal air velocity of 1.5 m/s under dry conditions, the air-side heat transfer coefficient of the slit-fin-and-tube heat exchanger was 1.6 times greater than that of the corrugated-fin type, while the air pressure drops for both heat exchangers were approximately the same. They also observed that as the width of the slit became narrower, the air-side heat transfer coefficient increased. The air-side heat transfer coefficient also increased due to the latent heat in the wet condition. The power consumption of a room air conditioner was reduced by 20% when the slit-fin-and-tube heat exchanger was used to cool the same heat exchange area as the corrugated-fin type.

Ito et al. [18] compared the performance of the slit-fin-and-tube heat exchanger to the plain-fin-and-tube heat exchanger. They found that the heat transfer coefficient of the slit-fin-and-
tube heat exchanger for a car air-conditioner was 35% higher than that of the plain-fin-and-tube
heat exchanger. However, while the heat transfer coefficient increased when the slit-fin-and-tube
heat exchanger was used under dehumidifying conditions, the condensate, in the form of droplets
and as a film, narrowed the air path, thus increasing the air pressure drop. Ito et al. [19] found
experimentally that the wettability of the fin surface could be improved by roughening the fin
surface of the heat exchanger. They observed that the rough surface changed the dropwise
condensate on the fin to the filmwise condensate, which was removed easily by the gravitational
force. Therefore, in order to decrease the air pressure drop, the fin surface was roughened. The air
pressure drop in wet conditions decreased by 30% after the hydrophilic treating on the fin surface,
and there was no effect on the heat transfer coefficient of the slit-fin-and-tube heat exchanger.

Even though the slit-fin-and-tube heat exchanger provided a fairly good average heat
transfer coefficient, the temperature differences between the fins and the air in the downstream
region were very small due to the poor mixing of the air stream along the slit-fin. In order to
overcome this disadvantage of slit-fin-and-tube heat exchangers, a super-slit-fin-and-tube heat
exchanger was developed by Arai et al. [20]. A super-slit-fin-and-tube heat exchanger, which had
a louvered-slit configuration, was developed by observing the flow patterns along the fin surfaces
for various fin configurations using a water channel model. The heat transfer coefficient was
improved by 23% using the super-slit-fin-and-tube heat exchanger, but an increase in the air
pressure drop of 17% was also observed.

In 1983, Nakayama and Xu [21] developed predictive correlations for the Colburn $j$ factor
and the friction factor for plain-fin-and-tube and slit-fin-and-tube heat exchangers with a
staggered tube arrangement. They applied the appropriate heat transfer correlation to the each
zone of a fin that was divided into various regions. The numerical analysis for fin temperature and
heat transfer were developed and checked with a simplified heat transfer model. The optimum
design was developed for the plain-fin-and-tube and the slit-fin-and-tube heat exchangers in terms
of the number of tube rows with a specific fan power and a given temperature difference.
Uncertainties for the $j$ and friction factor correlations were ±10% for $250 \leq \text{Re}_{db} \leq 3000$,
$0.15 \text{mm} \leq \delta_f \leq 0.2 \text{mm}$, $1.8 \text{mm} \leq \ell \leq 2.5 \text{mm}$, and $0.2 \leq \phi \leq 0.35$ for both the plain-fin-and-tube and the
slit-fin-and-tube heat exchangers. Nakayama and Xu also recommended that a slit-fin-and-tube
heat exchanger with less than four tube rows to be used for an optimum design.
C. Other Fin-and-Tube Heat Exchangers

Chang et al. [22] reported on the performance characteristics of a louvered-fin-and-tube heat exchanger. Seven louvered-fin-and-tube heat exchangers with varying fin spacings and numbers of tube rows, and one two-row, plain-fin-and-tube heat exchanger were tested under dry conditions. The frontal velocity range was from 0.5 to 5m/s and the energy balance between the air and the tube side was within 5%. Colburn $j$ factor and $f$ factor for the plain-fin-and-tube heat exchanger were calculated based on the experimental data and were in agreement with the correlations of Gray and Webb [23] and McQuiston [10]. The $j$ and $f$ factors for the louvered-fin-and-tube heat exchanger were higher than those for the plain-fin-and-tube heat exchanger, and the differences in the $f$ factors under dry and wet conditions were greater than in the $j$ factors. Chang et al. observed that $j$ and $f$ factors were not affected by the fin spacing, and the number of tube rows had no significant effect on the $f$ factors. For a Reynolds number was less than 3000, the $j$ factor decreased as the number of tube rows increased.

In 1995, Fu et al. [24] investigated the effect of an anti-corrosion coating on the thermal characteristics of a louvered-fin-and-tube heat exchanger under dehumidifying conditions. The experiment was conducted with three anti-corrosion coated louvered-fin-and-tube heat exchangers with various fin spacings, and one non-coated louvered-fin-and-tube heat exchanger with a louver angle of 32 degrees. Fu et al. provided the experimental data of the air-side performance on the louvered surface under dry and wet conditions. Total $j$ factor was calculated using the total heat transfer coefficient, and the sensible $j$ factor was determined by the air-side heat transfer coefficient. They found that even though the total $j$ factor did not depend on the operation inlet condition, the sensible $j$ factor and friction factor did, especially the relative humidity. They also concluded that the Colburn sensible $j$ factor did not depend on fin spacing in dry conditions, but the total $j$ factor, friction factor, and sensible $j$ factor all depended on fin spacing. The friction factor and the heat transfer coefficient of the louvered-fin heat exchanger in wet conditions were found to be much greater than those in dry conditions. Moreover, the anti-corrosion coating had a negligible effect on the thermal hydraulic characteristics of the louvered-fin-and-tube heat exchanger.

Yun and Lee [25] studied scaled-up and prototype experiments on the heat transfer characteristics of various kinds of fin-and-tube heat exchangers with interrupted surfaces. Fin-
and-tube heat exchangers used in air-conditioning systems with five different kinds of fin shapes in a two-row, staggered arrangement were tested. One plain-fin-and-tube, two different geometries of slit-fin-and-tube, one louvered-fin-and-tube, and the reference fin-and-tube heat exchangers were tested. The data were compared to prototype test results and the results were in relatively good agreement with each other. The uncertainty of the heat transfer coefficient and the pressure drop on the scaled-up experiment were 3.6~4% and 2.8~5% respectively. The $j$ and $f$ factors for five different fin shapes were reported. The slit-fin-and-tube heat exchanger had higher $j$ factors among the five kinds of fin shapes and the louvered-fin-and-tube heat exchanger had the highest $f$ factors. Both the slit-fin-and-tube and the louvered-fin-and-tube heat exchangers showed higher $j$ and $f$ factors than the plain-fin-and-tube heat exchanger. Yun and Lee recommended a slit-fin with a group of four, rectangular, arrayed slits in the central section as an optimal fin geometry.

A wavy-fin-and-tube heat exchanger was tested in 1993 [26] and correlations for the Nusselt numbers and the friction factor under dehumidifying conditions were reported by Mirth and Ramadhyani in 1994 [27]. Five different wavy-fin-and-tube heat exchangers with a staggered tube arrangement were tested and individually correlated. Mirth and Ramadhyani found that the heat transfer coefficient was greater for shorter coils, but the length dependency decreased as the Reynolds number increased due to vortices formed at the base of the tubes. They also reported that larger fin spacings showed a higher heat transfer coefficient. General Nusselt number correlations were developed from the dry experiment data, which accounted for the fin spacing and the coil length. All the data fell within 20% of the correlation. The correlation to predict the friction factor in wet conditions was also developed by increasing the wet friction factor relative to the dry friction factor to an uncertainty of 4 to 11%.

Youn et al. [28] reported the pressure drop and the heat transfer characteristics of a wavy- and a wavy-slit-fin-and-tube heat exchanger under dry conditions. A plain-fin-and-tube heat exchanger was used to validate previous works. The wavy-fin-and-tube heat exchanger showed 20~45% higher friction factors than the plain-fin-and-tube heat exchanger. $j$ and $f$ factor correlations were developed for the specific heat exchanger geometry. Youn et al. concluded that the wavy-fin-and-tube heat exchanger was better to use at higher velocities and wider fin spacing,
and the wavy-slit-fin-and-tube heat exchanger at higher velocities and a smaller number of tube rows.

1.2.3 Designing Fin-and-Tube Heat Exchanger

The heat exchanger is one of the most important components of the air-conditioning system. In order to improve the efficiency of the heat exchanger, there have been numerous studies on designing heat exchanger geometry. The plain-fin-and-tube heat exchanger has changed its shape to corrugated, slitted, and louvered-fin-and-tube heat exchangers, inner grooved tubes were developed, and the tube arrangement has been studied. Fin spacing and the number of tube rows are also important in designing the optimum fin-and-tube heat exchanger along as are the fin shapes.

Some of existing literature conclude that the condensation on the fin-and-tube heat exchanger surface influenced the heat transfer performance. In order to minimize the formation of the condensate on the heat exchanger surface, Hong studied the effect of hydrophilic surface coating on various fin-and-tube heat exchangers [29]. Hong improved the surface wettability by coating the wavy-, lanced-, and louvered-fin-and-tube heat exchangers. Hydrophilic-coated heat exchangers were tested along with uncoated heat exchangers, and the results were compared to an uncoated heat exchanger. A maximum 45% reduction in the wet to dry pressure drop ratio of the fin-and-tube heat exchanger was observed. Hong reported that the hydrophilic coating did not affect the heat transfer performance, though the wet pressure drop decreased. A model to predict the carry-over velocity was developed and compared to the experimental data. Hong presented the contact angle data obtained from a sessile-drop goniometer test. However, because a static test procedure was adopted, no measurement of contact angle hysteresis was obtained. The result between the advancing and receding contact angle was all that could be achieved through such an approach. A convergence of the wettability of the coating was observed as the number of the dry/wet cycles increased. Hong found that after approximately 1,000 wetting cycles, all the test surfaces (coated and uncoated) exhibited contact angles of approximately 60 degrees.
1.2.4 Condensate Retention Modeling

Korte and Jacobi [30] studied the condensate retention on the plain-fin-and-tube heat exchanger with fin spacing of 4, 8, 10 and 12 fpi. They observed the effect of the condensate retention on the air-side heat transfer performance by comparing the $j$ and $f$ factors in dry and wet conditions. Proposed techniques for modeling condensate retention were discussed. The mass of retained condensate was calculated by multiplying the total volume of droplets and the density of the water. By knowing the diameter of each droplet and the contact angles between the drop and the surface, the volume of each droplet was approximated. The drop distributions were determined using a technique suggested by Graham's [31] work, and the maximum diameter of the droplet on the vertical surface was determined by the force balances. An initial model of the condensate retention for wider fin spacing (4fpi) was developed to predict the quantity of condensate retention as a function of the heat exchanger geometry, advancing and receding contact angles, and air-side Reynolds number. The model overpredicted the condensate retention by approximately 15% at high air-flow rates, but at low air-flow rates, it predicted the results fairly well. Further retention modeling was recommended for closer fin spacing and different heat exchanger geometry.

1.2.5 Conclusions

Condensate retention and shedding profoundly affect the heat transfer and pressure drop performance of evaporators operating under wet-surface conditions. Furthermore, condensate retention has important implications on the air quality, as water provides a medium for biological activity on the air-handling surfaces. Its importance to thermal performance and air quality notwithstanding, the open technical literature provides no clear guidelines for condensate management. In point of fact, there is disagreement as to the overall impact of condensate retention on thermal performance, and there is no general model for predicting retention. Guidelines for condensate management and predictive tools for fin-and-tube heat exchangers are needed by engineers designing the next generation of strip-fin air-side surface technology. There have been only a few studies of slit-fin-and-tube heat exchangers under dehumidifying conditions and the impact of the condensation in this heat exchanger geometry remains unclear.
1.3 Objectives

The objectives of this project are to obtain thermal and retention data under dehumidifying conditions for a slit-fin-and-tube heat exchanger, to develop a model of the condensate retention that can predict the mass of retained water on the slit-fin-and-tube heat exchangers at steady state, and to provide correlations for the $j$ and $f$ factors under dry and wet conditions. These correlations will allow prediction of the thermal performance of the slit-fin-and-tube heat exchanger as a function of fin spacing, the number of tube rows, and the coating on fin surface. The experimental methods and analytical modeling were to be combined in order to validate the model and the correlation.

The specific objectives are as follows:

- **Thermal data for fin-and-tube heat exchangers**: Measurements of sensible heat transfer, latent heat transfer, air-side pressure drop, and mass of retained condensate for plain-fin-and-tube and slit-fin-and-tube heat exchangers over the range of conditions typical to air-conditioning system were obtained. The thermal data allowed for a comparison to technical literature and a characterization of slit-fin-and-tube heat exchanger performance relative to the plain-fin-and-tube heat exchangers.

- **$j$ and $f$ factors correlation**: Based on the experimental data, correlations for the $j$ and $f$ factors for the plain-fin-and-tube and slit-fin-and-tube heat exchangers were developed. This correlation will predict the thermal performance of the specific fin-and-tube geometries under certain range.

- **A model of condensate retention**: A model to predict the mass of the retained condensate on slit-fin-and-tube heat exchangers at steady state was developed. The model accounted for surface tension, gravity, pressure, and shear forces on condensate elements.

- **Design Guidelines**: The design methods and guidelines of this project for condensate management and thermal performance of the slit-fin-and-tube heat exchanger will lead to recommendations for new slit-fin-and-tube configurations to improve evaporator performance for the wet operating condition typical to air-conditioning applications.
Figure 1.1 Principle of slit-fin (Ito et al., 1977)
CHAPTER 2 - EXPERIMENTAL APPARATUS AND PROCEDURE

A closed-loop wind tunnel was designed and constructed to operate with a horizontal air-flow. It allowed both the real-time and steady-state retention measurements as well as the heat transfer performance data measurement. This chapter describes the experimental apparatus and instruments used for this experiment and the physical geometry of specimens.

2.1 Experimental Apparatus and Instruments

The test apparatus consisted of three main components; a closed-loop wind tunnel, a test section, and a single-phase coolant loop. A calibrated electronic balance was used to measure the mass of retained condensate, and a contact angle goniometer was used to measure the contact angles of water droplet on the specimen surface. A computerized data acquisition system was used to automate the experiment and to avoid possible human errors.

2.1.1 Wind Tunnel

The wind tunnel had a closed-loop configuration to make thermal and humidity control easier in the confined laboratory. A schematic of the wind tunnel is shown in Figure 2.1. The air in the thermal mixing chamber was brought to a uniform temperature and humidity through macroscopic mixing caused by static mixing plates installed at the chamber inlet. At the exit of this chamber, the flow was conditioned with a honeycomb flow straightener and screens. These devices provided a uniform, steady flow through a rectangular area with contraction ratio of 9:1. The two-dimensional flow contraction reduced the free-stream turbulence intensity. Thus, the flow would have uniform velocity, humidity, and temperature profiles with low turbulence intensity at the measurement station upstream of the test section. Air was drawn by a diaphragm air pump through fluted tubes, that were installed in the inlet and outlet air-side measurement stations. The capacity of the humidifier was 11kg/hr of steam. Downstream of the test section consisted of a second measurement station, a flow conditioning section, and a diffuser. Within the diffuser were electrical resistance strip heaters and a steam injection system to
control the temperature and humidity within the tunnel. These devices were located immediately upstream of the blower to take advantage of mixing caused by the axial-flow fan. The fan could provide a volumetric flow rate up to 20 m$^3$/min. The secondary strip heaters were located downstream of the fan to deliver higher heat to the test section and also to maintain the desired inlet condition easily. Up to 8kW can be added to the airflow with both first and secondary heaters.

The air dew point, temperature, and flow rate were controlled within the closed loop wind tunnel and recorded through the computer using data acquisition system. The air dew point was measured using a chilled-mirror hygrometer at the inlet and outlet of the measurement stations of the wind tunnel to determine mass heat transfer rates. The measured dew point provided a control signal to a PID controlled humidifier that supplies water vapor to the wind tunnel, maintaining the desired dew point throughout the experiment. The uncertainty of chilled-mirror hygrometer was ±0.2°C. The air-side temperatures were measured with type-T thermocouples installed at the inlet and outlet measurement stations located upstream and downstream of the test section. The thermocouple grids consisted of six-thermocouples for inlet, and twelve-thermocouples for outlet—thermocouple readings were used and averaged to provide the inlet and outlet air-side average temperatures. A vacuum flask filled with ice was used as a reference for each individual thermocouple. The ice was well-stirred to ensure the ice bath was uniform. The air temperature approaching the test heat exchanger was controlled manually by varying power supplies for both strip heaters. The air velocity was measured with a constant-temperature thermal anemometer and controlled by a motor controller on the blower. The wind tunnel was well insulated to avoid condensation on the wind tunnel walls.

2.1.2 Test Section

The test section consisted of the body, frame, and support. It was constructed with clear acrylic to enable visualization and was insulated to avoid possible heat loss. The test section allowed both real-time (transient) and steady-state measurements of the retained condensate on the heat exchanger surface. The body of the test section provided an easy access for a constant-temperature thermal anemometer used to measure air velocity at the
face of the heat exchangers. The anemometer was inserted along five traverses, as shown schematically in Figure 2.2. From these data, an average face velocity was determined for each test condition. The calibrated uncertainty of the anemometer was ±1%. The air-side pressure drop across the heat exchanger was measured with an electronic micro-manometer using static pressure taps located at the upstream and downstream of the test body. The uncertainty of the electronic micro-manometer was ±0.0005 in. of water. A funnel, made with vinyl and a plastic tube, was installed at the bottom of test body to drain the water during the experiment. The frame of the test section provided a mechanical interface with the test heat exchanger. The frame mated with the heat exchanger inlet and outlet so there was no leakage of flow; however, the frame allowed the test heat exchanger to slide freely up and down. The frame also allowed an easy specimen removal for measuring the steady-state mass of retained condensate. Two different sizes of frame were provided due to the two different lengths of the specimens. The test section support held the heat exchanger within the frame and rested on an electronic balance to allow the real-time measurements of retained condensate mass. A schematic of the test section is provided in Figure 2.3. The test section design facilitated in situ measurements of retained condensation mass.

2.1.3 Coolant Loop

The pumps in the coolant loop circulated a diluted single-phase ethylene glycol and water mixture with inhibitors (DOWTHERM 4000) on the tube side of the heat exchanger. The temperature of coolant was measured using type-T immersion thermocouples installed in the insulated copper tubing approximately 2 m upstream and downstream of the heat exchanger. Each thermocouple in inlet and outlet was referenced to the ice bath individually. The heat exchanger was connected to the insulated copper tubing using an insulated flexible, reinforced, PVC tubing and quick-connect valves. An oscillating-piston flow meter was used to measure coolant flow rate with an uncertainty of ±0.5%. A transmitter attached to the flow meter provided a 1-5 VDC pulse with a frequency proportional to the volumetric flow rate. The number of pulses was counted over a time cycle using a programmable timer/counter. The temperature and the number of pulses were recorded to the data acquisition system over a time period.
2.1.4 Retained Condensate Measurements

A calibrated electronic balance was used to measure the real-time and steady-state retained condensate. The real-time condensate was measured by placing the electronic balance under the supporter of the test section and starting the timer. The steady-state condensate was measured by removing the heat exchanger from the test section and weighting the heat exchanger with an electronic balance with the readability of 0.1g and reproducibility of 0.1g.

2.1.5 Contact Angle Measurements

The contact angle of the fin stock was measured with contact angle goniometer. A fin stock held by the specimen holder was placed on the viewing platform and the droplet was drawn on the specimen with a micro-syringe. Advancing and receding contact angles were measured by rotating the platform. A telescope with a magnification ratio of 7:1 and the crosshairs on the lens were used to view and measure the droplet contact angle. The droplet was illuminated with the light source and was covered to avoid the evaporation of the droplet.

2.1.6 Thermocouple Calibration and Data Acquisition System

All Type-T thermocouples were calibrated with a NIST traceable mercury-in-glass thermometer using an isothermal temperature controlled bath. The calibrated data of the individual thermocouple were fit to the fifth order polynomials and the polynomial was used in the data acquisition system. The computerized data acquisition system provided the measured values of all the temperature and the coolant flow rates in a data file for a subsequent analysis. It also averaged the inlet and outlet air-side temperatures. The data acquisition system sampled 25 channels and averaged them over 11 measurements. The averaged values were then recorded at 45-second intervals. When the averaged values recorded in the data acquisition system was varied in a small range, the experimental condition reached a steady state. The steady-state experimental conditions were determined by monitoring the air and coolant inlet temperatures.
2.2 Physical Geometry of Specimens

Four different types of heat exchangers were tested in this experiment. All of the heat exchanger samples were made of aluminum fins and inner grooved copper tubes in a staggered arrangement. The tubes were expanded mechanically within the collared fins to minimize thermal contact resistance. The width and height of the heat exchanger samples were 305 mm and 203 mm, respectively. The fin thickness was 0.08 mm and the outside diameters of the tubes with a collar was 7.42 mm. Fin spacing of 1.3, 1.5 and 1.7 mm with two-row and three-row heat exchangers were tested. Hydrophilic coated plain-fin-and-tube and slit-fin-and-tube heat exchangers were also tested in order to investigate the effects of coating on condensate retention. Plain-fin-and-tube heat exchangers with different fin spacings were tested to validate the experiment by comparing to previous works. The specification of the slit-fin-and-tube heat exchanger used in this experiment is shown in Figure 2.4.

A counter-cross flow arrangement was used for two-row and three-row heat exchangers. The circuiting arrangement is shown graphically in Figure 2.5. The turn-around bends were enclosed with an aluminum sheet and the enclosure was filled with an expanding-foam insulation/sealant. The coolant supply and discharge were connected using quick-connect valves for easy installation. Schematic of the heat exchanger with an enclosure is provided in Figure 2.6. Geometrical information for each heat exchanger is listed in Tables 2.1 and 2.2. The differences in fin spacing, number of tube rows, coating, and the geometry of the fin (plain and slit) would allow to test the effects of each design component on the heat exchanger performance under wet conditions.

2.3 Experimental Procedures

This study used four different experiments; dry and wet heat transfer performance, the real-time condensate retention measurement, and the steady-state condensate retention measurement. A total of 22 heat exchangers were used for these four experiments. Each heat exchanger was tested over the parameter space given in Table 2.3.
2.3.1 Dry Experiment

Dry experiments were conducted to investigate the effect of the experiment condition and the effect of the moisture on heat exchanger surface by comparing the heat transfer performance under dry conditions to that of wet conditions. The blower and heater were turned on to begin the experiment. The heater was adjusted to keep the inlet temperature constant to a desired value of 35°C, and the blower was adjusted to the air face velocity of 1.5 m/s for initial set up of the experiment. The chiller was started and the flow rate of the coolant was set to the desired value. Air and coolant inlet temperatures were used to determine whether steady-state experimental conditions prevailed. The steady-state experimental condition was determined when the variations of the air and coolant inlet temperature were less than ±1°C and ±0.3°C, respectively. After the experimental condition reached a steady-state experimental condition, usually about an hour after the experiment was initially begun, the velocity adjusted to a desired velocity. The inlet temperature of 35°C was constant throughout the experiment. The coolant temperatures were varied for some dry experiments to adjust the inlet coolant temperature according to the dew point of the air which depended on the lab conditions. Typically it took about an hour to reach steady-state for each run when varying the air face velocity in 0.25 m/s interval. The average air inlet temperature and the coolant inlet temperature varied by less than ±0.05°C and ±0.3°C, respectively while the data were recorded. The air face velocity was measured with a constant-temperature anemometer and the pressure drop across the heat exchanger was measured manually.

2.3.2 Wet Experiments

The wet experiments were conducted by fixing the air dry-bulb temperature at 35°C and the air wet-bulb temperature at 24°C. The inlet air temperature, inlet dew point, and the coolant inlet temperature were used to determine whether steady-state conditions prevailed. The experimental condition had reached steady-state conditions when the air, coolant, and dew point inlet temperatures were varied by less than ±1°C, ±0.3°C, and ±0.2°C, respectively. A dummy heat exchanger was installed until the condition reached the steady-state condition. Three different experiments were conducted under wet
condition: real-time condensate measurement, steady-state condensate measurement, and heat transfer performance measurement.

A. Real-time Condensate Retention Measurement (Transient)

The real-time condensate measurement was conducted by replacing the dummy heat exchanger with the test heat exchanger after the facility reached steady-state. The electronic balance was placed under the supporter and the mass was recorded in 30-second intervals as soon as the test heat exchanger was installed. A steady-state mass of condensate was typically attained in less than 30 minutes. After the mass of condensate reached steady-state, the heat exchanger was carefully removed from the test section. A tray was placed under the heat exchanger as the heat exchanger was removed to prevent the water falling from the bottom of the heat exchanger. The mass of heat exchanger with tray was measured. The mass of the heat exchanger and the tray were measured again after they were completely dried.

B. Steady-State Condensate Retention Measurement

Steady-state condensate retention was measured using a similar procedure to the real-time condensate retention experiment. The heat exchanger was exposed under condensing conditions for at least an hour in order to make sure a steady-state mass of condensate was attained. In a regimen of experiments, face velocity was varied over a range of 0.75m/s to 2.5m/s to observe the effect of the face velocity on condensate accumulation.

C. Heat Transfer Performance under Wet Condition

The heat transfer experiment was conducted at an inlet air temperature of 35°C and an inlet dew point of 25°C. The heat exchanger was placed in the test section and sealed by the duct tape to minimize the heat lost from the gap between the specimen and the frames. It took at least three hours to reach steady-state conditions at the beginning velocity of 0.7m/s. The heat exchanger was exposed to the wet conditions for two hours for each face velocity before data were recorded. The inlet air temperature was controlled
to maintain ±1°C. The heat exchanger was tested at 0.25m/s intervals with the range of 0.75m/s to 2.5m/s.

The amount rate of water draining from the heat exchanger was measured by collecting the condensation in a graduated cylinder over a time period. The data acquisition system was started and the thermal performance data were collected in a 45-second interval. At the end of each run, thermal performance data were also recorded manually. The pressure of the lab environment was measured with a barometer and the reading was calibrated with the temperature of the lab as the manufacturer suggested.
Figure 2.1 Apparatus schematic

Figure 2.2 Velocity measurement location
Figure 2.3 Test section schematic

Figure 2.4 Specification of slit-fin
Figure 2.5 Counter-cross flow arrangement circuiting for two-row heat exchanger
Coolant in

Turn-around bends blocked off and insulated

Coolant out

Aluminum Sheets

Figure 2.6 Schematic of heat exchanger with an enclosure
Table 2.1 Heat exchanger specification

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Table 2.2 General heat exchanger information

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Table 2.3 Ranges of condition used for experimental study of sensible and latent heat transfer, air-side pressure drop, and condensate retention on the test heat exchangers.

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<td>Inlet air dry-bulb temperature</td>
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<tr>
<td>Inlet wet-bulb temperature</td>
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<tr>
<td>Face velocity</td>
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<td>Coolant supply temperature</td>
<td>Down to 0°C</td>
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CHAPTER 3 - EXPERIMENTAL RESULTS AND DISCUSSION

The air-side heat transfer performances of plain-fin-and-tube and slit-fin-and-tube heat exchangers were observed under both dry and wet conditions. The mass of the retained condensate was measured in two different ways: real-time and steady-state.

3.1 Condensate Retention Results

Two different experiments were conducted to measure the quantity of the retained condensate on both plain-fin-and-tube and slit-fin-and-tube heat exchanger surfaces. The inlet dew point and inlet air temperatures were kept at 23.9°C and 34°C, respectively.

3.1.1 Transient Condensate Retention

The real-time mass of condensation was measured in order to understand the dynamic behavior of the retained condensate. The test section was constructed so as to measure the mass of the retained condensate over a specified time period. The real-time mass of the condensate validated the steady-state mass measurement. A plain-fin-and-tube heat exchanger with a fin spacing equal to 1.5 mm was tested at two different face velocities, and the results are plotted in Figure 3.1. The time required to reach a steady-state is longer and the steady-state value is higher at lower velocities. Figure 3.1(a) shows a decrease in retained condensate at about 450 seconds; however, this decrease is due to the falling inlet dewpoint, which occurred when the PID controller adjusted the water vapor rate to accommodate the dehumidification—such behavior was due to the apparatus and was ignored. According to Korte and Jacobi [30], the quantity of retained condensate on a plain-fin-and-tube heat exchanger with wide fin spacing increases and asymptotically approaches a steady-state mass with time. However, as the fin spacing narrows, an overshoot is observed before reaching a steady-state mass of retained condensate. The mass of the retained condensate gradually increases until it reaches a maximum value, then decreases to the steady-state value as shown in Figures 3.1(b) and 3.2. After an overshoot, a minor fluctuation in the steady-state period was observed for an heat exchanger with a fin spacing of 1.3 mm, as shown in Figure 3.2(a). This fluctuation may have been due to the periodic shedding of the condensation when the gravitational
force was greater than the surface tension retaining force. The real-time mass of the retained condensate for a slit-fin-and-tube heat exchanger at different face velocities is shown in Figure 3.3. As the face velocity increases, the mass of condensate decreases. The condensate rate decreases between 400 to 800 seconds at a face velocity of 2 m/s. This decrease may be due to the humidifier recycling period and the shedding of the retained condensate. In comparison to the plain-fin-and-tube heat exchanger, there were more condensate bridges between the fins and between the slits. As a result of the shedding due to the gravitational and air forces, an oscillation in the condensate rate for is seen at higher velocities. Figure 3.3(b) shows a simultaneous increase in the mass of retained condensate and the frequency of oscillation. The steady-state mass of retained condensate for Figure 3.3(b) was determined when the variation in the mass of retained condensate got smaller.

3.1.2 Steady-State Condensate Retention

The steady-state mass of the retained condensate was measured to see the effects of heat exchanger geometry on the quantity of retained condensate. Plain-fin-and-tube heat exchangers with different geometries were tested to validate the experiment with existing work. Figure 3.4 shows that the plain-fin-and-tube heat exchanger retained more condensate if it was uncoated. A significant effect on the quantity of retained condensate was observed when the fin was treated with hydrophilic coating, causing a decrease of almost 50% in the mass of the retained condensate. The figure also clearly shows that the three-row, plain-fin-and-tube heat exchanger retains more condensate than the two-row, plain-fin-and-tube heat exchanger. The number of tube rows has a significant effect on the mass of the retained condensate, as seen when the retained condensate is divided by the total heat transfer area $A_{tot}$ (see Figure 3.5). The figure indicates that the two-row, plain-fin-and-tube heat exchanger retains more condensate per area than the three-row plain-fin-and-tube heat exchanger. This trend can be attributed to the sweeping and the shedding of the condensation. The droplet size increases as it accumulates with other droplets as moving in the direction of the air-flow force. The droplets move in the direction of the air-flow until they reach a maximum size. Some of the droplets reached the maximum size sweep away due to the air-flow force and some shed downward due to
the gravitational force. At a fixed velocity, a longer fin length will result in a greater amount of droplet sweeping and shedding due to the increase in time for the droplet to reach its maximum size. Therefore, the loss of condensation due to the sweeping and shedding is greater for the three-row heat exchanger than that of the two-row heat exchanger. However, at higher air velocities, the amount of retained condensate becomes independent of the length of the fin because most of the condensate is swept away due to the air-flow force, which explains the decrease in the discrepancy between the two-row and three-row heat exchangers seen in Figure 3.5. The amount of the sweeping of the retained condensate is observed to be dependent of the air-flow force. As the air-flow force increased, the increase in the amount of the condensate which swept away from the heat exchanger was observed at the outlet of the test section.

The mass of the retained condensate for plain-fin-and-tube heat exchangers with varying fin spacings is plotted in Figure 3.6. As fin spacing increases, the mass of retained condensate decreases, which is also observed with coated heat exchangers. Similar trends appear when the plain-fin-and-tube heat exchangers with varying fin spacings are plotted as the mass of the retained condensate per total heat transfer area, as shown in Figure 3.7. The effect of fin spacing decreases as the velocity increases for coated heat exchangers.

A comparison between the slit-fin-and-tube and the plain-fin-and-tube heat exchangers is shown in Figure 3.8. The figure shows that the slit-fin-and-tube heat exchanger retains more condensate than a plain-fin-and-tube heat exchanger with the same fin spacing. The mass of the retained condensate for a three-row heat exchanger is greater than for a two-row heat exchanger for both plain-fin-and-tube and slit-fin-and-tube heat exchangers. However, when the mass of the retained condensate divided by the total heat transfer area, the plain-fin-and-tube heat exchangers retain less condensate than the slit-fin-and-tube heat exchangers regardless of heat exchanger geometry, as shown in Figure 3.9. This trend can be explained by the differences in the maximum size of droplets that are retained on the fin surface and the amount of droplets reaching the maximum size. The maximum droplet retained on the slit-fin-and-tube heat exchanger is smaller than that on plain-fin-and-tube heat exchanger due to the slits. In addition, the air-flow barely moves the condensate on the slit-fin-and-tube heat exchanger because the
passage of condensate is blocked by the slits. As a result, the droplet cannot accumulate with other droplets and cannot develop to the maximum size. Therefore, the shedding effect on slit-fin-and-tube heat exchanger is smaller than that of the plain-fin-and-tube heat exchanger.

The effect of fin spacing on the slit-fin-and-tube heat exchanger is similar to the trends observed for the plain-fin-and-tube heat exchanger. Figure 3.10 shows how fin spacing affects the amount of the retained condensate on slit-fin-and-tube heat exchangers. Heat exchangers with wider fin spacing retain less condensate than those with closer fin spacing. However, a counter result occurs when the retained condensate is divided by the total area, as shown in Figure 3.11. This result may be due to the condensate bridge, which forms when the condensate drops accumulate as they are shed downward. The mechanism of the condensate bridge is shown in Figure 3.12. As the fin spacing narrows, more condensate bridges form, and a smaller amount of retained condensate is observed. Figure 3.11 also shows that as the velocity increases, the two slit-fin-and-tube heat exchangers with different fin spacing have trends that appear to merge at a certain quantity of condensate. This trend is due to the condensate sweeping: at higher velocities, the condensate bridges are swept away due to air-flow forces.

3.2 Condensate Accumulation Effects on Air-Side Thermal Performance of Heat Exchanger Surface

In order to understand the effects of condensate accumulation on the air-side heat transfer performance, the heat transfer coefficient and pressure drop were measured for various heat exchanger geometries. The effective air-side heat transfer coefficient was determined by ARI Standard 410 and the wet fin efficiencies were determined by applying the air-side heat transfer coefficient, Schmidt fin efficiency techniques and the sector method. The specimen was assumed to be fully wet when the surface temperature of the heat exchanger was below the dew point of the air stream at all points. It was assumed to be dry when the ratio of sensible heat \( Q_{\text{sens}} \) to total heat \( Q_{\text{tot}} \) was greater than or equal to 0.95, per ARI standard 410 [32]. A detailed presentation of the data reduction procedure is presented in Appendix A.
3.2.1 Air-Side Heat Transfer Performance Results

In order to determine the effects of condensate accumulation on the heat exchanger surface, both dry and wet experiments were conducted for plain-fin-and-tube and slit-fin-and-tube heat exchangers. The dry experiments provided the baseline for the wet experiments by comparing the differences in their heat transfer coefficients and pressure drops.

Figure 3.13 presents the sensible air-side Nusselt number versus Reynolds number under dry and wet conditions for plain-fin-and-tube heat exchangers with a fin spacing of 1.5 mm. The figure shows an increase in the Nusselt number with an increase in the Reynolds number. The Nusselt numbers are about 40% higher for dry conditions than for the wet experiments. The coating has very little effect on the Nusselt number; under wet conditions, the coated heat exchanger has a slightly lower sensible Nusselt number than that of the uncoated heat exchanger, and for dry conditions, the coated heat exchanger has a slightly higher sensible Nusselt number than that of the uncoated heat exchanger. These coating effects are within the experimental uncertainty. The effect of the number of tube rows is presented in Figure 3.14. The sensible air-side Nusselt number was higher for the two-row heat exchangers than that of the three-row heat exchangers; the differences were again within the experimental uncertainty, but the trends are clear.

Slit-fin-and-tube heat exchangers show trends similar to the plain-fin-and-tube heat exchangers. Figure 3.15 presents results of a slit-fin-and-tube heat exchanger with a 1.7 mm fin spacing. The figure shows an increase in the Nusselt number with an increase in the Reynolds number, and the Nusselt number was higher when the experiment was conducted under dry conditions. The difference between dry- and wet-surface increases as the Reynolds number based on tube collar diameter increases. McQuiston [9] observed an increase in sensible heat transfer performance for wider fin spacing under wet conditions and a decrease when the fin spacing becomes closer. Korte and Jacobi [30] observed a higher air-side sensible Nusselt number under wet conditions than under dry conditions for fin spacing of 4 fpi. However, when a heat exchanger with 8 fpi was tested, the Nusselt number under wet conditions was sometimes higher and sometimes lower than the corresponding dry values. Mirth and Ramadhyani [26] observed the heat transfer performance of the wavy-fin heat exchanger with fin spacing between 1.47 to
3.05 mm. They reported that the air-side sensible Nusselt number of the wavy-fin heat exchanger with fin spacing less than 2.11 mm was lower for wet conditions compared to dry surface data. These works indicate that the heat transfer performance under wet conditions is dependent on the fin spacing. When the fin spacing gets closer, the Nusselt number under wet conditions tends to be lower than the dry condition value. For the slit-fin-and-tube heat exchanger, the coating effect on Nusselt number shows a trend similar to that of the plain-fin-and-tube heat exchanger as shown in Figure 3.15; the coated heat exchanger has a lower sensible Nusselt number than that of the uncoated heat exchanger under wet conditions. However, for dry conditions, the coated heat exchanger has a higher sensible Nusselt number than that of the uncoated heat exchanger. These effects are within experimental uncertainty, but the trends are consistent. The effect of the number of tube rows for slit-fin-and-tube heat exchangers shows the same trend as that of the plain-fin-and-tube heat exchangers. The two-row heat exchanger has a Nusselt number as much as 25% higher than the three-row heat exchanger in wet conditions, but slightly lower for dry conditions, as shown in Figure 3.16. For dry conditions, the difference is again within the experimental uncertainty. For wet conditions, the trend shown in Figure 3.16 can be attributed by the condensation effect on the slit-fin surface and the high heat transfer coefficient associated with developing flow in the first row [26]. The condensate accumulates more on the two-row, slit-fin-and-tube heat exchanger than the three-row, slit-fin-and-tube heat exchanger due to the shedding and sweeping effects. The condensate leads to the enhanced heat transfer for the two-row heat exchanger due to the surface roughness. Figures 3.17 and 3.18 show the effect of fin spacing under dry and wet conditions for slit-fin-and-tube heat exchangers, respectively. For dry conditions, heat exchangers with wider fin spacing tend to have a higher Nusselt number than heat exchangers with closer fin spacing, except for the heat exchanger with a fin spacing of 1.3 mm. A rapid increase in the Nusselt number for the heat exchanger with 1.3 mm spacing is observed as the Reynolds number increases. The results of the wet experiments show a slightly higher Nusselt number for the heat exchanger with a 1.3 mm fin spacing than that for the heat exchanger with a 1.5 mm fin spacing when the Reynolds number was below 1500. At Reynolds numbers above 1500, a different result is observed: the heat exchanger with a 1.3 mm fin spacing has a slightly lower Nusselt
number than that of the heat exchanger with a 1.5 mm fin spacing. This trend may be explained by the condensate bridge formed between the adjacent fins of heat exchangers. Bridges are formed between the adjacent fins and between the slits. Quantity of the condensate bridge increases as the fin spacing decreases. The heat exchangers with 1.3 mm and 1.5 mm fin spacing can form bridges more easily than the heat exchanger with a 1.7 mm fin spacing. The condensation on the fin surface changes the geometry of the heat exchangers, and the Nusselt number becomes independent of the fin spacing. As the air velocity increased, the bridge formed between fins can be removed by air forces. Heat exchangers with wider fin spacing show a slightly higher Nusselt number than those with closer fin spacing. These trends suggest that the heat transfer enhancement depends on the fin spacing as well as the condensate formed on fin surface for slit-fin-and-tube heat exchangers.

Figures 3.19 and 3.20 present the effect of fin geometry on the heat transfer coefficient. Both figures show a higher Nusselt number for the slit-fin-and-tube heat exchangers than the plain-fin-and-tube heat exchangers. Coating, however, appears to affect the heat transfer performance. The coated slit-fin-and-tube heat exchanger shows a slightly higher Nusselt number than that of the plain-fin-and-tube heat exchanger for both conditions, although the difference is within experimental uncertainty. The difference in the Nusselt number for both conditions are due to the difference in fin geometry. A thermal boundary layer developed on plain-fin surface becomes thicker away from the leading edge of the fin, and decreases the heat transfer coefficient. In order to minimize the thickness of the thermal boundary layer, the slits are raised. Slits create a new thermal boundary layer, and the high heat transfer coefficient at the leading edge can be maintained throughout the fin. Kang et al. [33] observed the effect of slit location and reported that the heat exchanger with a slit-fin has a higher heat transfer coefficient than that of plain-fin-and-tube heat exchanger.

3.2.2 Sensible $j$ and Friction Factors (Wet and Dry)

The heat transfer performance for various geometries of the heat exchangers are presented in terms of the sensible $j$ and friction factors. $j$ and $f$ factors are calculated for dry and wet conditions using equations 3.1 and 3.2.
\[ j = St_{air} Pr_{air}^{\frac{1}{2}} \]  

\[ f = \frac{2 \Delta P_{\text{air}} \rho_{\text{air}}}{G_{\text{air}}^2 \left( A_{\text{min}} \right)^2 \left( 1 + \sigma^2 \left( \frac{\rho_{\text{air, in}}}{\rho_{\text{air, out}}} \right) - 1 \right) \left( \frac{A_{\text{tot}}}{A_{\text{tot}}} \right) \left( \frac{\rho_{\text{air}}}{\rho_{\text{air, in}}} \right)} \]  

A. Plain-Fin-and-Tube Heat Exchanger

A two-row, uncoated, plain-fin-and-tube heat exchanger with a 1.5mm fin spacing was tested under dry and wet conditions, and the results are presented in Figure 3.21. The pressure drop across the heat exchanger is smaller and the air-side heat transfer coefficient is higher when the heat exchanger is tested under dry conditions. Most existing works report higher \( f \) factors for wet conditions. Guillory and McQuiston [5] observed a 30% increase in the friction factor under wet conditions compared to the dry values. When the heat exchanger is tested under condensing conditions, the condensate accumulates on the heat exchanger surface. The condensate increases the pressure drop across the heat exchanger, resulting in an increase in the \( f \) factor. The degradation of the \( j \) factor under wet conditions may be due to the condensate retention. According to Jacobi and Goldschmidt [34], condensate retention occurs with a deleterious effect of the heat transfer at lower Reynolds numbers. Wang et al. [14] and Uv and Sonju [35] support this argument for the degradation of \( j \) factors. Their data showed that at the Reynolds number based on tube collar diameter less than 2000, the \( j \) factor for wet conditions was lower than that of dry conditions. Sensible \( j \) and \( f \) factors for heat exchangers with different fin spacings are plotted in Figure 3.22. Under wet conditions, the \( j \) factors are independent of fin spacing. The \( f \) factors of the 1.3 mm fin spacing heat exchanger are lower than that of the 1.5 mm fin spacing heat exchanger, but higher than that of the 1.7 mm fin spacing heat exchanger. Figures 3.23 and 3.24 present the effect of the number of tube rows for a plain-fin-and-tube heat exchanger. The heat exchanger with two tube rows have slightly higher \( j \) and \( f \) factors under dry conditions, but the differences are within experimental uncertainty when the experiment is conducted under wet conditions. Under wet conditions, the data suggest the friction factors are independent of the number of tube rows while the \( j \) factors for the two-row heat exchanger are slightly higher than that of the three-row heat exchanger. This small effect may be caused by retained condensate on the
fin surface acting as a vortex generator, causing the higher heat transfer coefficient. Since the shedding effect of condensate is greater for three-row heat exchanger, the amount of condensate is smaller. This trend causes lower heat transfer coefficients than the two-row heat exchanger. The sensible $j$ and $f$ factors for plain-fin-and-tube heat exchanger under dry condition are independent of coating, but the $f$ factors for the uncoated heat exchanger are higher for wet conditions as shown in Figures 3.25 and 3.26. The retained condensate blocks the passage of the air-flow, results the higher pressure drop across the heat exchanger.

**B. Slit-Fin-and-Tube Heat Exchanger**

The sensible $j$ and $f$ factors were calculated for slit-fin-and-tube heat exchangers and the results are plotted in Figure 3.27. The figure shows that the $j$ and $f$ factors decrease as the Reynolds number increase for both conditions. The same trends as plain-fin-and-tube heat exchanger are observed. The $f$ factor is higher for wet conditions compared to that of dry conditions, but the $j$ factor under wet conditions is slightly lower than that of dry conditions. Figure 3.28 shows the effect of fin spacing on sensible $j$ and $f$ factors when the experiment is conducted under wet condition. The sensible $j$ factors for slit-fin-and-tube heat exchangers under wet conditions depend on the fin spacing. The heat exchanger with 1.7 mm has higher $j$ factors than the heat exchangers with 1.3 and 1.5 mm fin spacings. The $f$ factors also depends on the fin spacing as the Reynolds number increases; the heat exchanger with 1.3 mm fin spacing has slightly higher $f$ factors than other heat exchangers with different fin spacing. The $f$ factors at lower Reynolds numbers are independent of fin spacing. These effects are almost within experimental uncertainty, but the trends are consistent. The effect of the number of tube rows for the slit-fin-and-tube heat exchanger is also observed under dry and wet conditions. Under dry conditions, the $f$ factors are higher for the three-row heat exchanger while the $j$ factor is higher for the two-row heat exchanger as shown in Figure 3.29. This trend, however, changes when the experiment is conducted under wet conditions. As shown in Figure 3.30, the $f$ factor for the two-row heat exchanger is higher than that of the three-row heat exchanger. The effect of the number of tube rows for the $f$ factor variation increases as the Reynolds number increases. This $f$ factor variation is due to the
shedding and sweeping effects of the condensation. At lower Reynolds numbers, the airflow force stimulating the shedding and sweeping effects of the condensation is small. Therefore, the condensate stays on both heat exchanger surfaces as either drops or bridges. However, at higher Reynolds numbers, the airflow force became larger and shed the condensate on the fin surfaces. Since the shedding and sweeping of condensate for the three-row heat exchanger is significantly greater than the two-row heat exchanger, a smaller amount of condensate is retained on the three-row heat exchanger. Therefore, the pressure drop for the three-row heat exchanger becomes lower than that of two-row heat exchanger, as did the $f$ factor. The effect of the number of tube rows for the $j$ factors are inconclusive; the two-row heat exchanger showed the degradation of $j$ factors for three-row heat exchanger at lower Reynolds numbers and an enhancement is observed at higher Reynolds numbers compared to the three-row heat exchanger. This trend is within the experimental uncertainty. When the slit-fin-and-tube heat exchanger is tested under dry conditions, both the $j$ and $f$ factors are independent of coating, as shown in Figure 3.31. However, when the slit-fin-and-tube heat exchanger is tested under wet condition, the $f$ factor for the uncoated heat exchanger tends to be higher than that of the coated heat exchanger, as clearly shown in Figure 3.32. The figure shows that the coating on the fin surface affects only the friction factors. This result can be explained by the condensation effect on the pressure drop across the heat exchanger. The condensation is formed between the fins and on the surface of the fins, and blocks the passage of the air stream when the heat exchangers are under wet conditions. By coating the fin surface, the wettability is increased and the retained condensate is reduced, thus accounting for the pressure drop. The coating helps to reduce the pressure drop across the heat exchanger significantly when the heat exchanger is fully wet. The coating on the fin surface, however, has little or no effect on heat transfer performance. Hong [29] found there was a convergence in the wettability of the coatings after approximately 1,000 wetting cycles, and the contact angles of both coated and uncoated heat exchangers were almost the same after a certain amount of the wetting cycles. Therefore, the decrease in pressure drop in this experiment might change if the coated heat exchanger were exposed over 1,000 wetting cycles. Samsung Electronics Co., LTD presents contact angles after 500 wetting cycles as shown in Table 3.1. Figure 3.32 also shows a slightly higher $f$ factor for the
uncoated heat exchanger at lower Reynolds numbers. However, as the Reynolds number increases, the discrepancy of the \( j \) factor between coated and uncoated heat exchanger decreases. This discrepancy in the \( j \) factor due to the coating is caused by the condensation effect. Uncoated heat exchangers tend to retain more condensate than that of coated heat exchangers. The condensate increases the surface roughness, and therefore the \( j \) factor increases. However, at higher Reynolds numbers, the air-flow force removes the retained condensate, and decreases the \( j \) factor. The \( j \) factor then becomes independent of coating as the Reynolds number increases.

C. Effect of Fin Geometry

The plain-fin-and-tube heat and slit-fin-and-tube heat exchangers are compared to one another under dry and wet conditions in Figures 3.33 and 3.34. The slit-fin-and-tube heat exchanger has slightly higher \( j \) and \( f \) factors than the plain-fin-and-tube heat exchanger under dry condition. When the experiments are conducted under wet conditions, the \( j \) and \( f \) factors strongly depend on the fin geometry as the fin spacing widens. As Figure 3.35 shows, the slit-fin-and-tube heat exchanger with a 1.7 mm fin spacing has higher \( f \) and \( j \) factors than the plain-fin-and-tube heat exchanger with the same fin spacing.

3.3 Contact Angle Measurement

The contact angle is measured by a goniometer and measured values are shown in the Table 3.2. The coated fin stock show much smaller advancing and receding angles compared to the uncoated fin stock. The results of contact angle measurements will be used in developing the model of condensate retention.

3.3.1 Methods of Measuring Contact Angles

The contact angles are measured using similar techniques applied by Korte and Jacobi [30]. Several different sides and locations of the fin surfaces are observed to get the mean value of the contact angles. Figure 3.37 explains the two different techniques used in this experiment. The first technique (a) is to measure the advancing and receding contact angles by feeding or withdrawing the liquid using a syringe. The second
technique (b) is to measure the contact angles by tilting the solid surface and rotating the fin stock specimen to 90 degrees. The angles formed as the drop begins to move down the solid are advancing and receding contact angles. For both techniques, the vapor pressure is controlled by placing a cover over the specimen. The uncoated fin stock has average advancing and receding contact angles of 87.5° and 40.42° while the coated fin has advancing and receding contact angles of 9.64° and 4.25°. The mean values of advancing and receding contact angle will be used to determine the volume of condensate droplet and the maximum diameter of droplet. A detailed method of determining the volume and the diameter of droplet is discussed in Chapter 4.

3.4 Uncertainty

The uncertainty in the electronic balance was 0.5%, and could be negligible compared to other possible errors. The steady-state values from the real-time retention experiment and heat exchanger removal after prolonged exposure to condensing condition agreed to within 20%. This uncertainty might be due to the evaporation during the weighting process and possible human errors such as spilling some of the condensate when removing the heat exchanger. Uncertainty in air-side sensible Nusselt number was determined to be 11.8% along with air-side heat transfer coefficient of 11%. Uncertainty in Reynolds number based on collar diameter was approximately 7% and the $j$ and $f$ factor had an uncertainty of 13.13% and 12%, respectively. More detailed uncertainty analysis in the results of condensate retention and heat transfer experiments are discussed in Appendix C.
Figure 3.1 Real-time retention plots (Plain, $f_s=1.5$mm, 2rows, Uncoated)
Inlet air temperature $\sim 34^\circ$C, inlet dew point temperature $\sim 23.9^\circ$C
(a) $V_{\text{max}} = 1$ m/s (b) $V_{\text{max}} = 2$ m/s

Figure 3.2 Real-time retention plots, $V_{\text{max}} = 2$ m/s (Plain, 2rows, Uncoated)
Inlet air temperature $\sim 34^\circ$C, inlet dew point temperature $\sim 23.9^\circ$C
(a) $f_s = 1.3$mm (b) $f_s = 1.7$mm
Figure 3.3 Real-time retention plots (Slit, $f_i = 1.7$mm, 2 rows, Uncoated)
Inlet air temperature ~34°C, inlet dew point temperature ~23.9°C
(a) $V_{\text{max}} = 1.5$ m/s (b) $V_{\text{max}} = 2$ m/s
Figure 3.4 Steady-state condensate retention (Plain, $f_e = 1.5\text{mm}$)

Figure 3.5 Steady-state condensate retention $/A_{tot}$ (Plain, $f_e = 1.5\text{mm}$)
Figure 3.6 Steady-state condensate retention (Plain, 2 rows)

Figure 3.7 Steady-state condensate retention $/A_{tot}$ (Plain, 2 rows)
Figure 3.8 Steady-state condensate retention (Uncoated)

Figure 3.9 Steady-state condensate retention / A_t (Uncoated)
Figure 3.10 Steady-state condensate retention (Slit, 2rows, Uncoated)

Figure 3.11 Steady-state condensate retention /A_{tot} (Slit, 2rows, Uncoated)
Figure 3.12 Mechanism of condensate bridge (Korte and Jacobi, 1997)
Figure 3.13 Air-side sensible heat transfer results (Plain, $f_s=1.5\text{mm}$, 2rows)

Figure 3.14 Air-side sensible heat transfer results (Plain, $f_s=1.5\text{mm}$, Uncoated)
Figure 3.15 Air-side sensible heat transfer results (Slit, $f_s=1.3\text{mm}$, 2 rows)

Figure 3.16 Air-side sensible heat transfer results (Slit, $f_s=1.3\text{mm}$, Uncoated)
Figure 3.17 Air-side sensible heat transfer results under wet conditions (Slit, 2rows, Uncoated)

Figure 3.18 Air-side sensible heat transfer results under dry conditions (Slit, 2rows, Coated)
Figure 3.19 Air-side sensible heat transfer results (2rows, Uncoated)

Figure 3.20 Air-side sensible heat transfer results (2rows, Coated)
Figure 3.21 Sensible $j$ and $f$ factors (Plain, $f_5=1.5\text{mm}$, 2rows, Uncoated)

Figure 3.22 Effect of fin spacing under wet condition (Plain, 2rows, Uncoated)
Figure 3.23 Effect of number of tube rows under dry condition (Plain, $d_s=1.5$mm, Uncoated)

Figure 3.24 Effect of number of tube rows under wet condition (Plain, $d_s=1.5$mm, Uncoated)
Figure 3.25 Effect of coating under dry condition (Plain, $f_5=1.5$mm, 2rows)

Figure 3.26 Effect of coating under wet condition (Plain, $f_5=1.5$mm, 2rows)
Figure 3.27 Sensible $j$ and $f$ factors (Slit, $h_s=1.7$mm, 2rows, Uncoated)

Figure 3.28 Effect of fin spacing under wet condition (Slit, 2rows, Uncoated)
Figure 3.29 Effect of number of tube rows under dry condition
(Slit, $f_s=1.7\text{mm}$, Uncoated)

Figure 3.30 Effect of number of tube rows under wet condition
(Slit, $f_s=1.7\text{mm}$, Uncoated)
Figure 3.31 Effect of coating under dry condition (Slit, $f_s$=1.3mm, 2rows)

Figure 3.32 Effect of coating under wet condition (Slit, $f_s$=1.7mm, 2rows)
Figure 3.33 Effect of fin geometry under dry condition ($f_s=1.5\text{mm}$, 2 rows, Uncoated)

Figure 3.34 Effect of fin geometry under wet condition ($f_s=1.5\text{mm}$, 2 rows, Uncoated)
Figure 3.35 Effect of fin geometry under wet condition ($f_s=1.7\text{mm}$, 2 rows, Uncoated)

Figure 3.36 Effect of fin geometry under wet condition ($f_s=1.3\text{mm}$, 2 rows, Uncoated)
Table 3.1 Contact angles measurement before and after 500 wetting cycles (provided by Samsung Electronics Co., LTD)

<table>
<thead>
<tr>
<th>Coating</th>
<th>Initial Value (degrees, °)</th>
<th>After Cleaning (degrees, °)</th>
<th>After 500 Wetting Cycles (degrees, °)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uncoated</td>
<td>92</td>
<td>85</td>
<td>70</td>
</tr>
<tr>
<td>Coated</td>
<td>10</td>
<td>8</td>
<td>25</td>
</tr>
</tbody>
</table>

Table 3.2 Contact angle measurement by two different techniques
Techniques: (1) feeding and withdrawing liquid droplet (2) rotating sample to 90°

<table>
<thead>
<tr>
<th>Coating</th>
<th>Technique</th>
<th>Condition</th>
<th>$\theta_a$ (degrees, °)</th>
<th>Standard Deviation</th>
<th>$\theta_r$ (degrees, °)</th>
<th>Standard Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uncoated</td>
<td>1</td>
<td>New</td>
<td>86.50</td>
<td>8.92</td>
<td>41.83</td>
<td>6.23</td>
</tr>
<tr>
<td>Uncoated</td>
<td>2</td>
<td>New</td>
<td>88.50</td>
<td>4.10</td>
<td>39.00</td>
<td>3.75</td>
</tr>
<tr>
<td>Coated</td>
<td>1</td>
<td>New</td>
<td>9.33</td>
<td>0.82</td>
<td>4.67</td>
<td>0.82</td>
</tr>
<tr>
<td>Coated</td>
<td>2</td>
<td>New</td>
<td>9.94</td>
<td>9.94</td>
<td>3.82</td>
<td>16.24</td>
</tr>
</tbody>
</table>

Figure 3.37 Techniques for measuring advancing and receding contact angles
Techniques: (1) feeding and withdrawing liquid droplet (2) rotating sample to 90°
CHAPTER 4 – CORRELATION DEVELOPMENT AND RETENTION MODELING

This chapter presents $j$ and $f$ factor correlations for plain-fin-and-tube and slit-fin-and-tube heat exchangers under dry and wet conditions. The ranges of parameters for the correlations are discussed here. A simple model of retained condensate for slit-fin-and-tube heat exchanger is also developed and discussed in this chapter.

4.1 Experimental Data Comparison

In order to validate the experimental data, the dry data of the plain-fin-and-tube heat exchanger are compared to correlations developed in the extant literature. Figure 4.1 shows a comparison between the experimental data and the correlations proposed by Gray and Webb [23]. Their correlations are valid within $2400 < \text{Re}_{\text{D}} < 24700$, $1.97 < \text{St}_{\text{D}} < 2.55$, $1.7 < \text{St}_{\text{T}} < 2.58$, $0.08 < f_{\text{D}} < 0.64$, and $1 < n_{\text{T}} < 8$. Figure 4.1 shows that the correlations predict the experimental data of only approximately 60% of the $f$ factors and 40% of the $j$ factors within ±20%. The discrepancy between the experimental data and the predicted data may be due to the differences in the heat exchanger geometry. The dimensionless geometric parameters of the tube bank and the range of the Reynolds number used in this experiment are not within the valid ranges of the correlations, and the fin thickness is relatively thinner than that used in the correlations. In addition, the heat exchanger Gray and Webb used had no collar around the tubes. Therefore, the correlations were developed without accounting for the fin thickness and the tube collar. Since the fin thickness and tube collars affect entrance and exit pressure drops significantly, predicting the experimental data with these correlations may not be appropriate. Furthermore, the original data for Gray and Webb's correlation were based on four tube-row heat exchangers. In order to take into account the tube-row effect, Gray and Webb developed an empirical correlation using Rich's data [11]. Therefore, the effect of tube-row may be accounted for inaccurately. Figure 4.2 compares the wet experimental data with the correlations developed by Wang et al. [14]. The ranges of the parameters for the correlations are $300 < \text{Re}_{\text{Dcoll}} < 5500$, $1.82 < f_{\text{D}} < 3.2$, and $2 < n_{\text{T}} < 6$. Although the uncertainties of the correlation are approximately ±10% for both $j$ and $f$ factors, the $f$ factor correlation underpredicts and the $j$ factor correlation overpredicts the
present experimental data. These discrepancies are caused by the differences in heat exchanger geometry and the effect of condensate on the fin surface. Dropwise condensation is formed easily on the heat exchanger with wider fin spacing. Since the dropwise condensation does not reduce the air-flow significantly, pressure drops are small. However, for narrower fin spacing, condensation tends to form as a bridge. The condensate bridge blocks the passage of the air-flow and increases the heat exchanger pressure drop and, consequently the $f$ factor. The $f$ factor correlation proposed by Wang et al. was developed based on heat exchangers with relatively wider fin spacing. Thus, the $f$ factor correlation may underpredict the $f$ factors for the heat exchangers with fin spacings less than the valid range given by Wang et al. As shown in Figure 4.2 (a), the $f$ factors of the heat exchanger with 1.7 mm fin spacing are predicted within the uncertainty of the correlation, while the $f$ factors of the heat exchangers with 1.5 mm fin spacing are underpredicted. The $f$ factors for the coated heat exchanger with two tube rows are also predicted within the uncertainty. The coating removes the condensate accumulated on the heat exchanger surface, which helps the air flow through the heat exchanger without any obstacles.

4.2 $j$ and $f$ Factor Correlations

4.2.1 Introduction

There have been several $j$ and $f$ factor correlations developed for plain-fin-and-tube heat exchangers in the literature. However, those correlations can only predict the heat exchangers within certain ranges of variables. Therefore, comparison between the experimental data and the existing correlations may not be appropriate. Furthermore, there are only a few studies presenting $j$ and $f$ factor correlations for slit-fin-and-tube heat exchangers. In this section, the correlations of the $j$ and $f$ factors for both plain-fin-and-tube and slit-fin-and-tube heat exchangers are developed. The $j$ and $f$ factor correlations for plain-fin-and-tube and slit-fin-and-tube heat exchangers from other studies are given in Table 4.1 for reference.
4.2.2 Developing $j$ and $f$ Factor Correlations

$j$ and $f$ factor correlations are developed by accounting for the dimensionless geometric parameters for the heat exchanger and its tube bank as well as the dimensionless flow parameters of the air. The proposed equation for $j$ and $f$ factor correlations is given by Equation 4.1.

$$f, j = C(Re_{Dcoll})^a \left( \frac{f_s}{D_{Coll}} \right)^b \left( \frac{S_{t}N_{Z}}{D_{Coll}} \right)^c$$

(4.1)

where $C$, $a$, $b$, and $c$ are constants.

The dimensionless parameters for the correlations are determined by using Buckingham-$\Pi$ Theorem. The details of Buckingham-$\Pi$ Theorem are described in Appendix B. The values of the constants are determined by the program written in Engineering Equation Solver (EES). The root mean square (rms) is minimized with respect to the constants, $C$, $a$, $b$, and $c$ using variable metric method within the EES program in order to develop the best fit multiple regression. The "rms" is calculated by Equation 4.2.

$$rms = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (X_{Correlated,n} - X_{Experimental,n})^2}$$

(4.2)

where $n$ is number of data points.

The $j$ and $f$ factor correlations are developed for five different conditions due to the differences in fin geometry, coating, and experimental conditions. The correlations are presented in Table 4.2 and the ranges of the parameters are shown in Table 4.3.

4.2.3 Confidence in $j$ and $f$ Factor Correlations

Figure 4.3 shows that the correlations for the uncoated plain-fin-and-tube heat exchanger under wet conditions can describe approximately 80% of the $f$ factor data within 20% and 96% of the $j$ factor data within 10%. The comparison between the experimental data and the correlations for the uncoated slit-fin-and-tube heat exchanger
under dry conditions is presented in Figure 4.4. The correlations predict approximately 82% of the $f$ factor data within 20% and 88% of the $j$ factor data within 15%. Uncertainties of the correlation for each heat exchanger condition are shown in Table 4.4 and the figures for other conditions are presented in Appendix E. The success of these correlations indicates excellent repeatability in the experimental results for a wide range of experimental conditions and geometry.

Figure 4.5 compares the present $j$ and $f$ factor correlations for uncoated slit-fin-and-tube heat exchanger to the correlations developed by Nakayama and Xu [21]. The specimen used for this comparison has two tube rows with 1.7 mm fin spacing. The ranges of parameters for Nakayama and Xu's correlation are $250 \leq \text{Re}_D \leq 3000$, $0.15 \text{mm} \leq \delta \leq 0.2 \text{mm}$ and $1.8 \text{mm} \leq \delta \leq 2.5 \text{mm}$. Nakayama and Xu took into account the slit-fin configuration for their slit-fin-and-tube heat exchanger geometry. The number of slits for each slit set, the breadth of a slit in the direction of air-flow, and the width of the slit were accounted for the correlations. The figure shows that there is a small discrepancy in $f$ factors. The $f$ factor correlation developed by Nakayama and Xu is slightly less than the present correlations. In contrast, the discrepancy in $j$ factors increases as the Reynolds number increases. The discrepancies between the two correlations are due to the difference in heat exchanger geometry as well as the slit configuration. In addition, the difference in calculating fin efficiency may also be one of the reasons for the discrepancies. Furthermore, the discrepancies may have been because the correlations proposed by Nakayama and Xu are based on the heat transfer modeling and few experimental data. Nevertheless, the overall agreement with this independent study is excellent and it verifies the validity of the new results.

4.3 Developing Simple Condensate Retention Model For Slit-Fin-And-Tube Heat Exchanger

In order to develop a simple model for the condensate retention, a slit-fin stock is tested under a condensing condition. A photograph of the slit-fin stock with condensation is shown in Figure 4.6. This photograph is used to determine the distribution of the droplets on fin surface as well as the diameters of droplets.
4.3.1 Retained Condensate Geometry

Two major retained condensate geometries are considered to predict the total quantity of the condensation on slit-fin-and-tube heat exchanger. One is the droplets adhering to the fin and tube surfaces, and the other is the bridges between adjacent fins at fin-tube junctions. The equations for calculating the volume of droplet and the bridge are same as the equations that Korte and Jacobi [30] used for their condensate modeling. Figure 4.7 shows a droplet adhering to a surface at an inclination angle of $\alpha$ and Figure 4.8 shows a bridge between the adjacent fins at fin-tube junction. The volume of each droplet is approximated using Equation 4.3 with known diameter of the droplet and the advancing and receding contact angles.

\[
V_{\text{drop}} = \frac{\pi D_{\text{drop}}^3}{24} \left( \frac{2 - 3\cos\theta_M + \cos^3\theta_M}{\sin^3\theta_M} \right) \tag{4.3}
\]

where

\[
\theta_M = \frac{\theta_A + \theta_R}{2}, \quad D_{\text{drop}} = \frac{4A_{\text{drop}}}{\pi}
\]

$A_{\text{drop}}$ is obtained from Scion Image analysis

The volume of each fin-tube bridge is determined by Equation 4.4.

\[
V_{\text{bridge}} = L_{\text{max}} (l \cdot f_s) - A_1 f_s + A_2 l \tag{4.4}
\]

where $l =$ length of the condensate bridge obtained from Scion Image analysis, and
\[ A_1 = \left( \pi R_1^2 \left( \frac{\pi}{2} - \theta_R \right) \right) - \frac{l}{2} \left[ R_1^2 - \left( \frac{l}{2} \right)^2 \right]^{\frac{1}{2}} \]

\[ A_2 = \left( \pi R_2^2 \left( \frac{\theta_A - \pi}{2} \right) \right) + f_2 \left[ R_2^2 - \left( \frac{f_2}{2} \right)^2 \right]^{\frac{1}{2}} \]

\[ R_1 = \frac{D_{coil}}{2} \]

\[ R_2 = \frac{f_s}{2 \sin \left( \frac{\theta_A - \pi}{2} \right)} \]

The maximum length of the bridge is calculated by

\[ L_{\text{max}} = \frac{2f_s \cos \theta_R + 2f_A \cos (\pi - \theta_A) + \rho_l g A_1 - \rho_l g l A_2}{\rho_l g f_s} \]  

where \( \gamma \) is the surface tension of water at temperature, \( T \). The surface tension of water can be calculated using the following equation, which is valid over the temperature range \( 10^\circ\text{C} \leq T \leq 100^\circ\text{C} \).

\[ \gamma (\text{mN/m}) = 75.83 - 0.1477T \]

### 4.3.2 Proposed Retention Model

To provide a simple equation that predicts the mass of retained condensate on slit-fin-and-tube heat exchanger, the following assumptions are made to simplify the procedure of modeling an equation.

- The slit-fin-and-tube heat exchanger is fully wet.
- There are no bridges between the fins adjacent to each other except the fins at the fin-tube junction.
- Droplet distribution on the fin surface is independent of the droplets on the adjacent fin surface.
A condensate retention model is proposed as shown in Equation 4.7.

\[ m_{\text{cond}} = \left( \sum_{n=1}^{N} \frac{1}{A_{\text{drop},n} V_{\text{drop},n} \rho_l} \right) A_{\text{tot}} + \left( V_{\text{bridge}} \rho_l \right)^* N_{\text{v}} N_f \]  \hspace{1cm} (4.7)

where \( A_{\text{drop}} \) is obtained from Scion Image Analysis and \( N \) is number of droplet counted using Scion Image.

Equation 4.7 consists of two parts; mass of retained condensate based on the droplets and the mass of retained condensate based on the bridges.

4.3.3 Modeling Procedure

The modeling technique applied is similar to that which Korte and Jacobi [30] proposed. The important difference in the technique is that the droplet distributions are analyzed using Scion Image program. Scion Image program calculates the area of each droplet. By knowing the area for each droplet, the equivalent diameter of the droplet and the \( \Delta N_D \) can be calculated.

A slit-fin stock was divided into small zones as shown in Figure 4.9. Each zone contains a set of slits and a tube row. It is assumed that each zone retains exactly the same amount of condensate to simplify the model. A zone located in the middle of the tested slit-fin stock, which shows a clear image of droplet, is selected and the enlargement of this photograph is shown in Figure 4.10(a). Since the photograph does not clearly show each droplet on the fin stock for the use of Scion Image program, the droplet is colored in black as shown in Figure 4.10(b). Scion Image analyzed Figure 4.10(b) and the area of each droplet as well as total area covered by droplets were calculated. The length of the bridge, \( l \), at the fin-tube junction is assumed to be the same as the value of the major diameter for the bridge under fin-tube junction. The major diameter of the bridge is also given by Scion Image. An EES program is used to calculate the mass of retained condensate on the slit-fin-and-tube heat exchanger using Equations 4.3 to 4.7.

4.3.4 Retention Modeling Result

The proposed model is used to predict the quantity of retained condensate. According to the condensate retention experiment, the mass of retained condensate
decreases as the Reynolds number increases. Therefore, the maximum mass of retained condensate for each heat exchanger can be measured at 0 m/s face velocity. The maximum quantity of retained condensate for each slit-fin-and-tube heat exchanger is predicted. Each heat exchanger is different in number of tube rows and fin spacing. The predicted values are shown in Table 4.5. The measured mass of retained condensate is based on the range of the air velocity of 0.75 m/s to 2.5 m/s. The uncoated heat exchanger has advancing and receding contact angles of 87.5° and 40.42°, respectively. As shown in Table 4.5, the predicted mass of retained condensate is slightly lower than the measured mass of retained condensate for two-row slit-fin-and-tube heat exchanger. However, the results for the three-row heat exchangers are higher than the measured values of two-row heat exchanger. This discrepancy between the predicted and measured values is due to the assumption made to simplify the retention modeling procedure. The condensation on the selected zone with a slit set and a tube does not represent the other zone's condensate distribution completely. The droplet distribution is different in the direction of air-flow as well as in the direction of gravity. The size of the droplet may be another cause of poor prediction. The size of the droplet increases as the zone is away from the leading edge of fins due to the air-flow force, and the size of droplet decreases as the zone is away from the top of the heat exchanger due to the gravitational forces. These variations of droplet distribution and the distribution of the condensate are not accounted for the present model, and thus, the overpredicted mass of retained condensate is observed for three-row heat exchangers. Even though the model overpredicted the mass of retained condensate for three-row heat exchanger, the overall performance of this model was relatively good, especially in view of the complex physics.
Figure 4.1 Comparison of experimental data and the Gray & Webb correlations

Figure 4.2 Comparison of experimental data and the Wang et al. correlations
Table 4.1: \( j \) and \( f \) factor correlations proposed in existing literature

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Experimental condition</th>
<th>Author</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>Dry</td>
<td>McQuiston (1978b)</td>
<td>( j_{dry} = 0.004904 + 1.382(IP)^2 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>where,</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>( FP = Re_{dot}^{-0.25} \left( \frac{R}{R^<em>} \right)^{0.25} \left[ \frac{(S_T - 2R)P_f}{4(1-P_f \delta_f)} \right]^{-0.4} \left[ \frac{S_T}{2R^</em>} - 1 \right]^{-0.5} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>( R^* = \frac{A_{tot}}{A_{to}} ) and ( A_{tot} = \frac{4S_S S_T A_{min}}{\pi D_d D_{at} A_{fp}} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>For 4 rows of tubes</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>( j_{dry,4} = 0.0014 + 0.2618(JP) )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>For ( N_T ) number of tube rows</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>( j_{dry,N_T} = \left[ \frac{1 - 1280N_T Re_s^{-1.2}}{1 - 5120 Re_s^{-1.2}} \right] j_{dry,4} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>where,</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>( JP = Re_{dot}^{-0.4} \left( \frac{A_{tot}}{A_{to}} \right)^{-0.15} )</td>
</tr>
</tbody>
</table>

Range of variables and uncertainty:
- Tube diameter: 3/8 - 5/8 inches
- Tube spacing: 1 - 2 inches
- Fin pitch: 4 - 14 fins per inch
- Fin thickness: 0.006 - 0.01 inches
- Face velocity: 200-800 feet per minute
- Parameter: 0.08 < FP < 0.24
- Uncertainty: ±35% for f factors and ±10% for j factors
Table 4.1 $j$ and $f$ factor correlations proposed in existing literature (continued).

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Experimental condition</th>
<th>Author</th>
<th>Correlation</th>
<th>Range of variables and uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>Wet</td>
<td>McQuiston (1978b)</td>
<td>$j_{wet} = 0.004094 + 1.382[(FP)F(s)]^2$</td>
<td>Parameter (JP)J(s): 0.01 - 0.05</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>For 4 rows of tubes</td>
<td>Parameter (FP)F(s): 0.08 - 0.24</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$j_{wet,4} = 0.0014 + 0.2618(JP)J(s)$</td>
<td>Other range of variables are same as dry condition</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>For $N_{tr}$ number of tube rows</td>
<td>Uncertainty:</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$j_{wet,N_{tr}} = \left[ \frac{1 - 1280N_{tr} Re_{S_{tr}}^{-2.12}}{1 - 5120 Re_{S_{tr}}^{-2.12}} \right] j_{wet,4}$</td>
<td>$\pm 35%$ for $f$ factors and $\pm 10%$ for $j$ factors</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>where</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>FP and JP are given by dry condition.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>for film-type condensation;</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$J(s) = 0.84 + 0.00004 Re_{f_s}^{1.25}$ (Sensible $j$ factor)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$J(s) = (0.95 + 0.00004 Re_{f_s}^{1.25})F_f^2$ (Total $j$ factor)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$F(s) = \left( 0.6 + Re_{f_s}^{-0.15} \right) F_f^{-3}$ (Fanning friction factor)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>where $F_f = \frac{f_s}{f_s - \delta_f}$</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>for drop-type condensation;</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$J(s) = (0.9 + 0.000043 Re_{f_s}^{1.25})F_f^{-1}$ (Sensible $j$ factor)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$J(s) = (0.8 + 0.00004 Re_{f_s}^{1.25})F_f^4$ (Total $j$ factor)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$F(s) = \left( 0.325 + Re_{f_s}^{-0.05} \right) F_f^{-3}$ (Fanning friction factor)</td>
<td></td>
</tr>
</tbody>
</table>
Table 4.1 $j$ and $f$ factor correlations proposed in existing literature (continued).

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Experimental condition</th>
<th>Author</th>
<th>Correlation</th>
<th>Range of variables and uncertainty</th>
</tr>
</thead>
</table>
| Plain        | Dry                    | Nakayama & Xu (1983) | \[
  f_{dry, p} = 0.729 \text{Re}^{-0.6} \left( \frac{\delta_f}{f_s} \right)^{-0.6} \left( \frac{S_T}{D_{ot}} \right)^{-0.927} \left( \frac{S_T}{S_L} \right)^{0.515} \\
  j_{dry, p} = 0.479 \text{Re}^{-0.644} 
\] | $250 \leq \text{Re}_{Dh} \leq 3000$
$0.15\text{mm} \leq \delta_f \leq 0.2\text{mm}$
$0.18\text{mm} \leq f_s \leq 2.5\text{mm}$
Uncertainty:
$\pm 10\%$ for both $j$ and $f$ factors |
| Plain        | Dry                    | Gray & Webb (1986) | \[
  f_{dry} = f_f \frac{A_f}{A_{tot}} + f_t \left( 1 - \frac{A_f}{A_{tot}} \right) \left( 1 - \frac{\delta_f}{P_f} \right) \\
  f_f = 0.508 \text{Re}_{D_{ot}}^{-0.521} \left( \frac{S_T}{D_{ot}} \right)^{1.318} \\
\] where,
\[
  f_f = 0.508 \text{Re}_{D_{ot}}^{-0.521} \left( \frac{S_T}{D_{ot}} \right)^{1.318} \\
\] For 4 rows of tubes
\[
  j_{dry, 4} = 0.14 \text{Re}_{D_{ot}}^{-0.328} \left( \frac{S_T}{S_L} \right)^{-0.502} \left( \frac{f_s}{D_{ot}} \right)^{0.0312} \\
\] For $N_{tr}$ number of tube rows;
\[
  j_{dry, N_{tr}} = \left[ 0.991 \left( \frac{2.24 \text{Re}_{D_{ot}}^{-0.092} \left( \frac{N_{tr}}{4} \right)^{-0.031}}{\left( 1 - \frac{4}{N_{tr}} \right)^{0.607(4-N_{tr})}} \right) \right]^{0.991} \left( \frac{N_{tr}}{4} \right)^{0.031} \left( 1 - \frac{4}{N_{tr}} \right)^{0.607(4-N_{tr})} j_{dry, 4} \\
\] | $500 \leq \text{Re}_{D_{ot}} \leq 24700$
for 4 rows of tubes
$2400 \leq \text{Re}_{D_{ot}} \leq 24700$
for $n$ rows of tubes
$1.97 \leq S_T/D_{ot} \leq 2.55$
$1.70 \leq S_L/D_{ot} \leq 2.58$
$0.08 \leq f_s/D_{ot} \leq 0.64$
$1 \leq N_{tr} \leq 8$ or more
Uncertainty:
$95\%$ of data within $\pm 13\%$
for $f$ factors
$89\%$ of data within $\pm 10\%$
for $j$ factors with 4 tube rows. |
Table 4.1 \( j \) and \( f \) factor correlations proposed in existing literature (continued).

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Experimental condition</th>
<th>Author</th>
<th>Correlation</th>
<th>Range of variables and uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>Wet</td>
<td>Wang et al. (1997)</td>
<td>( f_{wet} = 28.209 \Re^{-0.5653} N^0.1026 \left( \frac{f_s}{D_{coll}} \right)^{-1.3405} \varepsilon^{-1.3343} )</td>
<td>( 300 \leq \Re_{Deqoll} \leq 5500 ) ( 1.82 \text{mm} \leq f_s \leq 3.20 \text{mm} ) ( 2 \leq N_r \leq 6 ) Uncertainty: 91% of data within ±10% for ( f ) factors 92% of data within ±10% for ( j ) factors</td>
</tr>
<tr>
<td>Slit</td>
<td>Dry</td>
<td>Nakayama &amp; Xu (1986)</td>
<td>( f_{dry,s} = f_{dry,p} (1 + F_j) ) ( j_{dry,s} = j_{dry,p} F_j ) ( F_j = 1 + 1093 \left( \frac{\delta_f}{\delta_a} \right)^{1.24} \phi_s^{0.944} \Re^{-0.58} + 1.097 \left( \frac{\delta_f}{\delta_a} \right)^{2.09} \phi_s^{2.26} \Re^{0.88} ) ( \phi_s = \frac{(2n_s - 1)S_s}{S_f S_L - \frac{\pi D_{st}^2}{4}} ) (ratio of slit area to total fin area)</td>
<td>Other range of variables are same as plain-fin Uncertainty: ±10% for both ( j ) and ( f ) factors</td>
</tr>
</tbody>
</table>
Table 4.2 $j$ and $f$ factor correlations

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Experiment Condition</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain, Uncoated</td>
<td>Wet</td>
<td>$f_{wet} = 2.109 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.6347} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j_{wet} = 0.08163 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.2664} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td>Slit, Uncoated</td>
<td>Dry</td>
<td>$f_{dry} = 1.024 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.5123} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j_{dry} = 0.2476 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.209} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td>Slit, Uncoated</td>
<td>Wet</td>
<td>$f_{wet} = 1.265 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.2991} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j_{wet} = 0.3647 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.1457} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td>Slit, Coated</td>
<td>Dry</td>
<td>$f_{dry} = 3.826 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.5959} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j_{dry} = 0.4813 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.1329} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td>Slit, Coated</td>
<td>Wet</td>
<td>$f_{wet} = 0.502 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.2593} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j_{wet} = 0.4559 \left( \frac{Re_{Dcoll}}{D_{Coll}} \right)^{-0.2382} \left( \frac{S_l N_r}{D_{Coll}} \right)$</td>
</tr>
</tbody>
</table>

Table 4.3 Range of parameter

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Range of parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain, Uncoated, Wet</td>
<td>$550 \leq Re_{Dcoll} \leq 2000$, $1.5mm \leq f_s \leq 1.7mm$, $N_r \leq 3$</td>
</tr>
<tr>
<td>Slit, Uncoated, Wet and Dry</td>
<td>$550 \leq Re_{Dcoll} \leq 2000$, $1.3mm \leq f_s \leq 1.7mm$, $N_r \leq 3$</td>
</tr>
<tr>
<td>Slit, Coated, Wet and Dry</td>
<td>$550 \leq Re_{Dcoll} \leq 2000$, $1.3mm \leq f_s \leq 1.7mm$, $N_r \leq 3$</td>
</tr>
</tbody>
</table>
Figure 4.3 Comparison of experimental data and present correlations for uncoated, plain-fin-and-tube heat exchanger under wet condition

Figure 4.4 Comparison of experimental data and present correlations for uncoated, slit-fin-and-tube heat exchanger under dry condition
Table 4.4 Uncertainties of $j$ and $f$ factor correlations

<table>
<thead>
<tr>
<th></th>
<th>Wet</th>
<th>$f$: 80% predicted within 20%</th>
<th>$j$: 96% predicted within 10%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain, Uncoated</td>
<td>Dry</td>
<td>$f$: 82% predicted within 20%</td>
<td>$j$: 88% predicted within 15%</td>
</tr>
<tr>
<td>Slit, Uncoated</td>
<td>Wet</td>
<td>$f$: 92% predicted within 20%</td>
<td>$j$: 90% predicted within 15%</td>
</tr>
<tr>
<td>Slit, Coated</td>
<td>Dry</td>
<td>$f$: 92% predicted within 10%</td>
<td>$j$: 87% predicted within 15%</td>
</tr>
<tr>
<td>Slit, Coated</td>
<td>Wet</td>
<td>$f$: 94% predicted within 20%</td>
<td>$j$: 95% predicted within 15%</td>
</tr>
</tbody>
</table>

Figure 4.5 Comparison of present correlations with Nakayama and Xu's correlations
Figure 4.6 Condensation on uncoated slit-fin surface
Figure 4.7 Droplet with a circular contact line on an inclined surface [30]

Figure 4.8 Forces acting on bridges retained between fins at fin-tube junction [30]
Figure 4.9 Slit-fin stock divided by small zones
Figure 4.10 Droplet distribution
Table 4.5 Comparison between predicted and measured mass of retained condensate

<table>
<thead>
<tr>
<th>HX</th>
<th>Predicted mass of retained condensate (g)</th>
<th>Measured mass of retained condensate (g)</th>
<th>% Decrease</th>
<th>% Loss degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1.3-2-Uncoated</td>
<td>225.6</td>
<td>229 ~ 257</td>
<td>87.5</td>
<td>40.42</td>
</tr>
<tr>
<td>S1.3-3-Uncoated</td>
<td>514.4</td>
<td>187 ~ 647.2</td>
<td>87.5</td>
<td>40.42</td>
</tr>
<tr>
<td>S1.5-2-Uncoated</td>
<td>193.8</td>
<td>216 ~ 219.8</td>
<td>87.5</td>
<td>40.42</td>
</tr>
<tr>
<td>S1.5-3-Uncoated</td>
<td>442.7</td>
<td>286 ~ 321.6</td>
<td>87.5</td>
<td>40.42</td>
</tr>
<tr>
<td>S1.7-2-Uncoated</td>
<td>172.5</td>
<td>178.3 ~ 228.13</td>
<td>87.5</td>
<td>40.42</td>
</tr>
<tr>
<td>S1.7-3-Uncoated</td>
<td>394.3</td>
<td>277 ~ 316.22</td>
<td>87.5</td>
<td>40.42</td>
</tr>
</tbody>
</table>
CHAPTER 5 - CONCLUSIONS AND RECOMMENDATIONS

Mass of retained condensate for real-time and steady-state has been measured and the condensate accumulation effects on air-side heat exchanger performance have been investigated using dry and wet experiments. Several conclusions have been drawn from these data. New $j$ and $f$ factor correlations for slit-fin-and-tube heat exchangers have been developed and compared to existing correlations. The new correlations consider the air-side Reynolds number based on collar diameter, fin spacing, and number of tube rows. A simple model to predict the condensate retention on uncoated, slit-fin-and-tube heat exchangers has been developed based on the heat exchanger geometry and contact angles. Further design methods and recommendations for the slit-fin-and-tube heat exchanger are presented in this chapter. $j$ and $f$ multipliers developed from present correlations are also discussed in this chapter.

5.1 Condensate Retention Characteristics

The real-time mass of condensation was measured in order to understand the dynamic behavior of the retained condensate, and the steady-state mass of the retained condensate was measured to see the effects of heat exchanger geometry on the quantity of retained condensate. Several trends have been observed through these experiments.

5.1.1 Conclusions from Real-Time Retention Data

- When a plain-fin-and-tube heat exchanger is tested at two different face velocities, the time required to reach a steady-state is longer and the steady-state value of retained condensate is higher at lower velocities.

- When the fin spacing gets smaller, an overshoot can be observed before reaching a steady-state mass of retained condensate. According to previous studies, the mass of retained condensate increases, asymptotically approaching a steady-state value for the heat exchangers with a wide fin spacing. However, as the fin spacing is reduced, a different trend is observed. The mass of retained condensate gradually increases until it reaches a maximum value, then decreases to the steady-state value.
• When a slit-fin-and-tube heat exchanger is tested, the retained condensate is formed as bridges between fins and slits. Due to the shedding effect of retained condensate, an oscillation in the condensate rate is seen at higher velocities.

5.1.2 Conclusions from Steady-State Retention Data

• The plain-fin-and-tube heat exchanger retains more condensate if it is uncoated. The quantity of retained condensate decreases as much as 50% when the fin is treated with a hydrophilic coating.

• A two-row, plain-fin-and-tube heat exchanger retains more condensate per total area than three-row, plain-fin-and-tube heat exchanger. This trend may be due to the shedding effect. The droplet size increases as it accumulates with other droplets. The droplets move in the direction of the air-flow until it reaches a maximum size, and is shed downwards due to the gravitational force.

• The effect of fin spacing on the retained condensate for the plain-fin-and-tube heat exchanger decreases as the velocity increases. This trend is also shown when the mass of retained condensate divided by the total area.

• Plain-fin-and-tube heat exchangers tend to retain less condensate than slit-fin-and-tube heat exchangers with same fin spacing and number of tube rows.

• A slit-fin-and-tube heat exchanger with wider fin spacing retains more condensate than that with closer fin spacing when the mass of retained condensate is divided by the total area. This decrease may be due to the condensate bridge that is formed when the condensate drops accumulate as they are shed downward.

5.2 Air-Side Heat Exchanger Performance

Experiments on air-side heat exchanger performance under dry and wet conditions were conducted in order to investigate the effect of the condensate accumulation. The air-side sensible heat transfer and pressure drop were observed for plain-fin-and-tube and slit-fin-and-tube heat exchangers.

• Pressure drop across the heat exchanger is greater and the air-side heat transfer coefficient is lower when the heat exchanger is tested under dehumidifying conditions for both plain-fin-and-tube and slit-fin-and-tube heat exchangers. When a heat
exchanger is tested under dehumidifying conditions, condensate accumulates on the heat exchanger surface. The condensate increases pressure drop across the heat exchanger, resulting in an increase in the $f$ factor. The degradation of the $j$ factor under wet conditions may be due to the condensate retention. The condensate retention occurs with a deleterious effect of the heat transfer at lower Reynolds number.

- Under wet conditions, friction factors for a plain-fin-and-tube heat exchanger are independent of number of tube rows while $j$ factors may increase slightly as the number of tube rows decreases. (the difference in $j$ factors is within experimental uncertainty).

- Sensible $j$ factors are independent of coating under wet conditions while $f$ factors increase when the slit-fin-and-tube heat exchanger is uncoated. Coating the fin surface changes dropwise condensate into filmwise condensate and helps to reduce the pressure drop across the heat exchanger by letting the air-flow pass through the heat exchanger easily.

- Sensible $j$ and $f$ factors for a slit-fin-and-tube heat exchanger under wet conditions are dependent of the fin spacing. $j$ factors increase as fin spacing increases. However, $f$ factors increase as fin spacing decreases at higher velocities and the $f$ factors become independent of fin spacing at higher velocities. These effects are small and within the experimental uncertainty, but the trends are consistent.

- $f$ factor increases as the number of tube row decreases, while the effect of the number of tube rows on $j$ factors are inconclusive. The effect of the number of tube rows for the $f$ factor variation increases as the Reynolds number increases.

5.3 $j$ and $f$ Factor Correlations

$j$ and $f$ factor correlations were developed for plain-fin-and-tube and slit-fin-and-tube heat exchangers. Five different conditions were used because experimental conditions, fin geometry, and fin coating affect the $j$ and $f$ factors significantly.

- $j$ and $f$ factor correlations account for air-side Reynolds number based on collar diameter, number of tube rows, and fin spacing.
• Most of the experimental $f$ factors and $j$ factors are predicted within 20% and 15%, respectively with present correlations, and trends are accurately modeled.

• The correlation for an uncoated slit-fin-and-tube heat exchanger under dry condition was compared to the correlations developed by Nakayama and Xu [21]. The $f$ factor correlation developed by Nakayama and Xu is slightly less than the present correlations and a discrepancy in $j$ factors increases as the Reynolds number increases. The discrepancies between the two correlations are due to the difference in heat exchanger geometry as well as the slit configuration. In addition, the difference in calculating fin efficiency causes the discrepancies. These differences notwithstanding, the overall agreement is reasonable and buttresses the current results.

5.4 Retention Modeling

A simple retention model to predict the mass of retained condensate on uncoated, slit-fin-and-tube heat exchangers was developed and compared to the experimental data. A slit-fin stock was tested under condensing conditions in order to investigate the drop distribution. The slit-fin stock was divided into several zones with a tube and a slit set. The drop distribution was analyzed using Scion Image. An equivalent diameter of each droplet was calculated using the area of each droplet given by Scion Image. The model accounts for the mass of condensate formed as droplets and as bridges between fin and fin-tube junction. The model predicts the maximum mass of retained condensate accurately for two-row, uncoated, slit-fin-and-tube heat exchangers; however, the prediction for three-row, uncoated, slit-fin-and-tube heat exchangers was slightly higher than the experimental data.

5.5 Recommendations for Future Experimental Studies

In order to validate and improve the utility of the proposed correlations, as new data become available they should be compared to the correlations. Data from similar heat exchangers may be useful in validating the correlations; whereas, data from heat exchangers outside the test matrix used in this study may be useful in extending the correlations to a wider parameter space. It would be especially useful to have data from
specimens that have been in service for an extended period. Such data would give insights into contact angle changes and the effects of such changes on thermal performance. These correlations should be adjusted as advances in our understanding are achieved.

In order to improve the condensate retention model, several different condensate geometries such as fillets retained at fin-tube junctions and the filmwise condensation should be identified and considered by the model. Air-flow forces on retained condensate should be more accurately determined. The variation of retained condensation along the height of the fin as well as the length in the direction of air-flow should be included in the modeling. The bridges formed between slits should be identified and included in the modeling procedure. In order to model various other coated fins, the drop distribution on a coated slit-fin zone should be analyzed and the force balance on filmwise condensation should be studied.

5.6 Recommendations for Design Methods and Guidelines of Heat Exchanger Development

The quantity of retained condensate can be decreased by coating the heat exchanger. This current study shows that the heat transfer performance is essentially independent of the coating, but there is a profound effect on pressure drop. The slit-fin-and-tube heat exchanger has better heat transfer performance than plain-fin-and-tube heat exchanger. However, the slit-fin-and-tube heat exchanger has higher pressure drop than the plain-fin-and-tube heat exchanger under dry conditions. Under wet conditions, the difference in the pressure drops between the plain- and slit-fin decreases. A slit-fin-and-tube heat exchanger with wider fin spacing has higher heat transfer performance and a lower pressure drop than that for a closer fin spacing. Since slits are raised from the fin surface, the minimum air-flow area of the slit-fin-and-heat exchanger gets smaller as the fin spacing gets narrower. As a result, as fin spacing decreases, more condensate bridges are formed. These condensate bridges increase the pressure drop and decrease the heat transfer performance due to the deleterious effect.
5.6.1 Optimum Fin Spacing

By extrapolating the results of heat exchanger performance testing, using the new correlations, an evaporator design improvement can be suggested. In this study, a coated, two-row, slit-fin-and-tube heat exchanger with 1.7 mm fin spacing has the best heat transfer performance under wet conditions, while having the lowest pressure drop among the other heat exchangers tested. In order to improve the efficiency of this evaporator by 10% while keeping the pressure drop constant, a fin spacing of 2.0 mm is recommended with same operating conditions and heat exchanger geometry. Table 5.1 shows a comparison between the slit-fin-and-tube heat exchangers with 1.7 mm and 2.0 mm fin spacing. When the fin spacing is increased from 1.7 mm to 2.0 mm, the $f$ factor increases by only about 2.5%, but the $j$ factor increases 12.6%. The overall exchanger volume for this $f_s=2.0$ mm design will have to be about 5% larger, but it uses about 12% less material because of the improved thermal performance. Experiments to confirm this improved design are recommended.

5.6.2 $j$ and $f$ Multipliers

$j$ and $f$ multipliers can be derived from the correlations. Even though the $j$ and $f$ multipliers are limited to certain ranges of parameters, these multipliers will give an idea of the wet performance for a slit-fin-and-tube heat exchanger by knowing the dry $j$ and $f$ factors. For completeness, these multipliers are presented in Table 5.2.
Table 5.1 Optimum fin spacing of coated, two-row, slit-fin-and-tube heat exchanger
$D_{coll}=7.416 \text{ mm}, L=25.0 \text{ mm}$

<table>
<thead>
<tr>
<th>$Re_{coll}$</th>
<th>$f_s (f_s = 1.7 \text{ mm})$</th>
<th>$L/D_{coll}$</th>
<th>$f_{wet}$</th>
<th>$f_{dry}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>600</td>
<td>0.2292</td>
<td>3.371</td>
<td>0.1495</td>
<td>0.01525</td>
</tr>
<tr>
<td>1500</td>
<td>0.2292</td>
<td>3.371</td>
<td>0.1179</td>
<td>0.01226</td>
</tr>
<tr>
<td>600</td>
<td>0.2707 ($f_s = 2.0 \text{ mm}$)</td>
<td>3.371</td>
<td>0.1534 (2.6%)</td>
<td>0.01717 (12.6%)</td>
</tr>
<tr>
<td>1500</td>
<td>0.2707 ($f_s = 2.0 \text{ mm}$)</td>
<td>3.371</td>
<td>0.1209 (2.5%)</td>
<td>0.0138 (12.6%)</td>
</tr>
</tbody>
</table>

Table 5.2 $j$ and $f$ multipliers

<table>
<thead>
<tr>
<th>Fin geometry</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slit, Uncoated</td>
<td>$f_{wet} = \left[ 1.235 \left(Re_{Dcoll} \right)^{0.2133} \left(\frac{f_s}{D_{Coll}} \right)^{0.4397} \left(\frac{S_L N_t}{D_{Coll}} \right)^{-0.3651} \right] f_{dry}$</td>
</tr>
<tr>
<td></td>
<td>$j_{wet} = \left[ 1.473 \left(Re_{Dcoll} \right)^{0.0633} \left(\frac{f_s}{D_{Coll}} \right)^{0.7775} \left(\frac{S_L N_t}{D_{Coll}} \right)^{0.0611} \right] j_{dry}$</td>
</tr>
<tr>
<td>Slit, Coated</td>
<td>$f_{wet} = \left[ 0.1312 \left(Re_{Dcoll} \right)^{0.3366} \left(\frac{f_s}{D_{Coll}} \right)^{0.3908} \left(\frac{S_L N_t}{D_{Coll}} \right)^{0.5034} \right] f_{dry}$</td>
</tr>
<tr>
<td></td>
<td>$j_{wet} = \left[ 0.9472 \left(Re_{Dcoll} \right)^{-0.1053} \left(\frac{f_s}{D_{Coll}} \right)^{-0.2871} \left(\frac{S_L N_t}{D_{Coll}} \right)^{-0.1801} \right] j_{dry}$</td>
</tr>
</tbody>
</table>
REFERENCES


APPENDIX A - DATA REDUCTION

Data reduction details are provided in this appendix. EES is used for data reduction, and a “parametric table” is used to streamline the EES implementation. The program is provided 13 input parameters in the table; these inputs are described in Table A.1. The program calculates 14 output variables; they are described in Table A.2. The EES equation worksheet for slit-fin-and-tube heat exchanger is provided in Table A.3, and all other parameters taken as known are described in Table A.4. In the reminder of this appendix, the equations used to reduce the data (see Table A.5) are discussed.

A.1 Coolant-Side Calculations

A.1.1 Flow Rate of Coolant

The flow rate of coolant is measured with a volumetric flow meter which provides a 5 volt dc pulse with $3.092 \times 10^5$ pulse per cubic meter of coolant. The mass flow rate can be calculated by multiplying the volumetric flow rate by the density of the coolant. Since the volumetric flow rate is measured at the outlet stream of the coolant, coolant outlet temperature is used to calculate the density of the coolant.

A.1.2 Coolant Properties

The coolant properties have been evaluated with specific gravity measured with a hydrometer to determine the volume fraction of the glycol. A 32.6% volume fraction has been determined for this experiment based on the tables provided by the manufacturer.

A1.3 Coolant-Side Heat Transfer Coefficient

The coolant-side heat transfer coefficient $h_i$ is determined by applying the correlation developed by Gnielinski [36] for turbulent flow. This correlation is used since it is applicable to the transitional Reynolds numbers of this study.

Reynolds number of the coolant

$$Re_c = \frac{\rho_c V_c D_c}{\mu_c}$$
Nusselt number of the coolant based on the inside tube

\[
Nu_c = \left( \frac{f_c}{8} \right) \left( \frac{Re_c - 1000}{Pr_c} \right)
\]

\[
f_c = \left( \frac{f_c}{8} \right) \left( \frac{Pr_c^{2/3} - 1}{Pr_c} \right)
\]

\[
Pr_c = \frac{C_P \cdot \mu_c}{k_c}
\]

Tube-side heat transfer coefficient

\[
h_i = \frac{Nu_c \cdot k_c}{D_c}
\]

A.2 Air-Side Calculations

A.2.1 Face Velocity Calibration

The velocity of air is measured using a constant-temperature thermal anemometer. The temperature is also measured to calibrate the velocity as the manufacturer recommended. The room temperature and the pressure are measured using a NOVA barometer located in the lab.

\[
V_{cali} = \frac{V_{read} \left( 273 + \frac{T_{read} \cdot \left( ^oC \right)}{294.1P_{atm}} \right) \cdot 101.325}{101.325}
\]

A.2.2 Air Properties

The properties of air are evaluated by Psychometric relations within EES by giving atmospheric pressure, temperature, and humidity ratio.

A.2.3 Heat Transfer Rates

The coolant and total heat transfer rates are calculated using the following equations. The energy balance is calculated for each wet experiment and only those data with uncertainties of less than 10% are used for \( j_{\text{wet}} \) & \( f_{\text{wet}} \) factors analysis.
Coolant-side Heat Rate

\[ Q_c = m_c C_p c (T_{out,c} - T_{in,c}) \]

Sensible Heat Rate

\[ Q_{sens} = m_{air} C_p \text{air} (T_{in,air} - T_{out,air}) \]

Total Heat Rate

\[ Q_{tot} = m_{air} (h_{in,air} - h_{out,air}) \]

Energy Balance

\[ EB = \frac{Q_{ave} - Q_{tot}}{Q_{ave}} \]

\[ Q_{ave} = \frac{Q_{tot} + Q_c}{2} \]

A.3 Fin Efficiency Calculations

A.3.1 ARI Standard 410 Equivalent Circular Area

The calculation of inner radius depends on the fin-tube connection. ARI Standard 410 [32] recommends the following equation for plate-type fins with collars touching the adjacent fin.

\[ r_{ij} = \frac{(D_{oi} + 2\delta)}{2} \]

A.3.2 Sector Method with Conduction

The sector method can be used to determine the fin efficiency of hexagonal fins of constant thickness attached to the round tubes. The hexagonal fin around each tube is divided into 8 different zone as shown in Figure A.1. Individual zone is then divided into 4 sectors. The number of sectors can be increased for better approximation.
The radius of each edge of sector is approximated and the radius ratio, $R_n$, and the surface area of each sector, $S_n$, are calculated as follows.

**Sectors with constant M edge** (for zone 2, 3, 6, and 7)

$$R_n = \frac{M}{r_y} \sqrt{\left(\frac{2n-1}{2N}\right)^2 + \left(\frac{L}{M}\right)^2}$$

$$S_n = \frac{r_y^2}{2} \left(R_n^2 - 1\right) \left[\tan^{-1}\left(\frac{nM}{NL}\right) - \tan^{-1}\left(\frac{(n-1)M}{NL}\right)\right]$$

where $n = 1, 2, 3 \ldots$ $N$ is number of sectors in each zone.

**Sectors with constant L edge** (for zone 1, 4, 5, and 8)

$$R_n = \frac{M}{r_y} \sqrt{\left(\frac{2n-1}{2N}\right)^2 \left(\frac{L}{M}\right)^2} + 1$$
\[ S_n = \frac{r_g^2}{2} \left( R_n^2 - 1 \right) \left[ \tan^{-1} \left( \frac{nL}{NM} \right) - \tan^{-1} \left( \frac{(n-1)L}{NM} \right) \right] \]

\[ n = 1, 2, 3 \ldots N \text{ is number of sectors in each zone.} \]

where

\[ M = \frac{S_r}{2}, \quad L = \frac{S_L}{2} \]

The calculated value, \( R_n \), is used with the modified Schmidt's [38] equation, as described by Hong and Webb [37]. The total fin efficiency can be calculated by the sum of the multiplication of fin efficiencies for each sector in each zone and \( S_n \) divided by the sum of surface areas of all eight sectors in each zone.

\[ \Phi_n = (R_n - 1)(1 + 0.35 \ln R_n) \]

\[ \eta_n = \frac{\tanh(mr_g \Phi_n) \cos(0.1mr_g \Phi_n)}{mr_g \Phi_n} \]

where

\[ m = \sqrt{\frac{2h}{\kappa, \delta_f}} \]

and

\[ \eta_f = \frac{\sum_{n=1}^{N} S_n \eta_n}{\sum_{n=1}^{N} S_n} \]

where \( n = 1, 2, 3, \ldots \), and \( N, N = \text{number of sectors in each zone.} \)
The radius ratio, $R_n$, has been approximated and modified for the slit-fin heat exchanger for better estimation of fin efficiency. Since accurate calculation for circular inner radius for slit-fin heat exchanger is complicated, an assumption of dividing sectors has been made. The $R_n$ of the sectors with constant $M$ edge can be modified by replacing $M$ and $L$ to $M_s$ and $L_s$, and by adding the height of slits to the sectors with constant $M_s$ edge as shown in Figure A.2. The values of $M_s$ and $L_s$ are measured directly. The surface areas and the radius of constant $L$ edge for slit fin heat exchanger can be calculated similar to the plain fin heat exchanger.

Figure A.2 Sector method with conduction (slit-fin)

Sectors with constant $M_s$ edge (for zone 2, 3, 6, and 7)

$$R_n = \frac{M_s \left( \frac{2n - 1}{2N} \right)^2 + \left( \frac{L_s}{M_s} \right)^2}{r_y} + h_s$$

$n = 1, 2, 3 \ldots N$, where $N$ is the number of sectors in each zone

$h_s$ = Height of slits measured directly from fin surface
A.4 Air-Side Heat Transfer Coefficient Calculation

The air-side heat transfer coefficient for wet condition is calculated based on ARI Standard 410. The air-side resistance of external fins and tube wall calculations are given by

$$R_f = \frac{1 - \eta_f}{\eta_f h_{eff} A_{tot}}$$

Where

$$h_{eff} = \frac{m'' h_{wet}}{C_{p,\text{air}}}$$

$m''$=slope of enthalpy-saturation temperature curve from Threlkeld, $di/dT$

The assumption of fully wet experimental condition is made when $T_{\text{surface}} < T_{dp}$ and $Q_{\text{sens}}/Q_{\text{tot}} < 0.95$. By iterating the data reduction equations and by knowing fin efficiency, the air-side heat transfer coefficient can be calculated.
**Table A.1 Input parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical Meaning</th>
<th>Method of Determination</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N_{	ext{tube row}}$</td>
<td>Number of tube rows</td>
<td>Geometry</td>
</tr>
<tr>
<td>$f_{\text{smm}}$</td>
<td>Fin spacing in mm</td>
<td>Geometry</td>
</tr>
<tr>
<td>$N_f$</td>
<td>Number of fin</td>
<td>Geometry</td>
</tr>
<tr>
<td>$V_{\text{read}}$</td>
<td>Air-flow face velocity (m/s)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{read}}$</td>
<td>Air-flow face temperature (°C)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$R_c$</td>
<td>Flow-rate of coolant (pulse/10s)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{in, air}}$</td>
<td>Dry bulb inlet temperature (°C)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{out, air}}$</td>
<td>Dry bulb outlet temperature (°C)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{dp, in}}$</td>
<td>Web bulb inlet temperature (°F)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{dp, out}}$</td>
<td>Web bulb outlet temperature (°F)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{in, c}}$</td>
<td>Coolant inlet temperature (°C)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$T_{\text{in, c}}$</td>
<td>Coolant outlet temperature (°C)</td>
<td>Measured during experiment</td>
</tr>
<tr>
<td>$\Delta P_{\text{air}}$</td>
<td>Pressure drop across heat exchanger (in water)</td>
<td>Measured during experiment</td>
</tr>
</tbody>
</table>

**Table A.2 Output variables**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{\text{air}}$</td>
<td>Face velocity (m/s)</td>
</tr>
<tr>
<td>$f$</td>
<td>Friction factor</td>
</tr>
<tr>
<td>$j$</td>
<td>$j$ factor</td>
</tr>
<tr>
<td>$R_{\text{Dh}}$</td>
<td>Reynolds number based on hydraulic diameter</td>
</tr>
<tr>
<td>$h_i$</td>
<td>Coolant-side heat transfer coefficient (KW/m²-K)</td>
</tr>
<tr>
<td>$h$</td>
<td>Air-side heat transfer coefficient (KW/m²-K)</td>
</tr>
<tr>
<td>$Q_{\text{tot}}$</td>
<td>Total heat transfer rate (KW)</td>
</tr>
<tr>
<td>$Q_{\text{sens}}$</td>
<td>Sensible heat transfer rate (KW)</td>
</tr>
<tr>
<td>$Q_c$</td>
<td>Coolant-side heat transfer rate (KW)</td>
</tr>
<tr>
<td>$E_{\text{B ratio}}$</td>
<td>Energy Balance</td>
</tr>
<tr>
<td>$N_{\text{u coll}}$</td>
<td>Nusselt number based on collar diameter</td>
</tr>
<tr>
<td>$R_{\text{Dcoll}}$</td>
<td>Reynolds number based on collar diameter</td>
</tr>
<tr>
<td>$R_{\text{Dc}}$</td>
<td>Reynolds number based on inside tube diameter</td>
</tr>
<tr>
<td>$V_c$</td>
<td>Coolant flow rate (m/s)</td>
</tr>
</tbody>
</table>
Table A.3 EES Equations under wet conditions
(for slit-fin-and-tube heat exchangers)

```plaintext
{****Humidity Calculation****}
PROCEDURE HUM1(Tdp_inC, P_readN, P_satl, Wl)

C8=-5.8002206*10^3
C9=-5.516256
C10=-4.8640239*10^-2
C11=4.1764768*10^-5
C12=-1.4452093*10^-8
C13=6.5459673

C1=-5.9745359*10^3
C2=-5.1523058*10^-1
C3=-9.677843*10^-3
C4=6.2215701*10^-7
C5=2.0747825*10^-9
C6=-9.484024*10^-13
C7=4.1635019

IF (Tdp_inC>0) THEN
    Tdp_inK:=Tdp_inC+273.15
    P_sat1:=Exp(C8/Tdp_inK+C9+C10*Tdp_inK+C11*(Tdp_inK)^2+C12*(Tdp_inK)^3+C13*ln(Tdp_inK))
    w1:=0.62188*P_sat1/(P_readN-P_satl)
ELSE
    Tdp_inK:=Tdp_inC+273.15
    P_sat1:=Exp(C1/Tdp_inK+C2+C3*Tdp_inK+C4*(Tdp_inK)^2+C5*(Tdp_inK)^3+C6*(Tdp_inK)^4+C7*ln(Tdp_inK))
    w1:=0.62188*P_sat1/(P_readN-P_satl)
ENDIF
END

PROCEDURE HUMAIR1(Tin_air, P_readN, P_airl, W_airl)

C8=-5.8002206*10^3
C9=-5.516256
C10=-4.8640239*10^-2
C11=4.1764768*10^-5
C12=-1.4452093*10^-8
C13=6.5459673

C1=-5.9745359*10^3
C2=-5.1523058*10^-1
C3=-9.677843*10^-3
C4=6.2215701*10^-7
C5=2.0747825*10^-9
C6=-9.484024*10^-13
C7=4.1635019
```

100
IF (Tin_air > 0) THEN
    Tin_airK := Tin_air + 273.15
    P_air1 := Exp(C8/Tin_airK + C9 + C10*Tin_airK + C11*(Tin_airK)^2 + C12*(Tin_airK)^3 + C13*ln(Tin_airK))
    W_air1 := 0.62188*P_air1/(P_readN - P_air1)
ELSE
    Tin_airK := Tin_air + 273.15
    P_air1 := Exp(C1/Tin_airK + C2 + C3*Tin_airK + C4*(Tin_airK)^2 + C5*(Tin_airK)^3 + C6*(Tin_airK)^4 + C7*ln(Tin_airK))
    W_air1 := 0.62188*P_air1/(P_readN - P_air1)
ENDIF
END

PROCEDURE HUM2(Tdp_outC, P_readN, P_sat2, w2)

C8 = -5.8002206*10^3
C9 = -5.516256
C10 = -4.8640239*10^(-2)
C11 = 4.1764768*10^(-5)
C12 = 1.4452093*10^(-8)
C13 = 6.5459673
C1 = -5.9745359*10^3
C2 = -5.1523058*10^(-1)
C3 = -9.677843*10^(-3)
C4 = 6.2215701*10^(-7)
C5 = 2.0747825*10^(-9)
C6 = -9.484024*10^(-13)
C7 = 4.1635019

IF (Tdp_outC > 0) THEN
    Tdp_outK := Tdp_outC + 273.15
    P_sat2 := Exp(C8/Tdp_outK + C9 + C10*Tdp_outK + C11*(Tdp_outK)^2 + C12*(Tdp_outK)^3 + C13*ln(Tdp_outK))
    w2 := 0.62188*P_sat2/(P_readN - P_sat2)
ELSE
    Tdp_outK := Tdp_outC + 273.15
    P_sat2 := Exp(C1/Tdp_outK + C2 + C3*Tdp_outK + C4*(Tdp_outK)^2 + C5*(Tdp_outK)^3 + C6*(Tdp_outK)^4 + C7*ln(Tdp_outK))
    w2 := 0.62188*P_sat2/(P_readN - P_sat2)
ENDIF
END

PROCEDURE HUMAIR2(Tout_air,P_readN: P_air2, W_air2)

C8=-5.8002206*10^3
C9=-5.516256
C10=-4.8640239*10^-2
C11=4.1764768*10^-5
C12=-1.4452093*10^-8
C13=6.5459673

C1=-5.9745359*10^3
C2=-5.1523058*10^-1
C3=-9.677843*10^-3
C4=6.2215701*10^-7
C5=2.0747825*10^-9
C6=-9.480424*10^-13
C7=4.1635019

IF (Tout_air>0) THEN

    Tout_airK:=Tout_air+273.15
    P_air2:=Exp(C8/Tout_airK+C9+C10*Tout_airK+C11*(Tout_airK)^2+C12*(Tout_airK)^3+C13*ln(Tout_airK))
    W_air2:=0.62188*P_air2/(P_readN-P_air2)

ELSE

    Tout_airK:=Tout_air+273.15
    P_air2:=Exp(C1/Tout_airK+C2+C3*Tout_airK+C4*(Tout_airK)^2+C5*(Tout_airK)^3+C6*(Tout_airK)^4+C7*ln(Tout_airK))
    W_air2:=0.62188*P_air2/(P_readN-P_air2)

ENDIF
END

PROCEDURE HUMSAT1(Tsin,P_readN: P_s1, ws1)

C8=-5.8002206*10^3
C9=-5.516256
C10=-4.8640239*10^-2
C11=4.1764768*10^-5
C12=-1.4452093*10^-8
C13=6.5459673

C1=-5.9745359*10^3
C2=-5.1523058*10^-1
C3=-9.677843*10^-3
C4=6.2215701*10^-7
C5=-2.0747825*10^-9
C6=5.480424*10^-13
C7=4.1635019
C6=-9.484024*10^(-13)
C7=4.1635019

IF (Tsin>0) THEN

\[ \begin{align*}
TsinK &= Tsin + 273.15 \\
P_{s1} &= \exp(C8/TsinK+C9+C10*TsinK+C11*(TsinK)^2+C12*(TsinK)^3+C13*\ln(TsinK)) \\
ws1 &= 0.62188*P_{s1}/(P_{readN}-P_{s1})
\end{align*}\]

ELSE

\[ \begin{align*}
TsinK &= Tsin + 273.15 \\
P_{s1} &= \exp(C1/TsinK+C2+C3*TsinK+C4*(TsinK)^2+C5*(TsinK)^3+C6*(TsinK)^4+C7*\ln(TsinK)) \\
ws1 &= 0.62188*P_{s1}/(P_{readN}-P_{s1})
\end{align*}\]

ENDIF

ENDIF

END

PROCEDURE HUMSAT2(Tsout,P_{readN}: P_{s2}, ws2)

C8=-5.8002206*10^3
C9=-5.516256
C10=-4.8640239*10^(-2)
C11=4.1764768*10^(-5)
C12=-1.4452093*10^(-8)
C13=6.5459673
C1=-5.9745359*10^3
C2=-5.1523058*10^(-1)
C3=-9.677843*10^(-3)
C4=6.2215701*10^(-7)
C5=2.0747825*10^(-9)
C6=-9.484024*10^(-13)
C7=4.1635019

IF (Tsout>0) THEN

\[ \begin{align*}
TsoutK &= Tsout + 273.15 \\
P_{s2} &= \exp(C8/TsoutK+C9+C10*TsoutK+C11*(TsoutK)^2+C12*(TsoutK)^3+C13*\ln(TsoutK)) \\
ws2 &= 0.62188*P_{s2}/(P_{readN}-P_{s2})
\end{align*}\]

ELSE

\[ \begin{align*}
TsoutK &= Tsout + 273.15 \\
P_{s2} &= \exp(C1/TsoutK+C2+C3*TsoutK+C4*(TsoutK)^2+C5*(TsoutK)^3+C6*(TsoutK)^4+C7*\ln(TsoutK)) \\
ws2 &= 0.62188*P_{s2}/(P_{readN}-P_{s2})
\end{align*}\]

ENDIF

ENDIF

END

PROCEDURE HUMCAL(Tout_{calc},P_{readN}: P_{calc}, wout_{calc})
\[
C8 = -5.8002206 \times 10^3 \\
C9 = -5.516256 \\
C10 = -4.8640239 \times 10^{-2} \\
C11 = 4.1764768 \times 10^{-5} \\
C12 = -1.4452093 \times 10^{-8} \\
C13 = 6.5459673 \\
C1 = -5.9745359 \times 10^3 \\
C2 = -5.1523058 \times 10^1 \\
C3 = -9.677843 \times 10^{-3} \\
C4 = 6.2215701 \times 10^{-7} \\
C5 = 2.0747825 \times 10^{-9} \\
C6 = -9.484024 \times 10^{-13} \\
C7 = 4.1635019
\]

IF (Tout_calc > 0) THEN

\[
\begin{align*}
T_{\text{out,calc}} &= T_{\text{out,calc}} + 273.15 \\
P_{\text{calc}} &= \text{Exp}(C8/T_{\text{out,calc}} + C9 + C10 \times T_{\text{out,calc}} + C11 \times (T_{\text{out,calc}})^2 + C12 \times (T_{\text{out,calc}})^3 + C13 \times \ln(T_{\text{out,calc}})) \\
w_{\text{out,calc}} &= 0.62188 \times P_{\text{calc}}/(P_{\text{read}} - P_{\text{calc}})
\end{align*}
\]

ELSE

\[
\begin{align*}
T_{\text{out,calc}} &= T_{\text{out,calc}} + 273.15 \\
P_{\text{calc}} &= \text{Exp}(C1/T_{\text{out,calc}} + C2 + C3 \times T_{\text{out,calc}} + C4 \times (T_{\text{out,calc}})^2 + C5 \times (T_{\text{out,calc}})^3 + C6 \times (T_{\text{out,calc}})^4 + C7 \times \ln(T_{\text{out,calc}})) \\
w_{\text{out,calc}} &= 0.62188 \times P_{\text{calc}}/(P_{\text{read}} - P_{\text{calc}})
\end{align*}
\]

ENDIF

END

PROCEDURE HUMCALTsm(Ts_m, P_readN: Psm, wsm)

\[
C8 = -5.8002206 \times 10^3 \\
C9 = -5.516256 \\
C10 = -4.8640239 \times 10^{-2} \\
C11 = 4.1764768 \times 10^{-5} \\
C12 = -1.4452093 \times 10^{-8} \\
C13 = 6.5459673 \\
C1 = -5.9745359 \times 10^3 \\
C2 = -5.1523058 \times 10^1 \\
C3 = -9.677843 \times 10^{-3} \\
C4 = 6.2215701 \times 10^{-7} \\
C5 = 2.0747825 \times 10^{-9} \\
C6 = -9.484024 \times 10^{-13} \\
C7 = 4.1635019
\]
IF (Ts_m>0) THEN

\[ T_{mK} = T_{m} + 273.15 \]
\[ P_{sm} = \exp(C8/T_{mK} + C9 + C10 \cdot T_{mK} + C11 \cdot (T_{mK})^2 + C12 \cdot (T_{mK})^3 + C13 \cdot \ln(T_{mK})) \]
\[ w_{sm} = 0.62188 \cdot P_{sm} / (P_{readN} - P_{sm}) \]

ELSE

\[ T_{mK} = T_{m} + 273.15 \]
\[ P_{sm} = \exp(C1/T_{mK} + C2 + C3 \cdot T_{mK} + C4 \cdot (T_{mK})^2 + C5 \cdot (T_{mK})^3 + C6 \cdot (T_{mK})^4 + C7 \cdot \ln(T_{mK})) \]
\[ w_{sm} = 0.62188 \cdot P_{sm} / (P_{readN} - P_{sm}) \]

ENDIF

END

{Fact}
\[ D_{lt} = 0.0064516 \text{ (Inside diameter of tube, m)} \]
\[ D_{ot} = 0.0072644 \text{ (Outside diameter of tube, m)} \]
\[ \delta = 7.60 \times 10^{-5} \text{ (Fin Thickness, m)} \]
\[ Ms = 0.0053721 \text{ (S_T=0.02165m)} \]
\[ Ls = 0.007314692 \text{ (S_L=0.0127)} \]

\[ WD = Ms \]
\[ LN = Ls \]

\[ Kt = 0.3387 \text{ (Kw/m-C, ARI410, C122000)} \]

\[ h_{fg} = 2501 \]
\[ D_{AB} = 0.26 \times 10^{-4} \text{ (Binary mass diffusion coefficient)} \]

\[ Le = k_{air} / (\rho_{air} \cdot C_{p_{air}} \cdot D_{AB}) \]

{Slit fin heat exchanger}

{***************Area Calculation***************}
\[ D_{coll} = D_{ot} + 2 \cdot \delta \text{ (Diameter of collar, m)} \]
\[ Nfs = Nf \times 2 - 1 \text{ (Number of Fin Surfaces)} \]
\[ Nt = (N_{tuberow} - 1) \times 10 + 9 \times (N_{tuberow} - 2) - 9 \times (N_{tuberow} - 3) \text{ (Number of Tube heat exchangers)} \]

{Heat Exchanger Dimensions}
\[ L_{fin} = N_{tuberow} / 2 \text{ (inch)} \]
\[ L_{f} = L_{fin} \times 0.0254 \text{ (Length of Fin, m)} \]
\[ H_{f} = 0.203 \text{ (Height of fin, m)} \]
\[ W_{dd} = 0.305 \text{ (Width of HX, m)} \]
\[ fs = fs_{mm} / 1000 \text{ (Heat exchanger fin spacing, m)} \]

{Channel Dimensions}
\[ L = 0.305 \text{(m)} \]
\[ D = L_{f} \text{(m)} \]
\[ A_{fs} = (H_{f} \times L_{f}) \times (Nt \times (D_{ot} + 2 \times \delta)^2) / 4 \text{ (Area per fin surface, m}^2) \]
\[ A_{tr} = (W_{dd} - Nf \times \delta) \times (D_{ot} + 2 \times \delta) \text{ (Area per tube row, m}^2) \]
\[ A_{f} = Nfs \times A_{fs} \text{ (Total fin area, m}^2) \]
\[ A_{t} = Nt \times A_{tr} \text{ (Total tube area, m}^2) \]
\[ A_{\text{tot}} = A_f + A_t + A_{\text{stotal}} \] (Total Area, m^2)

\[ A_f = H_f W_{dd} \] (Frontal Area, m^2)

\[ A_{\text{min}} = A_f - (\delta H_f N_f) - (10 D_{\text{coll}} (W_{dd} - N_f \delta)) \] (Minimum free flow area, m^2)

\[ A_i = \pi D_{it} L_t \] (Tube Length, m)

\[ B = A_{\text{tot}} / A_i \]

\[ L_t = W_{dd} N_t \] (Slit information)

\[ N_s = N_t \times 9 + 1 \] (Number of slit sets)

\[ A_s = 12 \times 0.00137414 \times 0.0014986 \] (Area of slit sets, m^2)

\[ A_{\text{stotal}} = N_s A_s \] (Total area of slit sets, m^2)

---

***************Coolant Side Calculation***************

\[ T_{in\_cF} = T_{in\_cC} \times 1.8 + 32 \]

\[ T_{out\_cF} = T_{out\_cC} \times 1.8 + 32 \]

\[ A_{ct} = (D_{it})^2 / 4 \pi \] (Cross Section Area of Tube, m^2)

\[ G_c = R_c / 10 / 1170.54 \times 0.003785 \] (Volumetric Flow Rate of Coolant, m^3/s)

\[ V_c = G_c / A_{ct} \]

\[ m_c = G_c / \rho_c \] (Kg/s)

\[ \rho_c = \left( -0.361004E^{-07} \times (T_{out\_cF})^3 \right) + \left( -0.281889E^{-04} \times (T_{out\_cF})^2 \right) - \left( 9.47314 \times (1E^{-03}) \right) \] (Density of Coolant, Kg/m^3)

\[ \nu_c = \left( -0.980059E^{-05} \times (T_{in\_cF})^3 \right) + \left( -0.232579E^{-02} \times (T_{in\_cF})^2 \right) - \left( 9.47314 \times (1E^{-03}) \right) \] (Viscosity of Coolant, Ns/m^2)

\[ k_c = \left( -0.207510E^{-07} \times (T_{in\_cF})^3 \right) + \left( -0.251828E^{-05} \times (T_{in\_cF})^2 \right) + \left( 0.259944E^{-03} \times (T_{in\_cF}) \right) + \left( 0.231444 / 1000 \right) \] (Conductivity of Coolant, W/mK)

\[ \nu_c = \left( -0.272010E^{-07} \times (T_{in\_cF})^3 \right) + \left( -0.370036E^{-05} \times (T_{in\_cF})^2 \right) + \left( 0.259944E^{-03} \times (T_{in\_cF}) \right) + \left( 0.231444 / 1000 \right) \] (Specific Heat of Coolant, KJ/KgK)

\[ \text{****Coolant-Side Heat Transfer Coefficient; The tube side heat transfer coefficient was determined by applying the correlation developed by Gnielinski[51] for turbulent flow.} \]

\[ \text{This correlation was chosen because it is applicable to the transitional Reynolds number of this study.} \]

\[ \text{****Reynolds Number Calculation****} \]

\[ \text{****Friction Factor of Coolant****} \]

\[ \text{****Coolant Prandtl Number****} \]

\[ \text{****Coolant Nusselt Number****} \]

\[ \text{****Tube Side Heat Transfer Coefficient****} \]

\[ h_i = \text{Nu}_c k_c / (D_{it}) \] (KW/m^2-K)

---

***************Air Side Calculation***************

\[ P_{\text{readN}} = P_{\text{readHg}} \times 3.38638815789 \] (conversion from inHg to KN/m^2)

\[ V_{\text{air}} = V_{\text{reading}} \times (273 + T_m) / (273 + 21.1) \times (101.325 / P_{\text{readN}}) \]

\[ T_m = (T_{\text{reading}} - 32) \times 1.8 \]
\[ T_{dp\_in\_C} = (T_{dp\_in\_F} - 32) / 1.8 \]
\[ T_{dp\_out\_C} = (T_{dp\_out\_F} - 32) / 1.8 \]

CALL HUM1(T_{dp\_in\_C}, P\_read\_N, P\_sat\_1, w1)
CALL HUMAIR1(Tin\_air, P\_read\_N, P\_air\_1, W\_air\_1)
CALL HUM2(T_{dp\_out\_C}, P\_read\_N, P\_sat\_2, w2)
CALL HUMAIR2(Tout\_air, P\_read\_N, P\_air\_2, W\_air\_2)

\[ RH\_in = P\_sat\_1 / P\_air\_1 \]
\[ RH\_out = P\_sat\_2 / P\_air\_2 \]

\{****Average*****\}
\[ T_{\_mair} = (Tin\_air + Tout\_air) / 2 \] \{Mean air temperature\}
\[ w_{\_mair} = (w1 + w2) / 2 \] \{Mean humidity ratio\}
\{****Air Properties*****\}
\[ \rho_{\_air} = \text{Density}(\text{AirH}_{20}, T = T_{\_mair}, P = P\_read\_N, w = w_{\_mair}) \] \{Kg/m\text{^3}\}
\[ \nu_{\_air} = \text{Viscosity}(\text{AirH}_{20}, T = T_{\_mair}, P = P\_read\_N, w = w_{\_mair}) \] \{Ns/m\text{^2}\}
\[ k_{\_air} = \text{Conductivity}(\text{AirH}_{20}, T = T_{\_mair}, P = P\_read\_N, w = w_{\_mair}) / 1000 \] \{KW/mK\}
\[ C_{p\_in\_air} = 1.006 + 1.845\times w_1 \] \{KJ/(KgK)\}
\[ C_{p\_out\_air} = 1.006 + 1.845\times w_2 \] \{KJ/(KgK)\}
\[ C_{p\_mair} = (C_{p\_in\_air} + C_{p\_out\_air}) / 2 \]
\[ h_{\_in\_air} = \text{Enthalpy}(\text{AirH}_{20}, T = Tin\_air, P = P\_read\_N, w = w_1) \] \{KJ/Kg\}
\[ h_{\_out\_air} = \text{Enthalpy}(\text{AirH}_{20}, T = Tout\_air, P = P\_read\_N, w = w_2) \] \{KJ/Kg\}

\{****Heat Transfer Rate*****\}
\[ m_{\_air} = V_{\_air} \times \rho_{\_air} \times A_{\_fr} \]
\[ V_{\_max} = (A_{\_fr} / A_{\_min}) \times \rho_{\_air} \times (\text{Density}(\text{AirH}_{20}, T = Tin\_air, P = P\_read\_N, w = w_1) / \rho_{\_air}) \times V_{\_air} \]
\[ G_{\_air} = V_{\_max} \times \nu_{\_air} \] \{Reynolds number of air based on outside diameter of tube\}
\[ q_{\_sens} = m_{\_air} \times C_{p\_mair} \times (Tin\_air - Tout\_air) \] \{KW\}
\[ q_{\_tot} = m_{\_air} \times (h_{\_in\_air} - h_{\_out\_air}) \] \{KW\}
\[ q_{\_ave} = (q_{\_tot} + q_{\_c}) / 2 \]
\[ BB = (q_{\_ave} - q_{\_tot}) / q_{\_ave} \]
\[ EB\_Ratio = (q_{\_ave} - q_{\_tot}) / q_{\_ave} \]
\[ EB\_Ratio\% = EB\_Ratio \times 100 \]
\[ BB\_Fu = (q_{\_c} - q_{\_ave}) / q_{\_ave} \]
\[ q_{\_c} = m_{\_c} \times C_{p\_c} \times (Tout\_c - Tin\_c) \]

\{****Reynolds Number*****\}
\[ Re\_Dot = (G_{\_air} \times (D_{\_ot})) / \nu_{\_air} \]
\[ Pr_{\_air} = C_{p\_mair} \times \nu_{\_air} / k_{\_air} \]
\[ Nu\_Dot = h_{\_wet} \times D_{\_ot} / k_{\_air} \]
\[ St_{\_air} = Nu\_Dot / (Re\_Dot \times Pr_{\_air}) \]

\{************Fin Efficiency Calculation************\}
\[ Ts\_m = (Tin + Tsout) / 2 \]

\{Sector method with conduction\}
\[ m_{\_pp} = 1.6896666098 + 0.045380979578 \times Ts\_m + 0.0012953947756 \times Ts\_m \times 2 + 2.8112344453e-05 \times Ts\_m \times 3 - 7.374055002e-08 \times Ts\_m \times 4 + 1.10049100756e-08 \times Ts\_m \times 5 \]
\[ h_{\_eff} = m_{\_pp} \times h_{\_wet} \times C_{p\_mair} \]
\[ m^2 = (2 \times h_{\_eff} / Kt \times \delta) \times 0.5 \]
\[ r_{if} = \frac{(D_{ot} + 2 \cdot \delta_{t})}{2} \] (Equivalent inner radius for fins with collars touching adjacent fin)

Nsectors = 4

{Counter side}

Duplicate \( n = 1, \) Nsectors

\{for zones 1, 4, 5 and 8\}

\[ \text{Sh}[n] = 0.00546608 \]

\[ R[n] = r_{if} \times ((2 \cdot n - 1)/(2 \cdot Nsectors))^{2} + (LN/WD)^{2})^{0.5} + \text{Sh}[n] \]

\[ (R[n]) = ((WD/r_{if})^{2} + ((2 \cdot n - 1)/(2 \cdot Nsectors))^{2} + (LN/WD)^{2})^{0.5} \]

\[ S[n] = r_{if}^{2}/2 \times ((R[n]^{2}) - 1) \times (\arctan(n \cdot WD/Nsectors/LN) - \arctan((n - 1) \cdot WD/Nsectors/LN)) \times \pi/180 \]

\[ \text{Rho}[n] = (R[n] - 1) \times (1 + 0.35 \times \ln(R[n])) \]

\[ H[n] = r_{if} \times \text{Rho}[n] \]

\[ \text{Eff}[n] = \tanh(m2 \times H[n]) \times \cos(0.1 \times m2 \times H[n])/(m2 \times H[n]) \]

\[ \text{Num}[n] = 4 \times \text{Eff}[n] \times \text{S}[n] \]

\[ \text{Den}[n] = 4 \times \text{S}[n] \]

\{for zones 2, 3, 6 and 7\}

\[ R[2 \times Nsectors + 1 - n] = (WD/r_{if}) \times ((2 \cdot n - 1)/(2 \cdot Nsectors))^{2} + (LN/WD)^{2}) + 1)^{0.5} \]

\[ S[2 \times Nsectors + 1 - n] = r_{if}^{2}/2 \times ((R[2 \times n + 1 - n]^{2}) - 1) \times (\arctan(n \cdot LN/Nsectors/WD) - \arctan((n - 1) \cdot LN/Nsectors/WD)) \times \pi/180 \]

\[ \text{Rho}[2 \times Nsectors + 1 - n] = (R[2 \times Nsectors + 1 - n] - 1) \times (1 + 0.35 \times \ln(R[2 \times Nsectors + 1 - n])) \]

\[ H[2 \times Nsectors + 1 - n] = r_{if} \times \text{Rho}[2 \times Nsectors + 1 - n] \]

\[ \text{Eff}[2 \times Nsectors + 1 - n] = \tanh(m2 \times H[2 \times Nsectors + 1 - n]) \times \cos(0.1 \times m2 \times H[2 \times Nsectors + 1 - n])/(m2 \times H[2 \times Nsectors + 1 - n]) \]

\[ \text{Num}[2 \times Nsectors + 1 - n] = 4 \times \text{Eff}[2 \times Nsectors + 1 - n] \times \text{S}[2 \times Nsectors + 1 - n] \]

\[ \text{Den}[2 \times Nsectors + 1 - n] = 4 \times \text{S}[2 \times Nsectors + 1 - n] \]

End

\[ \text{Num} = \text{SUM} (\text{Num}[n], \ n = 1, \text{Nsectors}) + \text{SUM} (\text{Num}[2 \times \text{Nsectors} + 1 - n], \ n = 1, \text{Nsectors}) \]

\[ \text{Den} = \text{SUM} (\text{Den}[n], \ n = 1, \text{Nsectors}) + \text{SUM} (\text{Den}[2 \times \text{Nsectors} + 1 - n], \ n = 1, \text{Nsectors}) \]

\[ \Phi = \text{Num}/\text{Den} \]

\[ \text{Eta} = (\Phi \times \text{Af} + \text{At})/\text{A}_t \text{ot} \] {Check}

\[ \text{Rf} = (1 - \text{Eta})/\text{h}_e/f/\text{A}_t \text{ot} \] {check}

\[ R_t = \ln(D_{ot}/D_{it})/2/\pi/K_t/L_t \] {Check}

\[ Rm = R_t + \text{Rf} \]

\[ \text{Ri} = 1/h_i/A_i \]

\{Heat Transfer Coefficient Calculation\}

\[ C1 = (Rm + \text{Ri})/C_p_{mair} \times h_{wet} \times A_{tot} \]

\[ C1 \times (h_{in\_air}-h_{s1}) = T_{sin}-T_{in\_c} \]

\[ CC2 \times (h_{out\_air}-h_{s2}) = T_{out}-T_{out\_c} \]

CALL HUMSAT1(Tsin, P_readN; P_s1, ws1)

CALL HUMSAT2(Tout, P_readN; P_s2, ws2)

{ws1 = HumRat(AirH2O, T = Tsin, P = 101.325, R = 1)}
\[ h_{s1} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_{\text{in}}, P=P_{\text{readN}}, w=w_{s1}) \]
\[ w_{s2} = \text{HumRat}(\text{AirH}_2\text{O}, T=T_{\text{out}}, P=101.325, R=1) \]
\[ h_{s2} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_{\text{out}}, P=P_{\text{readN}}, w=w_{s2}) \]
\[ \text{CALL HUMCALT}(\text{m}_s, \text{P}_\text{readN}; \text{P}_s, w_{s3}) \]
\[ h_{s3} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_s, P=P_{\text{readN}}, w=w_{s3}) \]
\[ m_{\text{water}} = \text{Volume}\times0.000001/\text{Time}\times\text{DENSITY}(\text{Water}, T=T_s, P=P_s) \]
\[ q_{\text{latent}} = m_{\text{water}}\times h_{s3} \]
\[ q_{\text{total}} = q_{\text{tot}} - q_{\text{latent}} \]
\[ q_{\text{totalave}} = \frac{q_{\text{tot}} + q_c}{2} \]
\[ \text{EB}_{\text{new}}\% = \frac{(q_{\text{totalave}} - q_{\text{total}})}{q_{\text{totalave}}} \times 100 \]

\{\textit{Counter Flow}\}

\[ h_{n1} = \frac{(h_{\text{in-air}} - h_{s1}) - (h_{\text{out-air}} - h_{s2})}{\text{lmhd}} \]
\[ h_{\text{in-air}} = (\exp(h_{n1})) \times (h_{\text{out-air}} - h_{s2}) + h_{s1} \]
\[ t_{n1} = \frac{(T_{\text{in-air}} - T_{\text{out-cC}}) - (T_{\text{out-air}} - T_{\text{in-cC}})}{\text{lmtd}} \]
\[ T_{\text{in-air}} = (\exp(t_{n1})) \times (T_{\text{out-air}} - T_{\text{in-cC}}) + T_{\text{out-cC}} \]
\[ q_{\text{sens_calc}} = h_{\text{wet}} \times A_{\text{tot}} \times \text{lmtd} \]
\[ T_{\text{out_calc}} = T_{\text{in-air}} - (q_{\text{sens_calc}} / m_{\text{air}}/C_p_{\text{mair}}) \]
\[ \text{CALL HUMCAL}(T_{\text{out_calc}}, P_{\text{readN}}; P_{\text{calc}}, w_{\text{out_calc}}) \]
\[ h_{\text{out-air}} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_{\text{out_calc}}, P=P_{\text{readN}}, w=w_{\text{out_calc}}) \]
\[ R_l_{\text{calc}} = \text{RelHum}(\text{AirH}_2\text{O}, T=T_{\text{out_calc}}, P=P_{\text{readN}}, w=w_{\text{out_calc}}) \]
\[ q_{\text{tot}} = A_{\text{tot}} \times \text{lmhd} / C_p_{\text{mair}} \times h_{\text{wet}} \]
\[ R_1 = (w_1 - w_{s1}) \times (1/\text{Le} \times (2/3) - 1) \times h_{\text{fg}} / (h_{\text{in-air}} - h_{s1}) \]
\[ R_2 = (w_2 - w_{s2}) \times (1/\text{Le} \times (2/3) - 1) \times h_{\text{fg}} / (h_{\text{out-air}} - h_{s2}) \]
\[ b_1 = (h_{\text{in-air}} - h_{s1}) / C_p_{\text{mair}} / (T_{\text{in-air}} - T_{\text{sin}}) - 1 \]
\[ b_2 = (h_{\text{out-air}} - h_{s2}) / C_p_{\text{mair}} / (T_{\text{out-air}} - T_{\text{sout}}) - 1 \]

\{\textit{f factor}\}

\[ j = S_{\text{air}} \times (P_{\text{air}}) \times (2/3) \]
\[ \text{Rho}_{\text{air-12}} = (\text{Density}(\text{AirH}_2\text{O}, T=T_{\text{air}}, P=P_{\text{readN}}, w=w_{1}) / \text{Density}(\text{AirH}_2\text{O}, T=T_{\text{air}}, P=P_{\text{readN}}, w=w_{2})) \]
\[ \text{Ratio}_m_{\text{t}} = A_{\text{min}} / A_{\text{tot}} \]
\[ \text{Ratio}_m_{\text{f}} = A_{\text{min}} / A_{\text{fr}} \]
\[ \text{DelP}_{\text{air}} = \text{DelP}_{\text{air}} \times 249,08891 \ [N/m^2] \]
\[ f = \left((2 \times \text{DelP}_{\text{air}} \times \text{Rho}_{\text{air}}) / (G_{\text{air}})^2\right) \times (\text{Ratio}_m_{\text{t}} - 1) \times (1 + (\text{Ratio}_m_{\text{f}})^2) \times (\text{Rho}_{\text{air-12}} - 1) \times (\text{Ratio}_m_{\text{t}}) \times (\text{Rho}_{\text{air}} / (\text{Density}(\text{AirH}_2\text{O}, T=T_{\text{air}}, P=P_{\text{readN}}, w=w_{1}))) \]
Table A.4 Known variables

<table>
<thead>
<tr>
<th>Variables</th>
<th>Physical meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_t$</td>
<td>Inside diameter of tube, m</td>
</tr>
<tr>
<td>$D_{ot}$</td>
<td>Outside diameter of tube, m</td>
</tr>
<tr>
<td>$\delta_f$</td>
<td>Fin Thickness, m</td>
</tr>
<tr>
<td>$W_d$</td>
<td>Width of heat exchanger, m</td>
</tr>
<tr>
<td>$H_f$</td>
<td>Height of fin, m</td>
</tr>
<tr>
<td>$S_T$</td>
<td>Transverse tube spacing, m</td>
</tr>
<tr>
<td>$S_L$</td>
<td>Longitudinal tube spacing, m</td>
</tr>
</tbody>
</table>

Table A.5 Definitions of basic calculated parameter

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G_{air}$</td>
<td>$V_{max} \cdot \rho_{air}$</td>
</tr>
<tr>
<td>$V_{max}$</td>
<td>$V_{air} \left( \frac{A_{fr}}{A_{min}} \right) \cdot \rho_{air,in} \cdot \rho_{air}$</td>
</tr>
<tr>
<td>$Re_{Dh}$</td>
<td>$\frac{G_{air} \cdot D_h}{\mu_{air}}$</td>
</tr>
<tr>
<td>$Re_{Deoll}$</td>
<td>$\frac{G_{air} \cdot D_{eoll}}{\mu_{air}}$</td>
</tr>
<tr>
<td>$Nu_{Deoll}$</td>
<td>$\frac{h \cdot D_{eoll}}{\kappa_{air}}$</td>
</tr>
<tr>
<td>$j$</td>
<td>$St \cdot Pr^{\frac{2}{3}}$</td>
</tr>
<tr>
<td>$St$</td>
<td>$\frac{Nu}{Re \cdot Pr} = \frac{h}{G_{air} \cdot C_{p,air}}$</td>
</tr>
<tr>
<td>$Pr_{air}$</td>
<td>$\frac{C_{p,air} \cdot \mu_{air}}{\kappa_{air}}$</td>
</tr>
<tr>
<td>$Pr_c$</td>
<td>$\frac{C_{p,c} \cdot \mu_{c}}{\kappa_{c}}$</td>
</tr>
<tr>
<td>$Le$</td>
<td>$\frac{Sc}{pr} = \frac{\kappa_{air}}{\rho_{air} \cdot C_{p,air} \cdot D_{AB}}$</td>
</tr>
<tr>
<td>$f$</td>
<td>$\frac{2 \Delta P_{HX} \cdot P_{air}}{G_{air}^2 \cdot A_{min}} \cdot \frac{A_{min}}{A_{tot}} \cdot \left( 1 + \sigma^2 \left( \frac{\rho_{air,in}}{\rho_{air,out}} - 1 \right) \frac{A_{min}}{A_{tot}} \left( \frac{\rho_{air}}{\rho_{air,in}} \right) \right)$</td>
</tr>
</tbody>
</table>
In the development of correlations for \( j \) and \( f \) factors, several dimensionless groups such as the Reynolds number (Re), the Nusselt number (Nu), the Eckert number (E), and the Prandtl number (Pr) were encountered. These dimensionless groups are useful because they reduce the number of final independent parameters for the correlations. Buckingham-II Theorem [39], a dimensional analysis, was used in order to determine non-dimensional groups obtained from the set of parameters for the correlations.

**B.1 Dimensional Analysis of \( j \) Factor**

First step of the Buckingham-II Theorem is predicting geometric and flow parameters that will influence the \( j \) factor correlation. Those parameters are then expressed in the form of fundamental dimensions using mass (M), length (L), time (t), and temperature (T). The parameters and their dimensions are shown in Table B.1 and the locations of the geometric parameters on fin surface are shown in Figure B.2.

Total of 15 parameters was predicted for the \( j \) factor correlation. Due to the complexity of the slit-fin geometry, the dimensions of each slit set were neglected. The Buckingham-II Theorem states that the number of dimensionless groups can be determined by the difference between the number of parameters (\( n \)) predicted and the number of fundamental dimensions (\( m \)) used to define the dimensions for all the parameters. In this case, 11 dimensionless groups were obtained. Therefore, the \( j \) factor correlation can be written as Equation B.1.

\[
\phi(\Pi_1, \Pi_2, \Pi_3, \cdots, \Pi_{n-m}) = 0
\]  

\( (B.1) \)

where

\( n = \) number of parameters

\( m = \) number of fundamental dimensions

The next step of the Buckingham-II Theorem is determining the repeating parameters. The number of repeating parameters should be equal to the number of
fundamental dimensions. Each repeating parameter should have different net dimensions. For example, an area \( (L^2) \) can not be chosen as a repeating parameter along with the length \( (L) \). Four parameters, \( D_{coll} \), \( C_{pair} \), \( \mu_{air} \), and \( \rho_{air} \) were chosen as the repeating parameters.

The third step is determining \( \Pi \) groups. The \( \Pi \) groups are the dimensionless groups and can be replaced by \( M^0L^0T^0\). The repeating parameters and each of the other parameters are combined to form the \( \Pi \) groups. The equations for the \( \Pi \) groups are shown in Table B.2. The exponents of each equation can be solved simultaneously by equating them. The values of exponents of \( M \), \( L \), \( t \) and \( T \) for each \( \Pi \) and the final form of \( \Pi \) groups are shown in Table B.3.

The fourth step is expressing the \( \Pi \) groups in the form of the dimensionless groups. The \( \Pi \) groups can be multiplied with the known dimensionless groups since a multiplication of two non-dimensional parameters yields a non-dimensional parameter.

\[
\Pi_1 = \frac{\kappa_{air}}{C_{pair} \mu_{air}} = \frac{1}{Pr} \\
\Pi_2 = \frac{D_{coll}^2 C_{pair} \rho_{air}^2 \Delta T}{\mu_{air}^2} \left( \frac{\mu_{air}}{V_{air} D_{coll} \rho_{air}} \right)^2 = \frac{C_{pair} \Delta T}{V_{air}^2} = \frac{1}{E} \\
\Pi_3 = \frac{m}{D_{coll} \mu_{air}} \frac{D_{coll}}{H} \frac{D_{coll}}{W} = \frac{m D_{coll}}{H W \mu_{air}} = \frac{V_{air} D_{coll} \rho_{air}}{\mu_{air}} = \text{Re}_{D_{coll}} \quad \text{where} \quad m = V_{air} H W \rho_{air} \\
\Pi_4 = \frac{D_{coll} \rho_{air}^2 q}{\mu_{air}^3} = \left( \frac{D_{coll} \rho_{air}^2 h A_{tot} \Delta T}{\mu_{air}^3} \right) \left( \frac{\mu_{air}}{V_{air} D_{coll} \rho_{air}} \right)^2 \left( \frac{C_{pair} \mu_{air}}{\kappa_{air}} \right) \left( \frac{V_{air}^2}{C_{pair} \Delta T} \right) = \frac{h A_{tot}}{\kappa_{air} D_{coll}} = \frac{N u A_{tot}}{D_{coll}} \\
\text{where} \quad q = h A_{tot} \Delta T
\]

The final step of the Buckingham-\( \Pi \) Theorem is rearranging the \( \Pi \) groups in the form of functional relationship as follows.

\[
\Pi_4 = f(\Pi_1, \Pi_2, \Pi_3, \Pi_5 \cdots, \Pi_{11})
\]
Replacing \( \Pi \) groups with dimensionless groups,

\[
\frac{NuA_{int}}{D_{coll}^2} = f\left(\frac{1}{Pr}, \frac{1}{E}, \frac{Re_{coll}}{D_{coll}}, \frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right)
\]  

(B.13)

Since

\[
\frac{A_{int}}{D_{coll}^2} = f\left(\frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right)
\]

the equation B.13 can be written as the following.

\[
Nu = f\left(\frac{1}{Pr}, \frac{1}{E}, \frac{Re_{coll}}{D_{coll}}, \frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right)
\]

The relationship between the Nusselt number and the \( j \) factor is

\[
\frac{Nu}{RePr^{\frac{1}{3}}} = j \]

Therefore, \( j \) factor can be expressed as follows.

\[
j = f\left(\frac{1}{E}, \frac{Re_{coll}}{D_{coll}}, \frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right)
\]

The Eckert number and the constant dimensionless groups for geometric parameters were neglected for the \( j \) factor correlation, and \( L \) was expressed in terms of number of tube rows (\( N_{tr} \)) and longitudinal tube spacing.
Equation B.14 shows the finalized functional relationship of $j$ factor.

$$j = f\left(\frac{Re_{De\text{coll}}}{D_{\text{coll}}}, \frac{f_s}{N_r S_n}, \frac{N_r S_n}{D_{\text{coll}}}, \frac{D_{\text{coll}}}{D_{\text{coll}}}\right) \quad (B.14)$$

The proposed $j$ factor correlation is,

$$j = C\left(\frac{Re_{De\text{coll}}}{D_{\text{coll}}}\right)^a \left(\frac{f_s}{D_{\text{coll}}}\right)^b \left(\frac{S_n N_r}{D_{\text{coll}}}\right)^c$$

where $C$, $a$, $b$, and $c$ are constant

The constant and the exponents were calculated by multiple regression technique using Engineering Equation Solver (EES).

### B.2 Dimensional Analysis of $f$ Factor

The procedure and the parameters involved in the dimensional analysis for $f$ factor are very similar to those of $j$ factor. The parameters $q$, $C_{p, \text{air}}$, $\kappa_{\text{air}}$ and $\Delta T$ can be neglected for the dimensional analysis of the $f$ factor because they do not influence $f$ factor. Instead, a new parameter, $\Delta P$ is added. There are total of 12 parameters with 3 repeating parameters and 3 fundamental dimension. The equations of $\Pi$ groups for $f$ factor are shown in Table B.4 and the values of exponents for each $\Pi$ and the final form of $\Pi$ groups are shown in Table B.5.

The $\Pi$ groups can be expressed in terms of the dimensionless group.

$$\Pi_1 = \left(\frac{D_{\text{coll}}^2 \rho_{\text{air}}}{\mu_{\text{air}}^2}\right) \left(\frac{\mu_{\text{air}}}{V_{\text{air}} D_{\text{coll}} \rho_{\text{air}}}\right)^2 \left(\frac{D_{\text{coll}}}{L}\right) \frac{\Delta P D_{\text{coll}}}{V_{\text{air}}^2 \rho_{\text{air}} L}$$

$$\Pi_2 = \frac{m}{D_{\text{coll}} H \rho_{\text{air}}} \frac{D_{\text{coll}}}{W} = \frac{m D_{\text{coll}}}{H W \rho_{\text{air}}} = \frac{V_{\text{air}} D_{\text{coll}} \rho_{\text{air}}}{\mu_{\text{air}}} = Re_{De\text{coll}}$$

where $m = V_{\text{air}} H W \rho_{\text{air}}$
The $\Pi$ groups for $f$ factors then are arranged in the form of functional relationship,

$$\Pi_1 = f(\Pi_2, \Pi_3, \Pi_4 \ldots, \Pi_9)$$

and therefore,

$$\frac{\Delta P D_{coll}}{V_{air}^2 \rho_{air} L} = f\left(\frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right) \quad (B.24)$$

The relationship between $f$ factor and $\Pi_1$ is,

$$f = \frac{2\Delta P D_{coll}}{V_{air}^2 L}$$

and, $f$ factor can be expressed as the following.

$$f = f\left(\frac{\delta_f}{D_{coll}}, \frac{f_s}{D_{coll}}, \frac{H}{D_{coll}}, \frac{W}{D_{coll}}, \frac{L}{D_{coll}}, \frac{S_T}{D_{coll}}, \frac{S_L}{D_{coll}}\right)$$

By neglecting constant dimensionless groups and replacing $L$ to $N_T S_L$, the final form of the functional relationship is,

$$f = f\left(\frac{f_s}{D_{coll}}, \frac{N_T S_L}{D_{coll}}\right) \quad (B.25)$$

The proposed $f$ factor correlation is,

$$f = C\left(\frac{\delta_f}{D_{coll}}\right)^a \left(\frac{f_s}{D_{coll}}\right)^b \left(\frac{S_L N_T}{D_{coll}}\right)^c$$

where $C$, $a$, $b$, and $c$ are constant.
Table B.1 Parameters involved in \( j \) and \( f \) factor correlations

<table>
<thead>
<tr>
<th>Geometric parameter of heat exchanger</th>
<th>Dimension</th>
<th>Key parameter</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>( D_{\text{coll}} )</td>
<td>( L )</td>
<td>( \kappa_{\text{air}} )</td>
<td>( \frac{ML}{t^3T} )</td>
</tr>
<tr>
<td>( \delta_f )</td>
<td>( L )</td>
<td>( \mu_{\text{air}} )</td>
<td>( \frac{M}{Lt} )</td>
</tr>
<tr>
<td>( f_s )</td>
<td>( L )</td>
<td>( C_{p,\text{air}} )</td>
<td>( \frac{L^2}{t^2T} )</td>
</tr>
<tr>
<td>( H )</td>
<td>( L )</td>
<td>( \rho_{\text{air}} )</td>
<td>( \frac{M}{L^2} )</td>
</tr>
<tr>
<td>( W )</td>
<td>( L )</td>
<td>( \Delta T )</td>
<td>( T )</td>
</tr>
<tr>
<td>( L )</td>
<td>( L )</td>
<td>( \cdot m )</td>
<td>( \frac{M}{t} )</td>
</tr>
<tr>
<td>( S_T )</td>
<td>( L )</td>
<td>( q )</td>
<td>( \frac{ML^3}{t^5} )</td>
</tr>
<tr>
<td>( S_L )</td>
<td>( L )</td>
<td>( \Delta P )</td>
<td>( \frac{M}{Lt^2} )</td>
</tr>
</tbody>
</table>

Figure B.1 Location of each geometric parameter on fin surface
Table B.2 Equations of $\Pi$ groups for $j$ factor

<table>
<thead>
<tr>
<th>Equation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Pi_1 = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \kappa_{air} = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d \left( \frac{ML}{t^3} \right)$</td>
<td>(B.2)</td>
</tr>
<tr>
<td>$\Pi_2 = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \Delta T = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d (T)$</td>
<td>(B.3)</td>
</tr>
<tr>
<td>$\Pi_3 = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \cdot m = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d \left( \frac{M}{l} \right)$</td>
<td>(B.4)</td>
</tr>
<tr>
<td>$\Pi_4 = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \cdot q = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d \left( \frac{ML^2}{t^3} \right)$</td>
<td>(B.5)</td>
</tr>
<tr>
<td>$\Pi_5 = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \cdot \delta_f = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d (L)$</td>
<td>(B.6)</td>
</tr>
<tr>
<td>$\Pi_{11} = D_{coll}^a C_{p,air}^b \mu_{air}^c \rho_{air}^d \cdot S_k = (L)^a \left( \frac{L^2}{t^2 T} \right)^b \left( \frac{M}{L t} \right)^c \left( \frac{M}{L^3} \right)^d (L)$</td>
<td>(B.12)</td>
</tr>
</tbody>
</table>
Table B.3 Exponents of M, L, t, and T and final form of Π groups for j factor

<table>
<thead>
<tr>
<th>Equation</th>
<th>Exponents</th>
<th>Π</th>
</tr>
</thead>
<tbody>
<tr>
<td>B.2</td>
<td>a = 0, b = −1, c = −1, d = 0</td>
<td>( \Pi_1 = \frac{\kappa_{\text{air}}}{C_{\text{p,air}} \mu_{\text{air}}} )</td>
</tr>
<tr>
<td>B.3</td>
<td>a = 2, b = 1, c = −2, d = 2</td>
<td>( \Pi_2 = \frac{D_{\text{coll}}^2 C_{\text{p,air}} \rho^2 \Delta T}{\mu_{\text{air}}^2} )</td>
</tr>
<tr>
<td>B.4</td>
<td>a = −1, b = 0, c = −1, d = 0</td>
<td>( \Pi_3 = \frac{m}{D_{\text{coll}} \mu_{\text{air}}} )</td>
</tr>
<tr>
<td>B.5</td>
<td>a = 1, b = 0, c = −3, d = 2</td>
<td>( \Pi_4 = \frac{D_{\text{coll}} \rho_{\text{air}}^2 q}{\mu_{\text{air}}^2} )</td>
</tr>
<tr>
<td>B.6</td>
<td>a = −1, b = 0, c = 0, d = 0</td>
<td>( \Pi_5 = \frac{\delta_f}{D_{\text{coll}}} )</td>
</tr>
<tr>
<td>B.12</td>
<td>a = −1, b = 0, c = 0, d = 0</td>
<td>( \Pi_{11} = \frac{S_L}{D_{\text{coll}}} )</td>
</tr>
</tbody>
</table>

Table B.4 Equations of Π groups for f factor

<table>
<thead>
<tr>
<th>Non-dimensional Π-group Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Pi_1 = D_{\text{coll}}^a \mu_{\text{air}}^b \rho_{\text{air}}^c \Delta P = (L)^a \left( \frac{M}{Lt} \right)^b \left( \frac{M}{L^3} \right)^c \left( \frac{M}{Lt^2} \right) )</td>
</tr>
<tr>
<td>( \Pi_2 = D_{\text{coll}}^a \mu_{\text{air}}^b \rho_{\text{air}}^c m = (L)^a \left( \frac{M}{Lt} \right)^b \left( \frac{M}{L^3} \right)^c \left( \frac{M}{t} \right) )</td>
</tr>
<tr>
<td>( \Pi_3 = D_{\text{coll}}^a \mu_{\text{air}}^b \rho_{\text{air}}^c \delta_f = (L)^a \left( \frac{M}{Lt} \right)^b \left( \frac{M}{L^3} \right)^c (L) )</td>
</tr>
<tr>
<td>( \Pi_5 = D_{\text{coll}}^a \mu_{\text{air}}^b \rho_{\text{air}}^c \delta_f = (L)^a \left( \frac{M}{Lt} \right)^b \left( \frac{M}{L^3} \right)^c \left( \frac{M}{t} \right) )</td>
</tr>
</tbody>
</table>

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Table B.5 Exponents of M, L, t and T and final form of \( \Pi \) groups for \( f \) factor

<table>
<thead>
<tr>
<th>Equation</th>
<th>Exponents</th>
<th>( \Pi )</th>
</tr>
</thead>
<tbody>
<tr>
<td>B.15</td>
<td>( a = 2, b = -2, c = 1 )</td>
<td>( \Pi_1 = \frac{D_{\text{coll}} \rho_{\text{air}} \Delta P}{\mu_{\text{air}}^2} )</td>
</tr>
<tr>
<td>B.16</td>
<td>( a = -1, b = -1, c = 0 )</td>
<td>( \Pi_2 = \frac{m}{D_{\text{coll}} \mu_{\text{air}}} )</td>
</tr>
<tr>
<td>B.17</td>
<td>( a = -1, b = 0, c = 0 )</td>
<td>( \Pi_3 = \frac{\delta_f}{D_{\text{coll}}} )</td>
</tr>
<tr>
<td>B.23</td>
<td>( a = -1, b = 0, c = 0 )</td>
<td>( \Pi_9 = \frac{S_L}{D_{\text{coll}}} )</td>
</tr>
</tbody>
</table>
APPENDIX C - UNCERTAINTY ANALYSIS

Uncertainties in the results of condensate retention and heat transfer experiments depend on the uncertainties in experimental measurements. Uncertainties in experimental measurements are provided by either from manufacturer of the instrument or by some other type of calibration procedure. In this appendix, the uncertainties of all experimental measurements as well as the propagation of these uncertainties are presented.

C.1 Experimental Measurement Uncertainty

A total of 5 experimental measurements were taken during the retention experiment and a total of 13 experimental measurements were taken for heat transfer experiment. The uncertainties of the experimental measurements are summarized in Table C.1 and the uncertainties in air properties are shown in Table C.2.

Table C.1 Uncertainties in experimental measurement

<table>
<thead>
<tr>
<th>Experimental measurement</th>
<th>Uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_{\text{water}}$</td>
<td>$\pm 0.1 \text{g}$</td>
</tr>
<tr>
<td>$T$</td>
<td>$\pm 1 \text{s}$</td>
</tr>
<tr>
<td>$V_{\text{air}}$</td>
<td>$\pm 1 %$</td>
</tr>
<tr>
<td>$T_{\text{air, in}}$</td>
<td>$\pm 1 \text{°C}$</td>
</tr>
<tr>
<td>$T_{\text{air, out}}$</td>
<td>$\pm 0.5 \text{°C}$</td>
</tr>
<tr>
<td>$T_{\text{c, in}}$</td>
<td>$\pm 0.3 \text{°C}$</td>
</tr>
<tr>
<td>$T_{\text{c, out}}$</td>
<td>$\pm 0.3 \text{°C}$</td>
</tr>
<tr>
<td>$T_{\text{dp, in}}$</td>
<td>$\pm 0.2 \text{°C}$</td>
</tr>
<tr>
<td>$T_{\text{dp, out}}$</td>
<td>$\pm 0.2 \text{°C}$</td>
</tr>
<tr>
<td>$\Delta P_{\text{HX}}$</td>
<td>$\pm 0.0005 \text{ in water}$</td>
</tr>
<tr>
<td>$R_c$</td>
<td>$\pm 0.5 %$</td>
</tr>
<tr>
<td>$P_{\text{air}}$</td>
<td>0.4%</td>
</tr>
</tbody>
</table>
Table C.2 Uncertainties in air properties

<table>
<thead>
<tr>
<th>Air properties</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_{\text{air}}$</td>
<td>$\pm 3%$</td>
</tr>
<tr>
<td>$\mu_{\text{air}}$</td>
<td>$\pm 3%$</td>
</tr>
<tr>
<td>$C_{p, \text{air}}$</td>
<td>$\pm 1%$</td>
</tr>
<tr>
<td>$\kappa_{\text{air}}$</td>
<td>$\pm 3%$</td>
</tr>
</tbody>
</table>

C.2 Uncertainties in Data Reduction

The uncertainties in data reduction are determined using techniques by Kline and McClintock [40]. Equation C.1 shows the method of deriving the propagating uncertainty with the uncertainties in the experimental measurements.

$$W_X = \left[ \left( \frac{\partial X}{\partial Y_1} W_1 \right)^2 + \left( \frac{\partial X}{\partial Y_2} W_2 \right)^2 + \ldots + \left( \frac{\partial X}{\partial Y_n} W_n \right)^2 \right]^\frac{1}{2}$$  

(C.1)

Where

- $W_n$ = uncertainty of variable $n$, $n=1,2,3,\ldots,n$
- $W_X$ = propagating uncertainty in result
- $\frac{\partial X}{\partial Y_n}$ = partial derivative of result with respect to variable, $n$

C.2.1 Uncertainty in $V_{\text{max}}$

The uncertainty of the frontal air velocity is shown in Table C.1. The uncertainty in $V_{\text{max}}$ is calculated approximately 4.74% with an uncertainty in $A_{fr}$ of 1% and $A_{min}$ of 1.47%. The formula for the uncertainty in $V_{\text{max}}$ is shown in C.2.

$$\frac{W_{V_{\text{max}}}}{V_{\text{max}}} = \left[ \left( \frac{W_{V_{\text{air}}}}{V_{\text{air}}} \right)^2 + \left( \frac{W_{A_{fr}}}{A_{fr}} \right)^2 + \left( \frac{W_{A_{min}}}{A_{min}} \right)^2 + \left( \frac{W_{\rho_{\text{air},in}}}{\rho_{\text{air},in}} \right)^2 + \left( \frac{W_{\rho_{\text{air}}}}{\rho_{\text{air}}} \right)^2 \right]^\frac{1}{2}$$  

(C.2)

C.2.2 Uncertainty in Air-Side Reynolds Number

The uncertainty in air-side Reynolds number based on tube diameter with collar is determined by applying the following formula.
The uncertainty of the Reynolds number based on collar diameter is approximately 7% with the mass velocity, G of 5.6%.

C.2.3 Uncertainty in Coolant Mass Flow Rate

Equation C.4 calculates the uncertainty in coolant mass flow rate. The uncertainty of the coolant density based on outlet coolant temperature is 2.7% and the uncertainty of the coolant mass flow rate is 2.8%.

\[
\frac{W}{m_c} = \left[ \left( \frac{W_{G_{air}}}{G_{air}} \right)^2 + \left( \frac{W_{\mu_{air}}}{\mu_{air}} \right)^2 \right]^{\frac{1}{2}}
\]

(C.3)

C.2.4 Uncertainty in Air-Side Friction Factor

The uncertainty in air-side friction factor is determined by equation C.5. The uncertainty in momentum effects is neglected because the momentum effects are relatively small compared to the friction factor. The uncertainty in \( A_{tot} \) is 1.15% and the uncertainty in air-side friction factor is determined to be 12%.

\[
\frac{W_f}{f} = \left[ \left( \frac{W_{\Delta P_{HX}}}{\Delta P_{HX}} \right)^2 + \left( 2 \frac{W_{G_{air}}}{G_{air}} \right)^2 + \left( \frac{W_{\mu_{air}}}{\mu_{air}} \right)^2 + \left( \frac{W_{A_{min}}}{A_{min}} \right)^2 + \left( \frac{W_{A_{tot}}}{A_{tot}} \right)^2 \right]^{\frac{1}{2}}
\]

(C.4)

C.2.5 Uncertainty in Air-Side Sensible Heat Transfer Coefficient

The uncertainty in air-side sensible heat transfer coefficient is calculated using C.6. An uncertainty in coolant-side sensible heat transfer coefficient \( h_i \) of 10% is used based on the Handbook of Single-Phase Convective Heat Transfer. The uncertainty in air-side sensible heat transfer coefficient is determined to be 11% with an uncertainty in air-side mass flow rate of 10%.
Uncertainty in Air-Side Sensible Nusselt Number

The uncertainty in air-side sensible Nusselt number is calculated based on the uncertainties of air-side sensible Nusselt number, collar diameter, and conductivity of air. With uncertainty in air-side heat transfer coefficient of 11\%, the uncertainty of Nusselt number is determined to be 11.8\%.

\[
\frac{W_{N_t_{coll}}}{N_t_{coll}} = \left[ \left( \frac{W_h}{h} \right)^2 + \left( \frac{W_{D_{coll}}}{D_{coll}} \right)^2 + \left( \frac{W_{\kappa_{air}}}{\kappa_{air}} \right)^2 \right]^{1/2}
\]  

(C.7)

C.2.7 Uncertainty in Sensible j factor

The uncertainty in sensible j factor can be calculated using Equation C.8. With uncertainty of 4.36\% for Pr, the uncertainty in sensible j factor is determined to be 13.13\%.

\[
\frac{W_j}{j} = \left[ \left( \frac{W_h}{h} \right)^2 + \left( \frac{W_{G_{air}}}{G_{air}} \right)^2 + \left( \frac{W_{C_{p,air}}}{C_{p,air}} \right)^2 + \left( \frac{W_{Pr}}{Pr} \right)^2 \right]^{1/2}
\]  

(C.8)
APPENDIX D - DATA COMPARISON

Present data are compared to the data provided by Samsung Electronic Co. LTD. Each of plain-fin-and-tube and slit-fin-and-tube heat exchangers with 1.3 mm and 1.5 mm fin spacing tested are compared. Experiments are conducted under two different conditions. The range of parameters for the experimental data provided by Samsung is shown in Table D.1.

Table D.1 Range of condition for air-side heat transfer performance

<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Approximate Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet air dry-bulb temperature</td>
<td>20.99 to 21.01 °C</td>
</tr>
<tr>
<td>Inlet wet-bulb temperature</td>
<td>15.98 to 16.05 °C</td>
</tr>
<tr>
<td>Face velocity</td>
<td>0.746 to 2.502 m/s</td>
</tr>
<tr>
<td>Water supply temperature</td>
<td>49.87 to 50.14 °C</td>
</tr>
</tbody>
</table>

Figures D.1 (a), (b), (c), and (d) show the comparison between the present \( f \) and \( j \) factor data and the data provided by Samsung. These figures show fairly good agreement for both \( f \) and \( j \) factors. Figures D.2 (a), (b), (c), and (d) show the data for slit-fin-and-tube heat exchanger with 1.3 mm fin spacing. The data of heat exchangers with different number of tube rows and coating are compared to the data from Samsung with same heat exchanger geometry. \( f \) factors show relatively good agreement, but the present \( j \) factors are higher than the data from Samsung at low Reynolds numbers. This disagreement is large and in contrast with the comparison of our data to the correlation of Nakayama and Xu [21] (see Chapter 4). Our \( j \) factor data were slightly higher than those of Nakayama and Xu's correlation for a dry slit-fin-and-tube heat exchanger; the Samsung data are much higher than ours at low Reynolds numbers. Figures D.3 (a), (b), and (c) present a comparison of slit-fin-and-tube heat exchanger with 1.5 mm fin spacing. The results of the slit-fin-and-tube heat exchangers with 1.5 mm fin spacing show better agreement than those with 1.3 mm fin spacing. Figures D.4 (a), (b), (c), and (d) plot the present \( j \) and \( f \) factor data of plain-fin-and-tube heat exchanger and those of Samsung data under dehumidifying conditions. Both the \( j \) and \( f \) factors are approximately 20% lower than those data provided by Samsung. This disagreement should again be viewed with the favorable comparison of our plain-fin data to results from the open literature in mind (see
Chapter 4. The same trends are found for the slit-fin-and-tube heat exchangers as shown in figures D.5 and D.6.

Some of the discrepancies between our results and those provided by Samsung may be due to differences in data reduction methods. We adopted a hybrid approach, based on ARI 410 as modified by Hong and Webb for plain-fins and as described in appendix A for slit-fins. However, the relatively close agreement between our data and other results makes it unlikely that large discrepancies between our results and Samsung can be solely due to data reduction methods. Other possibilities included problems with the test apparatus or procedure; however, our energy-balance results are good and it is unlikely that such problem would escape this redundant check. It is possible that the specimens tested by Samsung differed from those we used. Such differences might be due to manufacturing variability or changes that occurred through environmental exposes. (such as fouling or wettability changes). The main conclusions of our work appear to be supported by the Samsung data and withstand the discrepancies in the data.
Figure D.1 Comparison between present data and data provided by Samsung.

- $f_{ds}$, $j_{ds} = f$ and $j$ factors under dry condition provided by Samsung
- $f_{dry}$, $j_{dry} = f$ and $j$ factors under dry condition (present data)

a) (Plain, $f_s=1.5$mm, 2rows, Uncoated)  b) (Plain, $f_s=1.5$mm, 2rows, Coated)

c) (Plain, $f_s=1.5$mm, 3rows, Uncoated)  d) (Plain, $f_s=1.5$mm, 3rows, Coated)
Figure D.2 Comparison between present data and data provided by Samsung.

- $f_{ds}, j_{ds} = f$ and $j$ factors under dry condition provided by Samsung
- $f_{dry}, j_{dry} = f$ and $j$ factors under dry condition (present data)

a) (Slit, $f = 1.3$mm, 2rows, Uncoated)  b) (Slit, $f = 1.3$mm, 2rows, Coated)
c) (Slit, $f = 1.3$mm, 3rows, Uncoated)  d) (Slit, $f = 1.3$mm, 3rows, Coated)
Figure D.3 Comparison between present data and data provided by Samsung.

- $f_{ds}, j_{ds}$ = $f$ and $j$ factors under dry condition provided by Samsung
- $f_{dry}, j_{dry}$ = $f$ and $j$ factors under dry condition (present data)

a) (Slit, $f_s=1.5$mm, 2rows, Uncoated)  
b) (Slit, $f_s=1.5$mm, 2rows, Coated)  
c) (Slit, $f_s=1.5$mm, 3rows, Uncoated)
Figure D.4 Comparison between present data and data provided by Samsung.

- $f_{ws}, j_{ws} = f$ and $j$ factors under wet condition provided by Samsung
- $f_{wet}, j_{wet} = f$ and $j$ factors under wet condition (present data)

a) (Plain, $f_s=1.3\text{mm}, 2\text{rows, Uncoated}$) b) (Plain, $f_s=1.5\text{mm}, 2\text{rows, Uncoated}$) c) (Plain, $f_s=1.5\text{mm}, 3\text{rows, Uncoated}$)
Figure D.5 Comparison between present data and data provided by Samsung.

\( f_{ws}, f_{ws} = f \) and \( j \) factors under wet condition provided by Samsung
\( f_{wet}, f_{wet} = f \) and \( j \) factors under wet condition (present data)

a) (Slit, \( f_s = 1.3\text{mm} \), 2rows, Uncoated)  b) (Slit, \( f_s = 1.3\text{mm} \), 3rows, Uncoated)
c) (Slit, \( f_s = 1.5\text{mm} \), 3rows, Uncoated)  d) (Slit, \( f_s = 1.5\text{mm} \), 3rows, Uncoated)
APPENDIX E - CORRELATION RESULT

Figure E.1 Comparison between experimental data and correlation (Slit, Uncoated, Wet)

Figure E.2 Comparison between experimental data and correlation (Slit, Coated, Dry)
Figure E.3 Comparison between experimental data and correlation (Slit, Coated, Wet)