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EFFECT OF HUMIDITY AND INCLINATION ANGLE ON MICROCHANNEL HEAT EXCHANGER PERFORMANCE

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ABSTRACT

The effect of inlet humidity conditions and inclination angle on the air side thermal hydraulic performance of a brazed aluminum heat exchanger has been investigated experimentally. The crossflow heat exchanger had flat tubes and folded fins with fin and louver pitch of 2.1 mm and 1.4 mm, respectively. The glycol temperature was held nearly isothermal at 1°C, while humidity of the 12°C inlet air varied from 0—80%. The Reynolds numbers based on louver pitch ranged from 80 to 400. This range of operating conditions permitted air-side heat transfer coefficients to be determined within ±15-20% for dry and wet conditions, respectively. Wet-surface heat transfer degradation was substantial and varied with Reynolds number, but was only a weak function of inclination angles less than 45 degrees. On the other hand, the wetted-surface pressure drop penalty was relatively small, only 3-14% across the range of inclination angles and Reynolds numbers. The results provide insights into the ability to maintain louver-directed flow under inclined and wet-surface donations.

NOMENCLATURE

A: area  s: saturation state
C_p: specific heat t: temperature
C_r: capacity ratio δ_f: fin thickness
D_h: hydraulic diameter θ: Inclination angle
F_d: flow depth γ: aspect ratio
h: heat transfer coefficient ε: effectiveness
H: fin height η: Efficiency
i: enthalpy Subscripts
k: conductivity f: fin
L: heat exchanger height i: glycol-side
L_p: louver pitch o: air-side
m: mass flow rate w: wet-surface or water
NTU: Number of transfer unit 1: input
Q: heat transfer rate 2: output
RH: relative humidity

INTRODUCTION

The effect of inlet humidity conditions and inclination angle on the air side thermal hydraulic performance of a brazed aluminum heat exchanger has been investigated experimentally. There are some publications on the effect of inclination angle and inlet humidity conditions on the heat transfer and pressure drop of the heat
exchangers. However, most of the published data have considered bare-tube banks, high-fin tube banks and conventional finned tube heat exchangers (Groehn, 1983, Monherit et al., 1986, Moore et al., 1979, Aarde et al. 1993, Chang et al. 1994, Kedzierski, 1997, Mirth et al., 1993, Mirth et al., 1994, Wang et al. 1997, Wang et al., 2000). Mirth et al (1993, 1994) showed that inlet humidity conditions affected the heat exchanger performance. On the other hand, Wang et al. (1997, 2000) reported that they did not influence significantly the sensible heat transfer coefficients, while their effect on the pressure drops depended on the heat exchanger configurations, especially the longitudinal tube pitch. When the longitudinal tube pitch was 22 mm, the effect of inlet conditions was negligible, while for the longitudinal pitch of 19.05 mm the friction factors for RH =90% were 5-25% larger than those for RH=50%.

A microchannel tube heat exchanger is one of the potential alternatives for replacing the conventional finned tube heat exchangers and has been considered as both evaporator and gas cooler for prototype CO2 air-conditioning systems (Kim and Bullard, 2000). Many investigators have studied the air-side heat transfer and pressure drop characteristics of the louvered fin and flat tube heat exchangers (Sahnoun et al., 1992, Chang et al., 1996, Chang et al., 1997, Kim et al., 2000, Chiou et al., 1994, Kim and Bullard, 2002, McLaughlin et al., 2000). However, only small amount of published data on the effect of the inclination angle on the performance of the brazed aluminum heat exchangers is available in the open literature. Recently, Osada et al. (1999) studied the effect of inclination on the heat transfer and pressure drop characteristics of the louvered fin automotive evaporators with larger flow depth (F_d=58 and 70 mm) and conducted condensate visualization tests. They reported that both the leeward and windward inclinations improved heat exchanger performance. Kim et al. (2001) investigated the effect of inclination angle (0°, ±30°, ±45°, and ±60° clockwise) on the heat transfer and pressure drop of a brazed aluminum heat exchanger with F_d =20 mm under dry and wet conditions. They found that the heat transfer performance for both dry and wet conditions was not influenced significantly by the inclination angle (-60°<θ <60°), while the pressure drops increased consistently with the inclination angle.

In our residential CO2 prototype system, the indoor heat exchanger was inclined to 67° off the vertical due to the space limitations in the wind tunnel (Beaver et al. 1999). The effect of inclination in those experiments was not known because the heat exchanger was too large to test at different angles. The purpose of this study is to provide experimental data on the effect of an inclination angle and inlet humidity conditions on the air-side thermal hydraulic performance for a smaller brazed aluminum heat exchanger under dry and wet surface conditions. A series of tests are conducted for the air-side Reynolds number range of 80-400 with variation of the inclination angles (0°, 14°, 30°, 45° and 67° clockwise) from the vertical position. The pressure drop characteristics are also addressed. A more detailed analysis of the wet surface data can be found in Kim et al. (2002), which also explores the issue of leeward and windward inclinations and louver directions at the inlet and outlet of the heat exchangers.

**EXPERIMENTAL SET UP**

**Test Apparatus**

Figure 1 shows a schematic diagram of the apparatus used in the study. It consists of a ducted airflow system, heat transfer fluid (glycol) circulation and data acquisition system. It is situated in a constant temperature and humidity chamber that can maintain temperature within ±0.5°C and absolute humidity ±2%. The air inlet conditions of the heat exchanger are maintained by controlling the chamber temperature and humidity. The air-side pressure drop through the heat exchanger is measured using a differential pressure transducer and the airflow rate is determined form the nozzle pressure difference.

**Heat Exchanger**

The tested heat exchanger has 38 parallel tubes (circuits) and fins with 17 louvers, 27° louver angle, flow depth of 27.9 mm, fin pitch of 2.1 mm, fin thickness of 0.1 mm, tube pitch of 9.9 mm. The louver pitch, louver length and fin height are 1.4 mm, 6.6 mm and 8.3 mm, respectively and the core size is 394 mm x 381 mm.

**Test Conditions and Methods**

Figure 2 shows simple schematic of a heat exchanger installation. The heat exchanger is installed in the test section, surrounded by insulation to protect it from heat loss and air leakage. For leeward (θ = 0°, 14°, 45°, and
67° clockwise) inclinations, a series of tests for wet conditions are performed in the range of the Reynolds number based on louver pitch of 80-400. As shown in Fig. 3, the partition above the heat exchanger allows the air to turn and enter almost normal to the coil, and then it is then turned again through the duct. However for \( \theta = 67^\circ \), an upstream duct was added to prevent air leakage, and may have forced the air to enter the heat exchanger at a more oblique angle. The inlet air temperature and relative humidity ranges are 12°C and 0-80%, respectively, and glycol inlet temperatures are 0–2.5°C.

**DATA REDUCTION**

The total heat transfer rate used in the calculation is the arithmetic average of air- and glycol-side heat transfer rate (Q). The data reduction process is the same as Kim et al. (2001), so only a brief description is given here.

The \( \varepsilon \)-NTU equation for both fluids unmixed is

\[
\varepsilon = 1 - \exp \left[ \frac{NTU^{0.022}}{C_r} \{ \exp(-C_r NTU^{0.78}) - 1 \} \right]
\]  

(1)

Assuming zero glycol-side fouling resistance and wall resistance, the air-side heat transfer coefficient for dry-surface conditions can be obtained from the following equations

\[
\varepsilon = \frac{\dot{Q}}{\dot{m}_c C_p (t_{ia} - t_{ii})}, \quad NTU = \frac{UA}{\dot{m}_c C_p}, \quad C_r = \frac{\dot{m}_c C_p}{\dot{m} C_p}
\]

(2)

\[
1 = \frac{1}{UA} h_i A_i + \frac{1}{\eta_i h_i A_i}
\]

(3)

Assuming zero glycol-side fouling resistance and wall resistance, the air-side heat transfer coefficient for wet-surface conditions can be obtained from the following equations

\[
\varepsilon = \frac{\dot{Q}}{\dot{m}_c (t_{ia} - t_{i,\omega})}, \quad NTU = \frac{U_{\omega, A_{\omega}}}{\dot{m}_c}, \quad C_r = \frac{\dot{m}_c b'_i}{\dot{m} C_p}
\]

(4)

\[
1 = \frac{b'_i}{U_{\omega, A_{\omega}}} h_i A_i + \frac{b'_w}{\eta_{\omega, h_{\omega, A_{\omega}}}}
\]

(5)

Where \( b'_i \) and \( b'_w \) are the slope of the air saturated curve at the mean glycol temperature and the mean external surface temperature.

For the heat transfer coefficients on the glycol-side, two-dimensional duct flow was approximated since the duct aspect ratio (\( \gamma = 22.4 \)) is extremely large (Shah and London, 1978).

\[ Nu_f = 7.541 + 0.0235 \operatorname{Re}_f \operatorname{Pr} \frac{D_k}{L} \]

(6)

The surface effectiveness and the fin efficiency for the dry surface (Kuehn and Threlkeld, 1998) are

\[ \eta_o = 1 - \frac{A_f}{A_o} (1 - \eta_f) \]

(7)

\[ \eta_f = \frac{\tanh(m^* l)}{m^*}, \quad m^* = \sqrt{\frac{2h_o}{k_f \delta_f \left( 1 + \frac{\delta_f}{F_d} \right)}}, \quad l = H/2 - \delta_f \]

(8)

The solutions for efficiency of dry fins also apply for efficiency of wet fins if we substitute \( h_{\omega, A_{\omega}} \) for the wet fin in place of \( h_o \) for the dry fin. The overall heat transfer coefficient for the wet surface \( h_{\omega, A_{\omega}} \) is

\[ h_{\omega, A_{\omega}} = \frac{1}{\frac{C_p}{b_w h_o} + \frac{\eta_w}{k_w}} \]

(9)
Where \( h_s \) is the sensible heat transfer coefficient for the wet surface, and \( y_w \) is the thickness of condensation water film, which is neglected here.

Accounting for all instrument errors, property uncertainties, uncertainties for the heat transfer coefficients were estimated to be ±15-20% for dry and wet conditions, respectively. And uncertainties for the air-side pressure drops were about ±10% (Moffat, R.J., 1988).

RESULTS AND DISCUSSION

Figures 3-6 present the results for the heat transfer and pressure drop. Figure 3 shows how the air-side heat transfer coefficients for dry surface vary with face velocity and inclination angle. As expected, heat transfer coefficients increase with face velocity. The heat transfer coefficients for the dry conditions were not affected significantly by the inclination angles at low Reynolds number. The effect of inclination increases as Reynolds number increases. The heat transfer coefficients deteriorated substantially at 67°, especially when the Reynolds number was higher.

Figure 4 shows how the sensible heat transfer coefficients for wet surface vary with face velocity, inclination angle, and inlet relative humidity. Wet-surface heat transfer degradation was substantial and varied with Reynolds number. For the same inlet humidity (80% RH), heat transfer coefficients has a minimum when \( \theta = 0^\circ \). This may be due to the gravitational force effect on the promotion for condensate drainage. Recently, Kim et al. (2001) also reported a modest inclination angle promotes drainage so heat transfer coefficients increase. The heat transfer coefficients decrease with the increase of air inlet humidity, since higher inlet humidity will cause more condensate accumulation on the coil surface and this condensate acts as another thermal resistance for low Reynolds number flows studied here (Kim and Bullard, 2000). However, the inlet humidity effect on the heat transfer coefficient is not significant for the small inclination angles (\( \theta \leq 45^\circ \)). As shown in Figure 4, its effect increases with inclination angle and for \( \theta = 67^\circ \), the heat transfer coefficients decrease significantly with the increase of air inlet humidity.

Figure 5 presents air-side pressure drops vs. face velocity with variation of inclination angle. As expected, pressure drops for both dry and wet conditions increase systematically with face velocity and inclination angle. The pressure drops for wet conditions are 3-14% larger than those for dry conditions at the same face velocity. As shown in Figure 5, for \( \theta = 67^\circ \) a significant pressure drop increase was occurred. This result is similar to that by Kim et al. (2001) who reported pressure drops increased significantly when \( \theta \geq 60^\circ \). Furthermore, in case of \( \theta = 67^\circ \), there is an upstream duct which will causes the additional upstream losses associated with oblique air entrance to the heat exchanger.

Figure 6 shows the effect of air inlet humidity on the pressure drops. The inlet humidity does not influence significantly the pressure drops, a result similar to that for the conventional finned round tube heat exchangers with fully wet surface (Wang et al.1997). On the other hand, the previous test data with the micro-channel heat exchanger with smaller fin and louver pitch ratio (\( F_p / L_p = 1.4/1.7 \)) and larger flow depth (\( F_d = 41.8 \) mm) showed that the air inlet humidity affected systematically the air-side pressure drops (Boewe et al.1999) This difference probably is due to the difference of heat exchanger geometry. The heat exchanger used in this study has larger fin and louver pitch ratio (\( F_p / L_p = 2.1/1.4 \)) and smaller flow depth (\( F_d = 27.9 \) mm), and so the effect of condensate amount on the surface may be smaller compared to the heat exchanger with smaller fin pitch and larger flow depth, suggesting the inlet humidity effect on the pressure drops depends on heat exchanger configuration, especially fin pitch.

Under wet-coil conditions, data were not repeatable when the coil was vertical. We believe that the louvers in microchannel heat exchangers can become bridged with condensate under some conditions (e.g. at low face velocities in vertical orientation), so it appears as a flat fin and has degraded performance. To see the water drainage effect on air-side pressure drop and heat transfer coefficient, we ran some tests at 0° angle attack, 80% humidity, and 900 cfm where face velocity is about 4 m/s. The results were not repeatable, but a systematic relationship was observed. Figure 7 shows that the air-side heat transfer coefficients decrease with the air side pressure drop, perhaps due to condensate bridging between fins, which decreases both by changing louver-directed to duct-directed flow. Upon close inspection of the experimental procedures, it was found that the high air pressure drop and air-side heat transfer coefficient occurred when the evaporator was initially dry and the 900
A cfm test was run before any lower air flow tests. It indicates that the condensate bridging between fins will not happen if there is no bridging initially and the air-flow rate is high enough. The low air pressure drop and air-side heat transfer coefficient happened when the 900 cfm experiment was ran after some lower air flow rate experiments under wet surface conditions. It suggests that there is bridging between fins when the air flow rate is low. And surprisingly, the bridging still persists as the flow rate is increased gradually to 900 cfm.

CONCLUSIONS

The effect of leeward inclination on the heat transfer coefficients and pressure drops of a multi-louvered fin heat exchanger for dry and wet surface conditions has been investigated experimentally with variation of air inlet humidity. Wet-surface heat transfer degradation was substantial and varied with Reynolds number. The heat transfer characteristics are influenced significantly by the inclination angle, especially for $\theta > 45^\circ$. The pressure drops for wet conditions are 3-14% larger than those for dry conditions, and increase consistently with inclination angle. The effect of air inlet humidity on the heat transfer and pressure drop is negligible in case of the larger fin pitch heat exchanger studied here.

We concluded, tentatively at least, that a modest angle of attack promotes drainage. Probably that is why the auto industry routinely tilts its flat plate-folded louvered fin evaporators about 10 degrees off the vertical. Apparently it is also beneficial for the thinner (16mm) heat exchangers used in our residential prototype. We also believe that the louveres in microchannel heat exchangers can become bridged with condensate under some conditions (e.g. at low face velocities in vertical orientation). Surprisingly, they remain bridged as airflow rate is increased. However, if the airflow is initially large and the coil then begins to condense moisture in response to rising humidity, the drainage appears to remain unimpaired.

REFERENCES


**Figures**

![Schematic diagram of test apparatus.](image)

Indices: a – air, e – evaporator, g – glycol, i – inlet, n – nozzle, o – outlet

Figure 1: Schematic diagram of test apparatus.
Figure 2: Schematic diagram of a heat exchanger installation [unit of length: mm]

Figure 3: Angle effect on $h_o$ under dry condition

Figure 4: Angle and inlet humidity effect on $h_o$ under wet condition
Figure 5: Angle effect on air-side pressure drop

Figure 6: Inlet humidity effect on air-side pressure drop

Figure 7: Water drainage effect