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Thermal-Hydraulic Evaluation of Compact Heat Exchangers for Energy Reuse in Organic Rankine Cycles

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Abstract. *This project focuses in the thermal-hydraulic evaluation of two-types of compact heat exchangers used for energy reuse in Organic Rankine Cycles. For this, it is used a multi-pass cross-flow plain fin-and-tube heat exchanger with a 1.31 m^2 total surface area and 2 rows of circular tubes, and a single pass cross-flow heat exchanger with multi-louvered fin array, a 0.96 m^2 total surface area and 16 rectangular micro-channels. In the experiments, the heat exchangers operate with a single-phase water internal flow, and external air flow with inlet velocities between 1 and 2 m/s and inlet temperatures between 15 and 60°C. For the plain fin-and-tube exchanger, the external convection coefficients obtained experimentally are between 44.5 and 61.6 W/m^2K, and for the cross-flow exchanger with multi-louvered fin at the same operating conditions, between 143.1 and 193.3 W/m^2K. The obtained results are compared with thermal-hydraulic correlations.The totality of experimental Colburn j-factor data were correlated within $\pm$ 15% for both plain fin-and-tube and multi-louvered with micro-channels heat exchangers, with mean deviation of 6.7% and 10.1%, respectively.*

Keywords: *exchanger, experimental, Colburn, friction.*

1. INTRODUCTION

Compact heat exchangers have a area to volume ratio higher than 700 m^2/m^3 for gas-liquid flows (Kays and London, 1998). When compared with traditional heat exchangers, the employment of compact heat exchangers has the advantages of weight and volume downsizing, besides the costs reduction related to the manufacture of these equipment (Wang *et.al*, 1999). Such heat exchangers are used in HVAC systems, industrial cooling systems, Organic Rankine Cycles (ORC) and automotive radiators. The main objectives of this project are:

- Thermal-hydraulic experimental evaluation for two configurations of compact heat exchangers used in ORC- a multi-pass cross-flow plain fin-and-tube, and a single pass crossflow exchanger with multi-louvered fin array and micro-channels- by the determining the Colburn's j-factor and the fanning f-factor for the heat exchangers;
- Heat transfer capacity and frictional performance comparison between these two types of heat exchangers;
- Validation of thermal-hydraulic correlations by the comparison between the theoretical and experimental results;
- Determination of more adequate methods for the modelling procedure of these equipment.

2. BIBLIOGRAPHIC REVIEW

In the research area of Heat Transfer applied to heat exchangers, it is very common to work with correlations that can preview the heat transfer and friction characteristics of heat exchangers at varied operation conditions. These correlations are very useful for selecting and sizing heat exchangers for different applications because they help to spare resources and time at experimental testing.

There are different fin patterns used in compact heat exchangers, from the simple plain-fin configuration to the enhanced wavy and louver fins that are used in order to improve the heat transfer capacity of the heat exchanger. Although the simple plain-fin-and-tube configuration is still widely used for air-cooled heat exchangers because of their reliability and lower friction on the air-side in comparison to the wavy and louver fins ones, Wang *et.al* (1999) indicate that before 1999, there were just a few correlations for smaller diameter tubes below 8 mm, and that it was necessary to determine air-side correlations for a wider range of heat exchangers geometries using a larger and reliable database. It is presented in the paper of Wang *et.al* (1999), the new correlation developed by the authors using a total of 74 samples of plain-fin-and-tube heat exchangers, 676 and 530 data points for the Colburn's j-factor and fanning f-factor, respectively. Other correlations for plain-fin-and-tube were presented by different authors. For example, Kim *et.al* (1999) developed a new correlation using data from 47 samples of heat exchangers of those used by other authors to develop their correlations, like Wang et al. (1999).

In many applications such as Organic Rankine Cycles, automotive radiators, HVAC and cooling systems, a more compact heat exchanger with higher heat transfer capacity is necessary because of space restrictions. For that case, Wang and Chang (1997) point out that heat exchangers with enhanced surfaces in the air-side are commonly used. These enhanced and interrupted surfaces improve the heat transfer capacity by increasing the surface area, and by breaking up the growth of thermal boundary layer from the leading edge, hence inducing turbulent mixing of air flow. As mentioned before, heat exchangers with wavy, louver and slit fin patterns are examples of these surfaces applications.

Wang and Chang (1997) presented in their paper that although some correlations for louver fin geometry was available, the performance was still very poor, and a new generalized correlation was necessary using a larger and reliable data bank. To develop this new correlation, the authors used 91 samples of louver fin heat exchangers with different types of channels- triangular, plate-and-tube and rectangular. Following this work, other correlations for louver fin heat exchangers were presented by different authors, like the correlation developed by Ryu and Lee (2015). The authors studied the effects of the geometric parameters of louvered fins to determine general correlations.

In this work, the correlations developed by Wang *et.al* (1999) and by Kim *et.al*, and by Wang and Chang (1997) and by Ryu and Lee (2015) are used to predict the heat transfer and friction characteristics of a plain-fin-and-tube heat exchanger and a heat exchanger with multi-louvered fin array, respectively. Furthermore, experimental tests results are gathered in order to validate these correlations.

3. EXPERIMENTAL APPARATUS AND PROCEDURE

There are two distinct procedures for designing a heat exchanger: Sizing the equipment to meet a specified capacity or rating an existing heat exchanger for an application (Kanefsky *et.al*, 1998). As mentioned before, the objective of this work is the thermal-hydraulic evaluation of two heat exchangers in order to obtain a reliable performance database that can be useful for rating these equipment for a specified application. For this purpose, experimental tests with these heat exchangers are conducted at the Heat Transfer Research Group's laboratory (HTRG) at EESC-USP.

3.1 Materials

For the experimental tests of heat exchangers, it is used the wind tunnel of the HTRG. This wind tunnel is made of reinforced stainless steel walls and it is with agreement within the standard testing methods of ASHRAE 33 and 51 (2016), and ASHRAE 41.2 (2018). A schematic diagram on Fig.1 represents the wind tunnel assembly. It operates in a closed-loop mode at which a centrifugal fan promotes the air circulation. The airflow velocity is controlled by a frequency inverter that is coupled together to the fan's electrical motor. The airflow temperature is controlled by a cooling coil connected to a chiller and by also a electrical resistances bench. The air humidity can be changed by a joint operation of the cooling coil and a humidity injector. Honeycomb flow passages are installed in the test section in order to reduce turbulence and to achieve a uniform air velocity profile.

In order to minimize the heat transfer between the environment and the wind tunnel walls, all its external surface was thermally insulated with low thermal conductivity blankets. A secondary fluid circuit, not shown in Fig. 1 provides the fluid that flows inside the compact heat exchanger at controlled operation conditions.

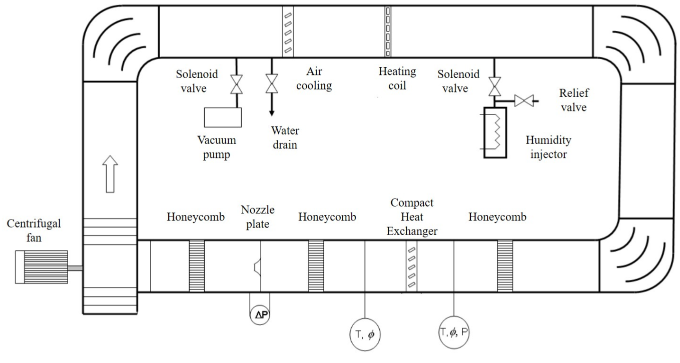


Figure 1. Schematic diagram of the wind tunnel used for experimental tests of heat exchangers.

The data acquisition is performed by data acquisition boards from National Instruments and the Labview (2007) software. The pressure, temperature and humidity sensors are plugged in the acquisition system terminals. Output channels control the frequency inverter and the solid state relays connected in series with the electrical resistance. In order to measure the air flow it is used a nozzle plate built in agreement to the standard method of ASME 1989. Besides that, the liquid flow that flows inside the heat exchanger is measured by turbine type flow meters.

The air inlet and outlet temperatures are measured with a 0.25 mm type-T thermocouples net. Besides that, additional thermocouples are installed at the the inlet and outlet surfaces of the tested heat exchanger. The relative air humidity is determined by the wet bulb temperature. Differential pressure sensors allow to obtain the pressure drop on the air side.

Two configurations of heat exchangers are tested in the wind tunnel: a multi-pass cross-flow plain fin-and-tube configuration (Heat Exchanger A); and a single-pass cross-flow with multi-louvered fin array and micro-channels configuration (Heat Exchanger B). These equipment are schematically represented by Figs. 2a and 2b available on Qasem and Zubair's (2018) paper.

(a) Multi-pass cross-flow plain fin-and-tube.

(b) Single-pass cross-flow with multi-louvered fin array and microchannels.

Figure 2. Geometric parameters' nomenclature of Heat Exchangers A and B.

Using the same geometric parameters' nomenclature used by Qasem and Zubair (2018), the Tab. 1 is set:

Table 1. Heat exchangers' geometric parameters.

Measurement	Symbol	Heat Exchanger- A	Heat Exchanger- B	Unity
Collar diameter	D_c	10.25	-	$[mm]$
Tube internal diameter	D_i	8.00	0.66	$[mm]$
Tube external diameter	D_o	9.95	2.00	$[mm]$
Fin depth	F_d	-	28.00	$[mm]$
Fin length	F_l	-	7	$[mm]$
Fin pitch	F_p	2.90	1.40	$[mm]$
Fin thickness	F_t	0.15	0.10	$[mm]$
Louver length	L_l	-	6.00	$[mm]$
Louver pitch	L_p	-	1.00	$[mm]$
Louver angle	θ	-	20.00	[°]
Longitudinal spacing	P_l	18.00	-	$[mm]$
Transverse spacing	P_t	25.50	-	$[mm]$
Tube depth	T_d	-	28.00	$[mm]$
Tube pitch	T_p	-	9.00	$[mm]$
Heat exchanger depth	L_d	38.50	32.00	$[mm]$
Heat exchanger lateral length	L_{lat}	215.00	216.00	$[mm]$
Heat exchanger vertical length	L_{vert}	215.00	143.00	$[mm]$
Number of fins between tubes	N_{fin}	74	129	-
Number of tube rows	N_{rows}	2	16	-
Number of each row's passages	N_t	6	1	-
Minimum free flow area	A_c	0.03	0.02	$[m^2]$
Frontal area	A_{front}	0.05	0.03	$[m^2]$
Fin surface area	A_f	1.24	0.76	$[m^2]$
Tube surface area	A_t	0.08	0.20	$[m^2]$
Total area	A_{tot}	1.33	0.96	$[m^2]$
Tunnel cross-sectional area	A_{tunnel}	0.08	0.08	$[m^2]$
Heat exchanger hydraulic diameter	D_h	3.55	2.64	$[mm]$

3.2 Data-reduction

In this section it is explained how the experimental Colburn j-factor and friction f-factor are determined.

3.2.1 Heat transfer capacity determination

First of all, the heat transfer rate is determined by the energy balance from Eq.(1):

$$\overline{q} = q_{air} = q_{water} = |\dot{m_{air}} c_{p,air}(T_{air,in} - T_{air,out})| = |\dot{m_{water}} c_{p,water}(T_{water,in} - T_{water,out})| \tag{1}$$

q_{air} and q_{water} are the heat transfer rate measured on the air and water side, respectively, they are not exactly the same in practical applications. Also, the liquid mass flow rate is indicated by $\dot{m}$, the specific heat by c_p and the inlet and outlet temperatures by T_{in} and T_{out}, respectively.

In order to obtain the overall heat transfer coefficient (U), and then the Colburn's j-factor, the $\epsilon - NUT$ is used (Kays and London, 1998). First, it is calculated the heat exchanger effectiveness (ϵ) by Eq.(2).

$$\varepsilon = \frac{\overline{q}}{C_{min}(T_{water,in} - T_{air,in})} \tag{2}$$

Where C_{min} is the smaller heat capacity of both fluids, i.e of water and C_{max} is the heat transfer capacity of air. Eq.(3) indicates that the number of transfer units (NTU) is a function of the heat exchanger effectiveness. In this work, NTU is calculated through an $\epsilon - NTU$ relation for cross-flow heat exchangers with both fluids unmixed, as showed by Eq.(3) (Kays and London, 1998).

$$NTU = -ln(1 - \epsilon) \tag{3}$$

After that, the overall heat transfer coefficient is determined by Eq.(4).

$$UA_{tot} = \frac{1}{R_{tot}} = NTUC_{min} \tag{4}$$

Where R_{tot} is the total thermal resistance.

The external convection coefficient (h_0) is obtained using Eq.(5). The left-side of the equation is the total thermal resistance. The first part on the right-side of the equation is the external convection resistance, the second one is tube the wall conduction thermal resistance and the last part is the internal convection resistance. Encrustation thermal resistances are not considered.

$$R_{tot} = \frac{1}{UA_{tot}} = \frac{1}{n_{0,array} h_0 A_{tot}} + \frac{ln(\frac{D_c}{D_i})}{2\pi k_{aluminum} L} + \frac{1}{h_i A_{int}} \tag{5}$$

The heat exchanger's material thermal conductivity is assumed as $k_{aluminum} = 200\ W/m.K$, the tube wall's internal area as A_{int} and the tubes' length as L. The fin array efficiency is denoted by $n_{0,array}$ and is given by Eq.(6), where $n_{0,fin}$ and m are given on Eq.(7) and Eq.(8), respectively (Kays and London, 1998).

$$n_{0,array} = (1 - \frac{A_f}{A_{tot}})(1 - n_{0,fin}) \tag{6}$$

$$n_{0,fin} = \frac{tanh(mL_c)}{mL_c} \tag{7}$$

$$m = \frac{h_0 P}{K_{aluminum} A_p} \tag{8}$$

Where P and A_p are the fin's perimeter and cross sectional area, respectively.

The internal convection coefficient for a single-phase liquid flow (h_i) is defined by Gnielinski's correlation (1976) on Eq.(10) or by the Graetz (Gz) number for thermal development region on Eq.(12)

For $3000 \leq Re_{Dh,water} \leq 5.10^6$ and $0.5 \leq Pr_{water} \leq 2000$:

$$f_{D,water} = (0.79.ln(Re_{Dh,water} - 1.64))^{-2} \tag{9}$$

$$Nu_{Dh,water} = \frac{(f_{D,water}/8)(Re_{Dh,water} - 1000)Pr_{water}}{1 + 12.7(f_{D,water}/8)^{1/2}(Pr_{water}^{2/3} - 1)} \tag{10}$$

For $Re_{Dh,water} \leq 3000$ and $0.5 \leq Pr_{water} \leq 2000$, considering the tubes' walls of the heat exchangers as isothermal:

$$x_{et} = 0.034 Re_{Dh,water} Pr_{water} \tag{11}$$

$$Gz_{Dh,water} = \frac{Re_{Dh,water} Pr_{water} D_h}{x_{et}} \tag{12}$$

$$\overline{Nu_{Dh,water}} = \frac{3.657}{tanh(2.264 Gz_{Dh,water}^{-1/3} + 1.7 Gz_{Dh,water}^{-2/3})} + 0.049 Gz_{Dh,water} tanh(Gz_{Dh,water}^{-1}) \tag{13}$$

Where The Darcy-Weisbach's friction factor ($f_{D,water}$) is given by Eq.(9) (Darcy and Weisbach, 1854). x_{et} is the thermal development length and is given by Eq.(11). $Nu_{Dh,water}$ and $Re_{Dh,water}$ are the Nusselt and Reynolds numbers based on the tube's hydraulic diameter at the operating conditions, as well as the Prandtl number denoted by Pr_{water}.

A system of equations is compound by isolating h_0 on Eq.(5) and Eq.(8). Thus, with all other terms defined, it is possible to calculate the external convection coefficient h_0 and the fin array efficiency $n_{0,array}$.

Reynolds and Colburn (1933) developed a widely used analogy for correlating heat, momentum and mass transfer. This analogy is used for determining the Colburn j-factor for the heat exchanger from its external convection coefficient, its hydraulic diameter and the air flow characteristics. It is presented on Eq.(14).

For $Pr_{air} \leq 0.6$:

$$j = \frac{Nu_{Dh,air}}{Re_{Dh,air} Pr_{air}^{1/3}} = \frac{f}{2} \tag{14}$$

3.2.2 Frictional characteristics determination

The Reynolds-Colburn analogy is also used to determine the friction fanning-factor related to an external air flow over a heat exchanger (f). The analogy correlates the friction factor to the convection coefficient as shows the Eq.(14). The load loss (ΔP_{air}) is obtained by Eq.(15) as defined by Kays and London (1998) and used by Wang and Chang (1997).

$$\Delta P_{air} = (\frac{f A_{tot} \rho_1}{A_c \rho_m} + (K_c + 1 - \sigma^2) + 2(\frac{\rho_1}{\rho_2} - 1) - (1 - \sigma^2 - K_e)\frac{\rho_1}{\rho_2})\frac{G_c^2}{2\rho_1} \tag{15}$$

Where ρ_1, ρ_2 e ρ_m are the air density at the inlet, at the outlet and the mean air density, respectively. Moreover, it is assumed as K_c and K_e the pressure loss coefficients for abrupt expansion and contraction, respectively (Kays and London, 1998). Finally, σ is defined as the fin array contraction ratio and G_c is given by $G_c = \rho_m * V_c$.

3.2.3 Comparison between different air-side geometries

The method used in this paper to compare the air-side thermal-hydraulic performance for the two different fin arrangements was proposed by Kays and London (1998). This comparison method consists in plotting external convection coefficient (h_0) vs. friction power ($E_{friction}$) in order to verify the relationship between heat transfer capacity and friction. This friction power is given by Eq.(16).

$$E_{friction} = \frac{\mu_{air}^3}{2\rho_{air}^2}(\frac{1}{D_h})^3 f Re_{Dh,air}^3 \tag{16}$$

3.2.4 Thermal-hydraulic correlations

The thermal-hydraulic correlations used to evaluate Heat Exchangers A and B are presented in Tab. 2. They are expressed in terms of the Reynolds number on the air-side and the heat exchanger geometric parameters. Some correlations' terms are not defined in this paper, if necessary, additional information should be sought in authors' respective papers. These correlations are used to preview the heat transfer capacity and frictional characteristics, by determining the Colburn j-factor and fanning friction f factor, for different operating conditions, configurations and geometries of heat exchangers.

Table 2. Authors' correlations summary.

Author	Colburn j-factor correlation	Fanning f-factor
Wang et.al (1999)- Plain fin	$j = 0.086 Re_{Dc,air}^X N_{rows}^Y (\frac{Fp}{Dc})^Z (\frac{Fp}{Dh})^M (\frac{Fp}{Pt})^{-0.93}$	$f = 0.0267 Re_{Dc,air}^{X1} (\frac{P_t}{P_l})^{Y1} (\frac{F_p}{D_c})^{Z1}$
Kim et.al (1999)- Plain fin	$j = j_3 1.043 (Re_{Dc,air}^{-0.14} (\frac{P_t}{P_l})^{-0.564} (\frac{F_p}{D_c})^{-0.123} (\frac{P_t}{D_c})^{1.17})^{(3-N_{rows})}$	$f = f_1(\frac{A_f}{A_{tot}}) + f_2(1 - \frac{A_f}{A_{tot}})(1 - \frac{F_t}{F_p})$
Wang and Chang (1997)- Louver fin	$j = Re_{Lp}^{-0.49} (\frac{\theta}{90})^{0.27} (\frac{F_p}{L_p})^{-0.14} (\frac{F_l}{L_p})^{-0.29} (\frac{T_d}{L_p})^{-0.23} (\frac{L_l}{L_p})^{0.68} (\frac{T_p}{L_p})^{-0.28} (\frac{F_t}{L_p})^{-0.05}$	$f = f_1 f_2 f_3$
Ryu and Lee (2015)- Louver fin	$j = Re_{Lp}^{(-0.484 - 1.887/lnRe_{Lp})} (\frac{F_d}{L_p})^{0.157} (2.24 - 0.588 ln(\frac{F_p}{L_p sin L_p}))$	$f = Re_{Lp}^{-0.433} (\frac{F_d}{L_p})^{0.185} (1.10 + 4.31(\frac{\theta}{90})^2 + 0.836 \frac{ln(F_p/L_p)}{(F_p/L_p)^2})$

4. RESULTS

The experimental results and the comparison between these and the theoretical results previewed by correlations are presented in this section. The results of comparisons are given for 100% of the available data in terms of the parcel of experimental results predicted within error bands ($\pm\phi$) and the absolute mean relative deviation (ε) obtained as follows, where N is the number of data points.

$$\phi = \pm max(\frac{j(f)_{cor} - j(f)_{exp}}{j(f)_{exp}}) \tag{17}$$

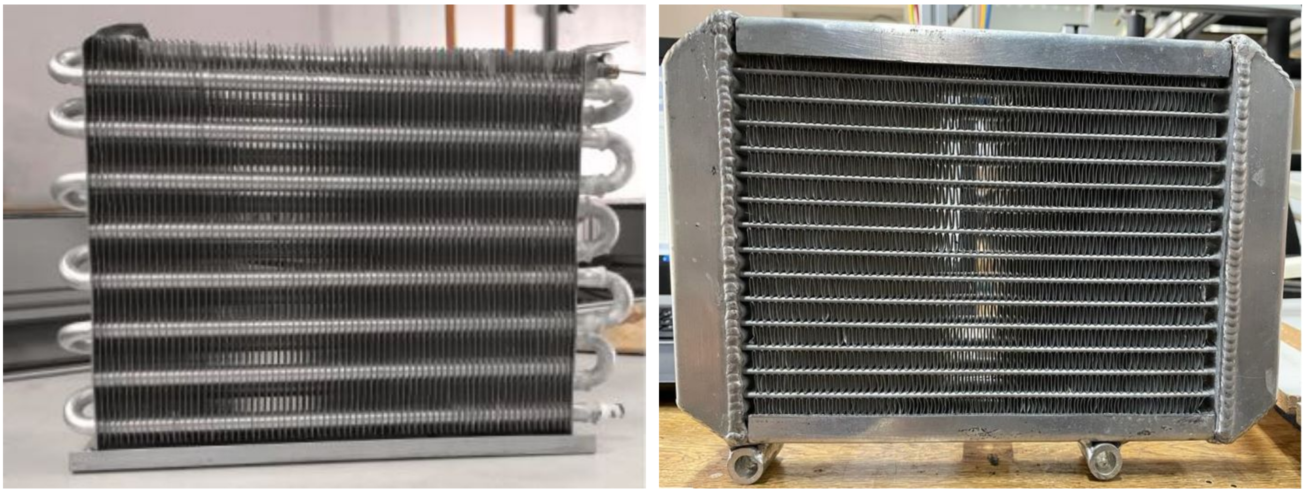
$$\varepsilon = \sum_{1}^{N} \left| \frac{1}{N}(\frac{j(f)_{cor} - j(f)_{exp}}{j(f)_{exp}}) \right| \tag{18}$$

Furthermore, it is shown in this section a heat transfer capacity and frictional performance comparison between the two tested types of heat exchangers. The software Matlab (2015) was used to perform the calculations and to generate the graphic plots. Results are presented for three different experiments using two compact heat exchangers shown in Fig.(3):

1. Heat Exchanger- A with hot water and cold air;

2. Heat Exchanger- A with cold water and hot air;

3. Heat Exchanger- B with hot water and cold air.



(a) Heat Exchanger- A with plain-fin-and-tube geometry. (b) Heat Exchanger- B with louver fin geometry.

Figure 3. Compact heat exchangers tested in the wind tunnel.

4.1 Experiment 1: Heat Exchanger- A with hot water and cold air

Inlet water temperature ($T_{water,inlet}$) and inlet air temperature ($T_{air,inlet}$) were set at approximately 50 °C and 17 °C. Water mass flow ($\dot{m_{water}}$) was maintained at 0.1 kg/s and inlet air velocity (V_{air}) from 1 to 2 m/s.

Using ϵ-NTU method and Reynolds-Colburn analogy, experimental Colburn j-factors were obtained and compared with Wang *et.al* (1999) and Kim *et.al* (1999) correlations predicted results. Figure (4) indicates how Re_{Dc} is related to Colburn j-factor and friction f-factor. This figure also indicates how obtained experimental results fit to correlational ones. Table 3, Fig.(5a) and Fig.(5b) present the statistical parameters obtained from the comparison between the predicted and the experimental results.

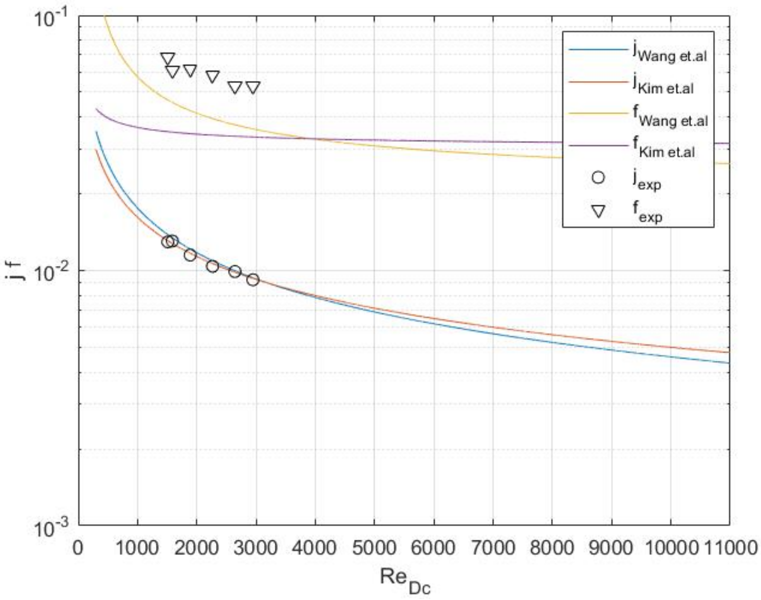
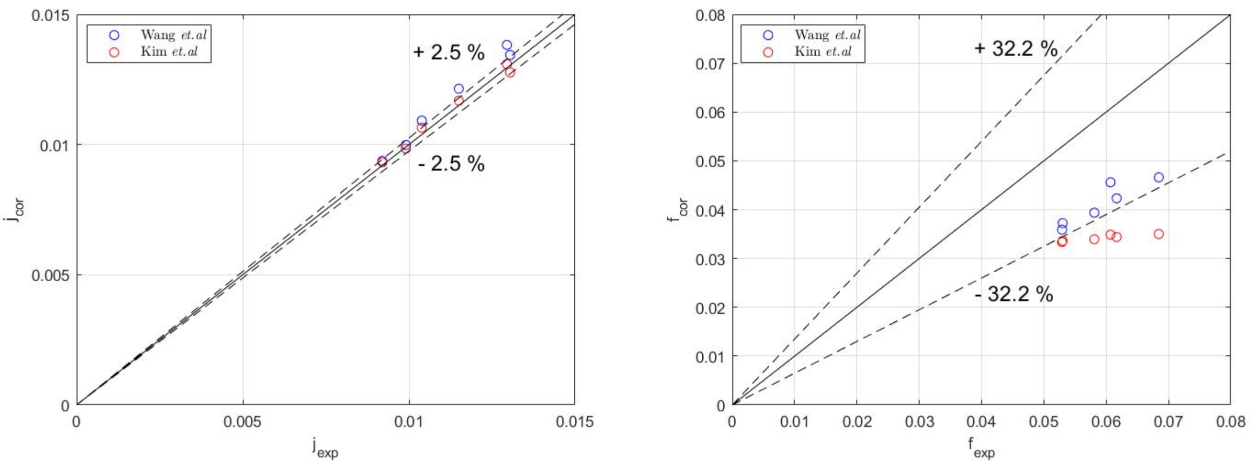


Figure 4. Experimental and correlational j and f factors.



(a) Comparison of heat transfer correlations with the experimental data. (b) Comparison of friction correlations with the experimental data.

Figure 5. Statistical parameters resulting from the comparison.

Table 3. Comparison of statistical results of Experiment 1.

Correlation	ϕ_j	ε_j	ϕ_f	ε_f
Wang et.al (1999)	6.8%	3.8%	32.2%	30.4%
Kim et.al (1999)	2.5%	1.5%	48.9%	41.8%

It is possible to verify that Kim's *et.al* (1999) correlation better fits experimental Colburn j factor when compared to Wang's *et.al* (1999), resulting in a ϕ_j of only 2.5% and a ε_j of 3.8%. However, Kim's frictional correlation underestimate f factor, hence Wang's frictional correlation was the one chosen that better fits the experimental results when a correction factor α is applied to predicted f-factor:

$$f_{exp} \cong \alpha f_{Wang} \quad (19)$$

$$f_{exp} \cong 1.44 f_{Wang} \quad (20)$$

When $\alpha = 1.44$, Wang's corrected frictional correlation represents the experimental data within $\pm$ 8.2% and $\varepsilon_f{}'$ of 2.9%.

Therefore, as Kim's correlation j-factor and Wang's corrected correlation f-factor correlations fit successfully the experimental data, it is possible to predict Heat Exchanger- A's external convection coefficient (h_0) and the air pressure drop (ΔP_{air}) for varied Re_{Dc}, hence facilitating the task of rating this heat exchanger for a specified application. Figure (6) presents the predicted curves.

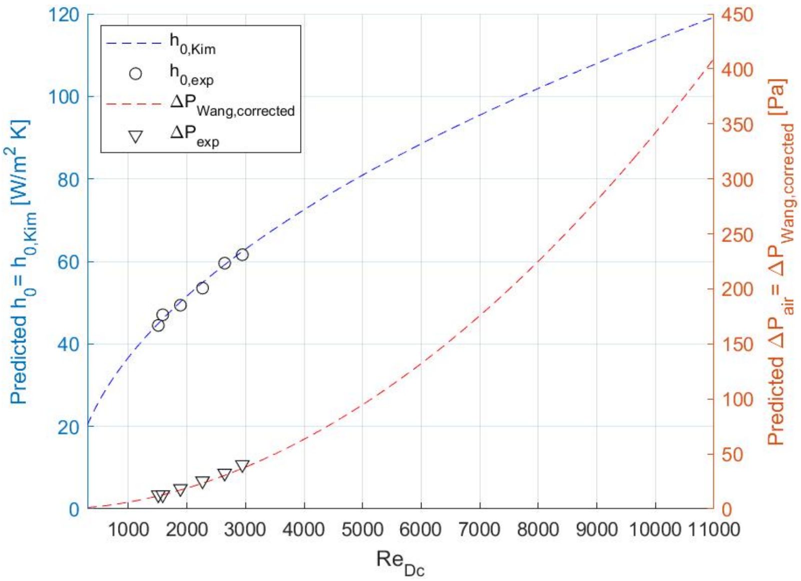


Figure 6. Experiment 1: Predicted h_0 and ΔP_{air}.

4.2 Experiment 2: Heat Exchanger- A with cold water and hot air

Inlet water temperature ($T_{water,inlet}$) and inlet air temperature ($T_{air,inlet}$) were set at approximately 35 °C and 55 °C. Water mass flow ($\dot{m_{water}}$) was maintained at 0.1 kg/s and inlet air velocity (V_{air}) from 1 to 2 m/s.

Experimental Colburn j-factors were obtained and compared with Wang *et.al* (1999) and Kim *et.al* (1999) correlations predicted results for this Experiment 2 setup. Figure (7) indicates how Re_{Dc} is related to Colburn j-factor and friction f-factor. It also indicates how obtained experimental results fit to correlational ones. Table 4, Fig.(8a) and Fig.(8b) present the statistical parameters obtained from the comparison between the predicted and the experimental results.

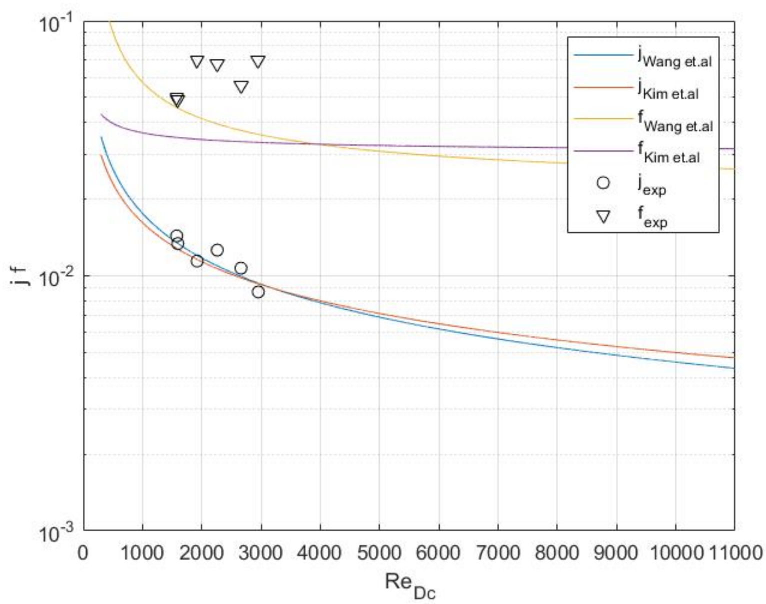
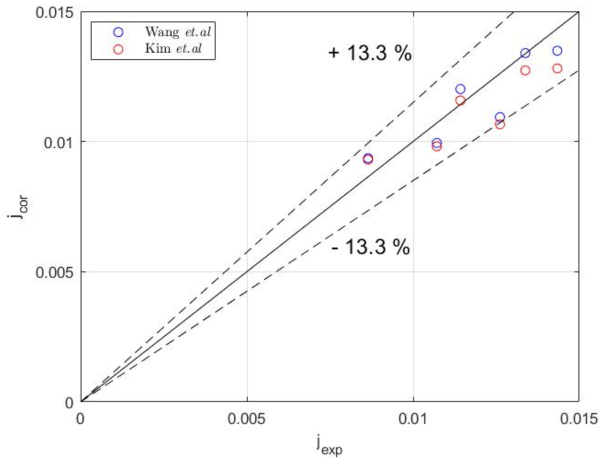
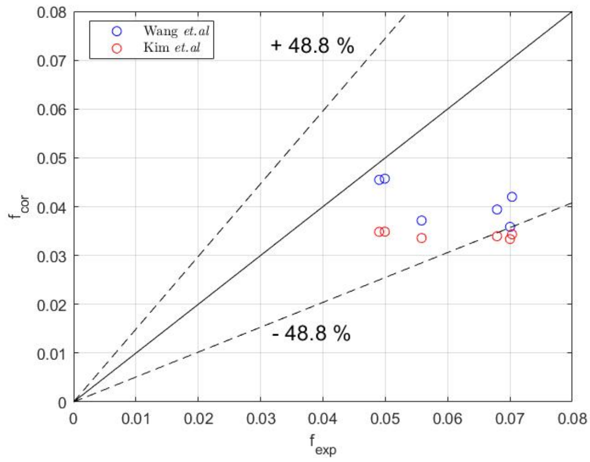


Figure 7. Experimental and correlational j and f factors.



(a) Comparison of heat transfer correlations with the experimental data.



(b) Comparison of friction correlations with the experimental data.

Figure 8. Statistical parameters resulting from the comparison.

Table 4. Comparison of statistical results for Experiment 2.

Correlation	ϕ_j	ε_j	ϕ_f	ε_f
Wang et.al (1999)	13.3%	6.7%	48.8%	30.0%
Kim et.al (1999)	15.5%	8.0%	52.3%	42.1%

In this Experiment 2, Wang's *et.al* (1999) correlation better fits experimental Colburn j-factor when compared to Kim's *et.al* (1999), resulting in a ϕ_j of 13.3% and a ε_j of 6.7%. However, the experimental and predicted data fitting was not as successful as in Experiment 1: ϕ_j more than quintupled and ε_j increased 76%.

Besides that, neither tested correlation was able to fit the experimental f-factor: Wang's correlation presented ϕ_f higher than 48% and ε_f of 30%. Not similar to Experiment 1, a correction factor α wasn't able to fit the predicted results to the experimental ones. A possible explanation for this deviation increase is that in Experiment 2 the air flow heating may create hot and cold zones near the nozzle plate, hence inducing natural convection on air flow and an inaccurate measurement of ΔP_{air}. Further investigation should be conducted in order to analyse the possible causes of the deviation increase.

Therefore, as Wang's correlation best-fitted the experimental results for this Experiment 2 setup, this correlation was used to predict Heat Exchanger- A's external convection coefficient at varied operating conditions (h_0) with hot air flow. Air pressure drop was not predicted with any correlation, since neither was able to successfully fit to experimental data. The task of rating Heat Exchanger- A to a specified application with hot air flow would be more complicated since further investigation should be conducted in order to predict frictional performance of this heat exchanger accurately.

4.3 Experiment 3: Heat Exchanger- B with hot water and cold air

In Experiment 3, the inlet water temperature ($T_{water,in}$) and the inlet air temperature ($T_{air,in}$) were set at approximately 17 °C and 54 °C, respectively. Water mass flow ($\dot{m_{water}}$) was maintained at 0.1 kg/s and inlet air velocity (V_{air}) from 1 to 1.75 m/s.

Experimental Colburn j-factors were obtained and compared with Wang and Chang (1997) and Ryu and Lee *et.al* (2015) correlations predicted results for this Experiment 3 with Heat Exchanger- B, a multi-pass cross-flow with corrugated louver fin array. Figure (7) indicates how Re_{Dc} is related to Colburn j-factor and friction f-factor. It also indicates how obtained experimental results fit to correlational ones. Table 5, Fig.(10a) and Fig.(8b) present the statistical parameters obtained from the comparison between the predicted and the experimental results.

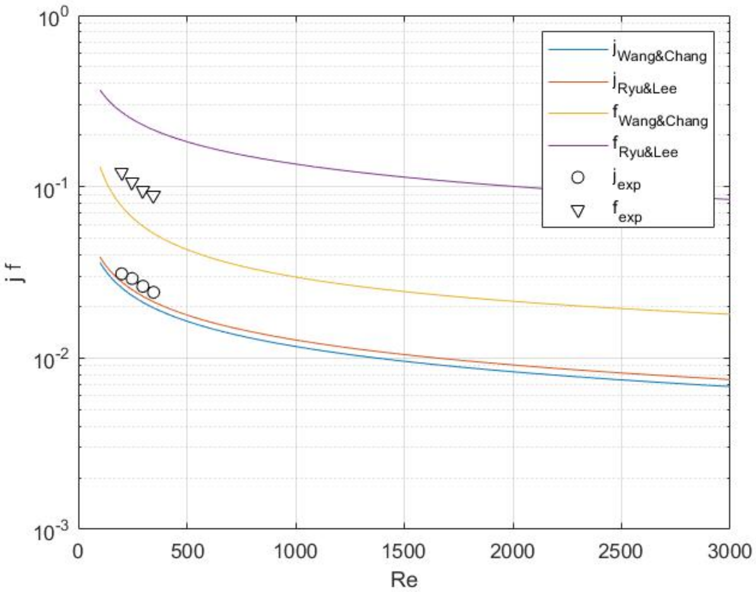
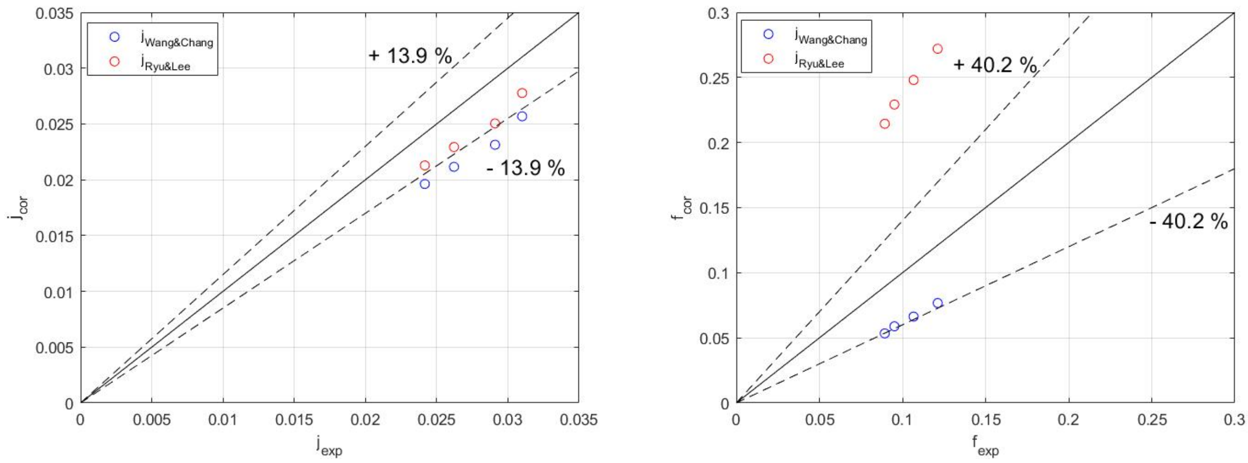


Figure 9. Experimental and correlational j and f factors.



(a) Comparison of heat transfer correlations with the experimental data. (b) Comparison of friction correlations with the experimental data.

Figure 10. Statistical parameters resulting from the comparison.

Table 5. Comparison of statistical results for Experiment 3.

Correlation	ϕ_j	ε_j	ϕ_f	ε_f
Wang and Chang (1997)	20.5%	17.5%	40.2%	38.2%
Ryu and Lee (2015)	13.9%	12.2%	140.9%	134.5%

It is possible to verify that Ryu and Lee's (2015) correlation better fits experimental Colburn j-factor when compared to Wang and Chang's (1997) correlation, resulting in a ϕ_j of 13.9% and a ε_j of 10.1%. A correction factor β is applied to reduce ϕ_j and ε_j in order to obtain a more accurate performance prediction of Heat Exchanger-B:

$$j_{exp} \cong \beta j_{Ryu\&Lee} \tag{21}$$

$$j_{exp} \cong 1.139 j_{Ryu\&Lee} \tag{22}$$

The same is done for Wang and Chang's frictional correlation:

$$f_{exp} \cong \alpha f_{Wang\&Chang} \tag{23}$$

$$f_{exp} \cong 1.62 f_{Wang\&Chang} \tag{24}$$

With corrections applied, Ryu and Lee's corrected heat transfer correlation and Wang and Chang's friction correlation represents the experimental data within $\phi_j{}' = \pm 2.0$ and $\varepsilon_j{}'$ of 1.17%, $\phi_f{}' = \pm 3.1\%$ and $\varepsilon_f{}'$ of 1.65%. Therefore, as the correlations fit successfully to the experimental data, it is possible to predict Heat Exchanger- B's external convection coefficient (h_0) and the air pressure drop (ΔP_{air}) for varied Re_{Lp}, hence facilitating the task of rating this heat exchanger for a specified application. Figure (11) presents the predicted curves based on the corrected correlations.

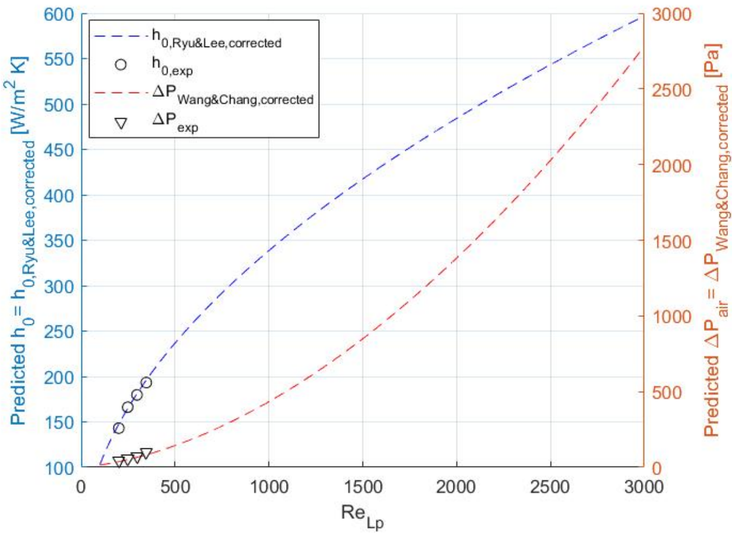


Figure 11. Experiment 3: Predicted h_0 and ΔP_{air}.

4.4 Heat transfer capacity and frictional performance comparison between Heat Exchangers A and B

Compact heat exchangers are used in different applications depending upon different criteria including heat transfer enhancements, friction power, and compactness. Improving heat transfer by enhancing the fin surface and thus interrupting the airflow result in a high friction power (Qasem and Zubair, 2018). Qasem and Zubair (2018) also reviewed the fact that plain-fin-and-tube heat exchangers are known to have low friction power, lower cost but low heat transfer capacity. In the other hand, louver-fin heat exchangers offer higher heat transfer capacity and also good compactness, however, they present high friction power.

In order to compare Heat Exchangers A and B at same operating conditions, data from Experiment 1 and 3 was used. First, a simple comparison between the Heat Exchangers' experimental effectiveness, external convection coefficient, heat transfer rate and air pressure drop was able to point out that for the same operating conditions, a heat exchanger with louver fin geometry has a more effective air-side thermal performance however higher friction than a plain-fin-and-tube configuration. Table 6 presents this comparison between Heat Exchanger A and B.

Table 6. Performance comparison between Heat Exchangers A and B.

V_{air} $[m/s]$	$\dot{m}_{water}$ $[kg/s]$	ϵ_A -	ϵ_B -	$h_{0,A}$ $[W/m^2K]$	$h_{0,B}$ $[W/m^2K]$	$q_{air,A}$ $[W]$	$q_{air,B}$ $[W]$	$\Delta P_{air,A}$ $[Pa]$	$\Delta P_{air,B}$ $[Pa]$
1.00	0.10	0.43	0.67	47.05	143.08	1294.60	2229.40	12.90	40.50
1.25	0.10	0.39	0.63	49.38	166.22	1362.60	2242.40	18.50	56.30
1.50	0.10	0.36	0.58	53.52	179.71	1504.80	2430.50	25.50	74.50
1.75	0.10	0.34	0.54	59.57	193.28	1608.50	1114.00	32.50	96.90
2.00	0.10	0.32	-	61.62	-	1661.10		40.30	-

Using the method presented by Kays and London (1998), in order to verify the relationship between heat transfer capacity and friction, a graphic of external convection coefficient (h_0) vs. friction power ($E_{friction}$) was plotted. In this graphic, the corrected correlations presented in Sections 4.1 and 4.3 are plotted together with experimental data in Fig.(12).

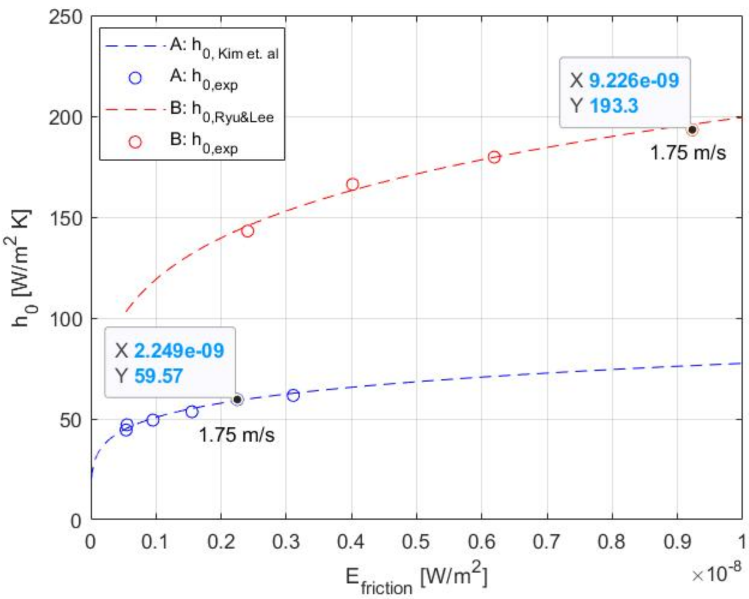


Figure 12. Performance comparison between Heat Exchangers A and B.

Analyzing the specific points at which $V_{air} = 1.75\ m/s$ in the wind tunnel: the external convection coefficient for Heat Exchanger- B is about 3.25 times higher than Heat Exchanger- A's h_0 at the same operating condition, even with a total surface area of 72% of A's total area. However, for this same air velocity, B's friction power is more than 4 times higher than A's. Thus, it is possible to verify that enhanced and interrupted surfaces improve the heat transfer capacity as it increases the surface area, and breaks up the growth of thermal boundary layer from the leading edge, hence inducing turbulent mixing of air flow and, consequently, higher friction power.

The correlations's curves on Fig.(12) shows that, for the same $E_{friction}$, a heat exchanger with louver fin geometry (B) is capable of achieving a higher h_0 than a plain-fin-and-tube one (A). It should be emphasized that selection of a surface configuration for a particular application is not as simple as just considering this performance indicator, for there are many additional considerations that should be contemplated (Kays and London, 1998).

5. CONCLUSION

The study presented in this paper tested the thermal-hydraulic evaluation method used for rating compact heat exchangers for different applications. For that, experimental tests were conducted at HTRG's laboratory in order to obtain performance indicators for two configurations of compact heat exchangers: a multi-pass cross-flow plain fin-and-tube, and a single pass cross-flow exchanger with multi-louvered fin array and micro-channels. The obtained j and f factors were compared with thermal-hydraulic correlations. 100% of experimental Colburn j-factor data were correlated within $\pm$ 15% for both plain fin-and-tube and multi-louvered with micro-channels heat exchangers, with mean deviation of 6.7% and 10.1%, respectively. However experimental friction f-factors were not successfully correlated by any of tested correlations and correction factors needed to be applied. Thus, it was possible to effectively predict heat transfer capacity and friction performance on the air-side of both tested heat exchangers. By the end of this work, a performance comparison between these two types of heat exchanger was conducted in order to investigate the effect of enhanced surfaces on the heat transfer capacity and friction of heat exchangers, showing the advantages of multi-louvered fin array and micro-channels over plain-fin-and-tube configurations for heat transfer capacity.

6. ACKNOWLEDGEMENTS

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