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EXPERIMENTAL ANALYSIS OF THE EFFECT OF COLD FLUID INLET TEMPERATURE ON THE THERMAL PERFORMANCE OF A HEAT EXCHANGER

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ABSTRACT

This paper focuses on the effect of inlet temperature of the cold fluid on the thermal performance of two mini channel flat-tube with multi-louvered fin compact heat exchangers experimentally. Two louvered fin heat exchangers in the same sizes with different number of fin rows are tested in a constant temperature test room by a wind-tunnel. The cold fluid flowing on the external side of the heat exchanger is air. The hot fluid flowing through the mini channels is water. The inlet temperature of the water passes through the mini channel flat-tube is predetermined and supplied at a temperature of 42ºC. Mass flow rate of the water is regulated by a water circulator at a rate of 0.025 kg/s. The mass flow rate of the air is measured 0.0472 kg/s. It is found that the model which has two fin rows is more effective in terms of overall thermal conductance, number of transfer units and effectiveness at all values of inlet temperature of the air. In addition, both of the heat exchangers have the highest thermal performance at an air inlet temperature of 24ºC.

INTRODUCTION

One of the important application in compact heat exchanger design is the extended surfaces with multi-louvered fins. At first it is commonly used in the automotive industry to reduce the weight and the volume of the heat exchangers. Nowadays, louvered fin geometries are extensively used in the area of electronic devices, charge air coolers, evaporator and condensers to reduce the weight and size, also to save energy. Louvered fins provide more surface area relatively to the plain fins. The louvers create a series of thin boundary layer and interrupt the air flow. Therefore, the air side thermal resistance of the heat exchanger reduces and overall thermal performance increases. With the increasing of the application area of the louvered fins, compact heat exchangers with louvered fin have been studied by a number of researchers both experimentally and numerically. Many geometrical parameters belong to the louvered fin heat exchangers such as fin pitch, fin height, louver pitch, louver height, louver angle, tube pitch and flow depth have been focused on by the researchers. Related studies can be found in the open literature.

Aoki et al. [1] investigated the heat transfer characteristics of a louver fin array experimentally. They examined different louver angles and fin pitches. The results showed that heat transfer coefficients decreased with increasing fin pitch at a low Reynolds number. Dong et al. [2] performed series of experimental studies on the air side heat transfer and pressure drop characteristics of multi-louvered fin and flat-tube heat exchangers. Different Reynolds numbers with different fin pitch, fin height, fin thickness, fin louver angle and flow length were used for the wind-tunnel tests. It was found that the fin length and fin pitch are the major effects on the characteristics of the heat transfer and the pressure drop. It was found that the heat transfer coefficient decreased with increasing of the fin length and the fin pitch and decreasing of the fin height at the same frontal velocity. The pressure drop increased with the increasing of the fin length and the fin pitch and decreasing of the fin height. Kim and Bullard [3] investigated the air-side heat transfer and pressure drop characteristics of multi-louvered fin and flat-tube heat exchangers experimentally. Series of tests were conducted on the air-side Reynolds numbers of 100–600 for 45 heat exchangers with different geometrical parameters at a constant water flow rate. The results demonstrated that the flow depth is one of the important parameters in terms of friction factor. In addition, correlations for Colburn *j*-factor and Fanning friction factor *f* were developed for the considered geometries. Li and Wang [4] performed an experimental study on the air-side heat transfer and pressure drop characteristics of the heat exchangers with multi-louvered fins and flat tubes. Experiments were conducted for heat exchangers with different numbers of louver regions at the air-side Reynolds numbers of 400–1600. The air-side thermal performance data were analyzed by using the effectiveness-*NTU* method. Colburn-*j* factor and Fanning friction factor *f* were presented as a function of Reynolds number. It is found that the $j/f^{1/3}$ ratio decreased with the increasing of Reynolds numbers and increased with the increasing number of louver regions.

Lyman et al. [5] studied on the large-scale louver models with varied fin pitch and louver angle experimentally. A method was presented for evaluating the heat transfer coefficients using various reference temperatures to define the convective heat transfer coefficients. The results showed that the thermal field surrounding a particular louver is the major effect on the heat transfer from that louver. Park and Jacobi [6] studied the air-side thermal-hydraulic performance of flat-tube aluminum heat exchangers experimentally. The heat transfer and pressure drop were measured at frontal air velocities from 0.5 m/s to 2.8 m/s for dry and wet surface conditions. Parametric effects on the heat transfer and the friction factor were investigated for both dry and wet conditions. It was found that the louver spacing was a significant design parameter under wet conditions. Park and Jacobi [7] developed an air-side data analysis method for a flat-tube louvered-fin heat exchangers under partially wet conditions. Park and Jacobi [8-9] generated correlations for the Colburn-*j* and friction factor by using the largest database in the literature for the flat-tube louvered-fin heat exchangers.

Qi et al. [10] focused on the geometrical factors of the louvered fins including flow depth, ratio of fin pitch and fin thickness, tube pitch, number of louvers and angle of louver. Fifteen samples were used from the experimental data to analyze the heat transfer and the fluid flow characteristics by using the Taguchi method. The results showed that the contribution ratios to the overall performance of the flow depth, the ratio of the fin pitch and fin thickness and the number of the louvers are 31.57%, 21.53% and 20.34%, respectively. Chang and Wang [11] developed a generalized heat transfer correlation for louvered-fin geometry. This data bank consisted of 91 samples of louvered fin heat exchangers with different geometrical parameters. It was shown that 89.3% of the corrugated louver fin data are correlated within ± 15% with a mean deviation of 7.55%. Atkinson et al. [12] performed a detailed evaluation of 2D and 3D numerical simulations of flow and heat transfer over the louvered fins. Two 2D models were used, both of which incorporate the effects of tube surface area and fin resistance on the overall heat transfer rate. It was found that all the models gave accurate predictions of the pressure losses, but only 3D models are in good agreement with the experimental observations in terms of overall heat transfer. Hsieh and Jang [13] investigated the successively increased or decreased louver angle patterns by numerically. 3D numerical analysis of the heat transfer and the fluid flow were carried out. The results indicated that the successively variable louver angle patterns could effectively enhance the heat transfer performance. Malapure et al. [14] performed three-dimensional simulations of a single and double row tubes with louvered fins. Effects of the louver pitch, louver angle, fin pitch, tube pitch, and Reynolds numbers on the thermal-hydraulic performance of the air side were investigated. The computed Stanton numbers and friction factors were found to be in good agreement with the experimental data at low Reynolds number. In addition, both the Stanton number and the friction factor increased with the decrease in fin pitch.

Perrotin and Clodic [15] presented the results of 2D and 3D CFD models of compact louvered heat exchangers for the determination of heat transfer and pressure drop characteristics. They compared the 2D and 3D steady simulations with the experimental results and the correlations of the literature. It was found that the difference between the heat transfer coefficients obtained from the 2D simulations and the experimental data is up to 80%. In addition, the heat transfer coefficient calculated with the 3D models was much closer to the experimental data. Tafti and Cui [16] performed three-dimensional simulations of the louver–tube junction geometries to investigate the effect on the friction and the heat transfer characteristics. Three Reynolds numbers based on the bulk velocity and louver pitch were calculated. According to the three-dimensional results the flow acceleration had a large impact on louver heat transfer locally. Comparisons with correlations derived from experiments showed that the computational modeling of a small subsystem can be used reliably to extract the performance data for the full heat exchanger.

Uğurlubilek et al. [17-18] investigated the effect of louver angle on the heat transfer and the pressure drop characteristics of mini channel flat-tube with louvered fin heat exchanger. Numerical simulations were performed for different louver angles at constant wall temperature boundary condition. The result showed that the pressure drop increases with the increasing of the louver angle which create more resistance to the flow. The pressure drop took its lowest value on which the geometry has the smallest louver angle of 20°. The rate of heat transfer took its highest value for the geometry having a louver angle of 26°. Furthermore, it was also seen that the relation between the louver angle and the heat transfer was not linear. Akyüz [19] conducted a numerical investigation for the effects of fin pitch and fin height on the thermo-hydraulic performance of an air-cooled, flat-tube heat exchanger. The thermo-hydraulic performance of the heat exchanger was evaluated using the performance factor $j/f^{1/3}$. Fin pitches of 1.50, 2.00 and 2.50 mm and fin heights of 8, 10, 12, 16 and 20 mm were studied. It was stated the model with a fin pitch of 1.50 mm, and a fin height of 8 mm has the best overall performance in the studied cases.

The present study investigates experimentally the thermal performance for two mini channel flat-tube heat exchangers for different inlet temperatures of the cold fluid. The heat exchangers used in the tests have identical size but different heat transfer area on the air side due to the different louvered fin row configuration. The thermal performance of the heat exchangers is compared by using both *LMTD* and *effectiveness-NTU* method.

EXPERIMENTAL SETUP

Test Apparatus

In this study, the thermal performances of two mini channel flat-tube heat exchangers with louvered fins have been studied experimentally at three different (15°C, 24°, 33°C) inlet temperature of the cold fluid (air). A plexi-glass wind-tunnel which has a 1m×1m cross-section and 2m long is located in an insulated test room as shown in Fig. 1. The dimensions of constant temperature room are 4.5m×3.5m×2.3m. The inlet condition of the air-side of the heat exchanger is maintained constant by controlling the temperature of the test room. The steady-state values are used to calculate the thermal performance of the heat exchangers. The experimental data are collected with a data-acquisition system. The experiments are repeated three times to get the average results. The temperature control of the test room is obtained by using an electric resistance heater and a refrigeration system using R-22 on the top of the perforated chrome ceiling. The wind-tunnel system is designed to suck the test room air over the louvered fin side of the heat exchangers by a centrifugal fan as shown in Fig. 1. The heat exchanger height is less than that of the tunnel inlet dimensions. Therefore, the bypass flow is eliminated by a thin layer of foam. Firstly, room air is sucked by the fan and air flow is forced to pass through the louvered fins. The inlet and exit temperatures across the air side of the heat exchangers are measured by T-type thermocouple grid. Both the inlet and the outlet temperature grids consist of four thermocouples in an evenly spaced array. Each thermocouple value is recorded with a data-acquisition system, and their average values are used as the cold fluid inlet and outlet temperature. After the air flow passes through the tested heat exchangers, it passes through a three layered screen set, nozzle set and again a three layered screen set, respectively. The screen sets are used to get uniform flow at the inlet and the exit of the nozzle set. At the inlet and the exit of the nozzle set, the pressure drop of air is measured for each surface of the wind-tunnel by digital manometers, and their average values are recorded. The mass flow rate of the air is measured in terms of the pressure drop across the nozzle set according to ASHRAE Standard 41.2. In this work, the pressure drop at the nozzle is observed as 20 Pa. The mass flow rate of air is 0.047 kg/s according to this pressure drop value. The mass flow rate and the inlet temperature of hot fluid (water) are regulated by a water circulator. The water is heated up to the temperature of 42°C and pumped to the mini channel flat-tube heat exchanger. The mass flow rate of the water is 0.025 kg/s. The water inlet and exit temperature is measured with T type thermocouples at the inlet and outlet port of the heat exchanger.

Uncertainty of the Test Apparatus

Standard error propagation rules, as described by Taylor and Kuyatt [20], are used to determine the total uncertainty by using the EES (Engineering Equation Solver). The uncertainties of all measured parameters are summarized in Table 1. Uncertainties of the average heat transfer rate Q, overall thermal conductance UA, number of transfer units NTU, and effectiveness ε are calculated about 3.88 %, 3.88%, 4.18% and 3.84 %, respectively.

Table 1. Summary of the uncertainty analysis

Parameters	Uncertainty
Air inlet temperature	0.4%
Air outlet temperature	0.4%
Mass flow rate of air	0.2%
Water inlet temperature	0.4%
Water inlet temperature	0.4%
Mass flow rate of water	0.15%

The Types of Test Heat Exchanger

In this study, two mini channel flat-tube heat exchangers with multi louvered fins are tested. Heat exchangers have identical frontal area of 160mm×160mm. As shown in Fig. 2 and Fig. 3, test heat exchangers are called Type-I and Type-II. Mini channel flat-tube is serpentine shaped and Type-I and Type-II have 9 and 7 tube passes, respectively. Type-I has one intermediate plate and two rows of louvered fins between the serpentine flat tubes. Type-II has two intermediate plates and three rows of louvered fins between the serpentine flat tubes. The terminology of test heat exchangers is given with Fig. 4. In Fig. 4, cross- section of *A-A* shows that the louvered fin geometry is the same for both Type-I and Type-II. However, the air side heat transfer areas are different, due to the louvered fin configuration between the serpentine flat tubes. Table 2 shows the similarities and differences of the heat exchangers in terms of geometrical properties. The notations used by Kays and London [21] are followed throughout the figures, tables and calculations.

Table 2. Geometric properties of the test heat exchangers

Property	Type-I	Type-II
F_h [mm]	8.2	8.2
F_p [mm]	1.5	1.5
L_h [mm	6	6
L_α [°]	26	26
L_p [mm]	1.1	1.1
F_d [mm]	16	16
P_t [mm]	0.6	0.6
T_p [mm]	19.4	28.2
a [mm]	17	-
b [mm]	-	25.8
A [m^2]	0.383	0.316
Number of tube pass [-]	9	7
Number of fin row [-]	16	18
Total length of flat-tube [mm]	353	331

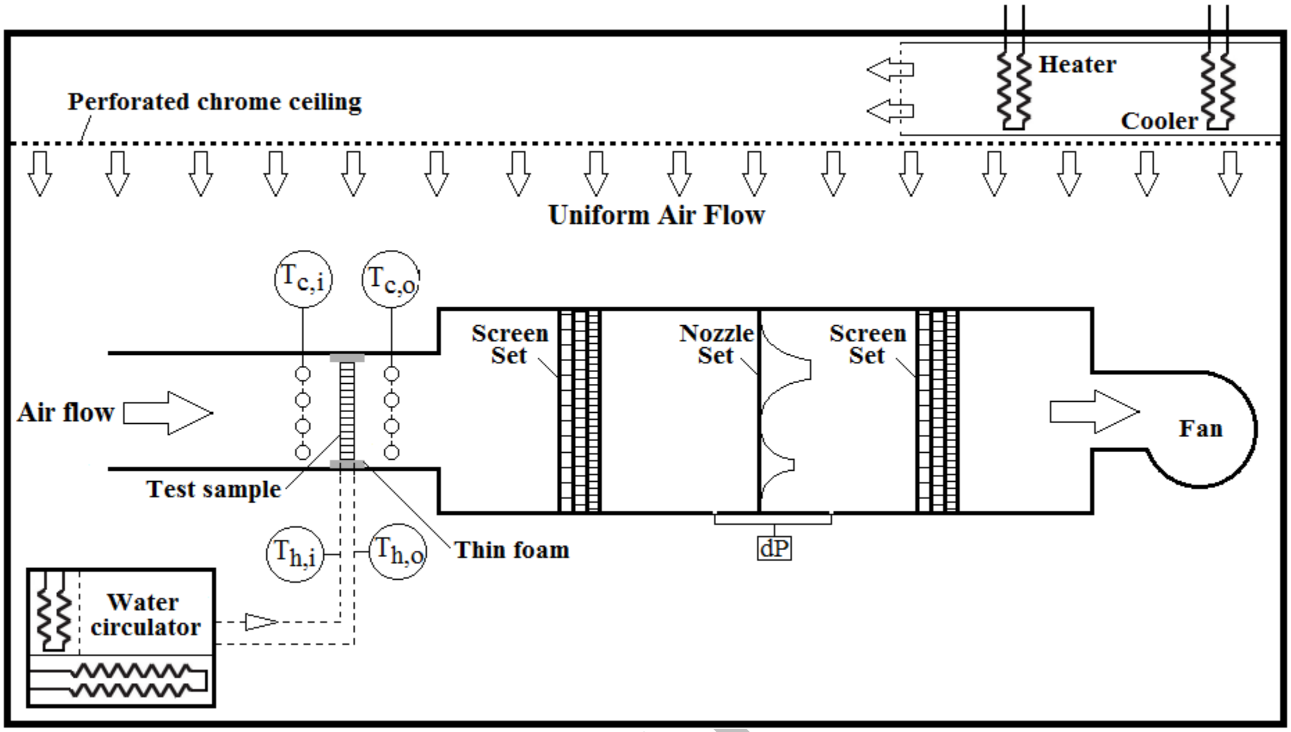

Fig. 1. Constant temperature test room, wind-tunnel and test apparatus

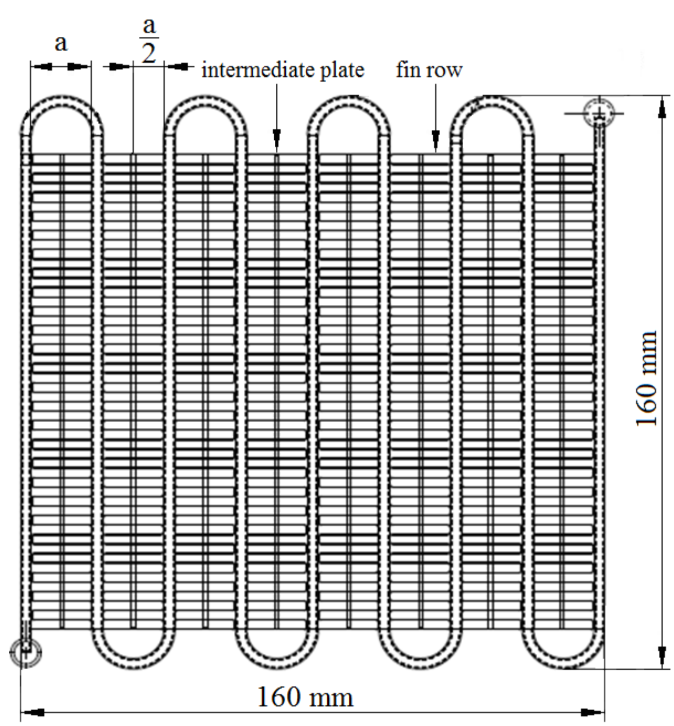

Fig. 2. Type-I

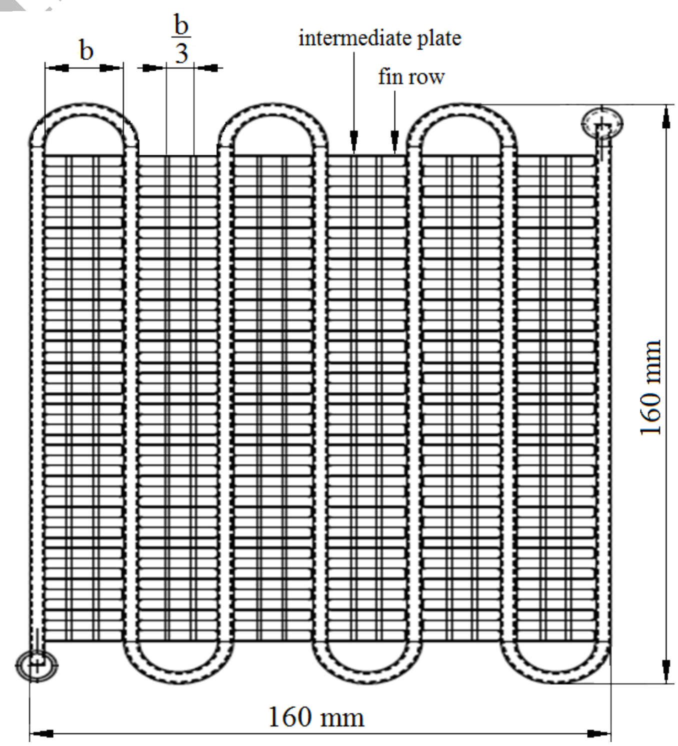

Fig. 3. Type-II

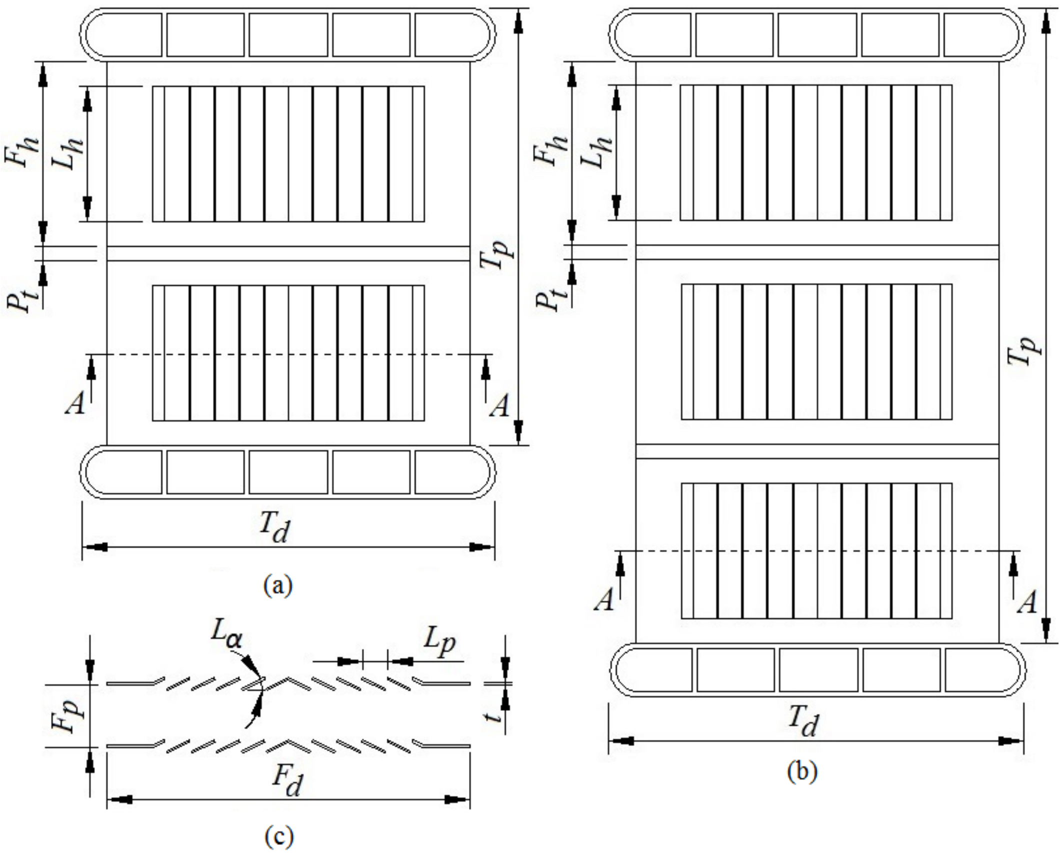

Fig. 4. Definition of the geometrical terminology of the heat exchangers
(a) Type-I (b) Type-II (c) Cross-sectional view of *A-A* for both models

Data reduction

Average heat transfer rate from the heat exchanger is given in Eq. 1.

$$Q = \frac{Q_c + Q_h}{2} \tag{1}$$

The heat transfer rate to the cold fluid can be calculated from the temperature increase at the air side as shown in Eq. 2. Similarly, heat transfer from the hot fluid can be calculated from the temperature decrease at the water side as shown in Eq. 3.

$$Q_c = C_c (T_{c,out} - T_{c,in}) \tag{2}$$

$$Q_h = C_h (T_{h,in} - T_{h,out}) \tag{3}$$

where C_c and C_h are the heat capacities of the air and the water as $C_c = m_c c_{p,c}$ and $C_h = m_h c_{p,h}$, respectively. Specific heat of the water is assumed as 4.178 kJ/kg°C, due to the small temperature difference at the hot side.

Variation of the temperature is taken into consideration on the cold side and the specific heat of the air is calculated as follows [22].

$$\begin{aligned} c_{p,c} = \left(\frac{8.31447}{28.97}\right)\Bigg[& 3.653 + \left(\frac{T_{c,in}+T_{c,out}}{2}\right)\left(-1.337\times10^{-3}\right) \\ & + \left(\frac{T_{c,in}+T_{c,out}}{2}\right)^2\left(3.294\times10^{-6}\right) + \left(\frac{T_{c,in}+T_{c,out}}{2}\right)^3\left(-1.913\times10^{-9}\right) \\ & + \left(\frac{T_{c,in}+T_{c,out}}{2}\right)^4\left(0.2763\times10^{-12}\right)\Bigg] \end{aligned} \tag{4}$$

The overall thermal conductance of the heat exchangers is calculated as follows.

$$UA = \frac{Q}{\Delta T_m} \tag{5}$$

where ΔT_m is the logarithmic mean temperature difference given as;

$$\Delta T_m = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\dfrac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)} \tag{6}$$

Number of transfer units (*NTU*) is calculated according to the following equation.

$$NTU = \frac{UA}{(mc_p)_{\min}} \tag{7}$$

The effectiveness method can be used to compare the heat exchangers by using the equations for the unmixed fluid [23],

$$\varepsilon = 1 - \exp\left[\frac{NTU^{0.22}}{C_r}\left\{\exp\left(-C_r NTU^{0.78}\right) - 1\right\}\right] \tag{8}$$

where;

$$\varepsilon = \frac{Q}{Q_{\max}} \quad \text{and} \quad C_r = \frac{(mc_p)_{\min}}{(mc_p)_{\max}} \tag{9}$$

RESULTS AND DISCUSSION

The exit temperatures of the cold and the hot fluids are measured for three different cold side inlet temperatures. The measurements are given in Table 3. The two types are compared by using the most important parameters of the heat exchanger; namely the overall thermal conductance, the number transfer units, and the effectiveness. First the logarithmic mean temperature difference ΔT_m, and the average heat transfer rate Q are calculated by using Eq. 5 and Eq. 1, respectively. They are presented in Table 4. Then the overall thermal conductance *UA* and the number of transfer units *NTU* are obtained via the Eq. 5 and Eq. 7, respectively. Table 4 includes these parameters as well as the effectiveness ε, obtained by Eq. 8 for both types calculated at three different inlet conditions.

The variation of the thermal conductance of both types can be observed in Fig. 5. At 24°C condition, the highest overall thermal conductance of 18.05 W/°C is obtained for Type-I. At the same time, it is indicated that, the overall thermal conductance of Type-I is higher at any inlet temperature of the air. The heat transfer area designates the physical size of a heat exchangers. Although the outer frames of the heat exchangers are the same, the configurations provide different heat transfer areas.

As it is seen in Table 2, Type-I and Type-II specimens have the total heat transfer area of 0.383 m^2 and 0.316 m^2, respectively. In the definition of the overall thermal conductance, the area *A* is included. It can understand that the greater heat transfer area provides greater overall thermal conductance, but an assumption of a monotonic parametric effect can cause misinterpretations. The overall thermal conductance first increases with the increasing of the inlet temperature of the cold fluid from 15°C to 24°C, then decreases with the increasing of the inlet temperature of the cold fluid from 24°C to 33°C for both heat exchangers. Due to the combined effects of the geometrical and the operational parameters, such results are frequently observed also in the literature [6-10].

Table 3. Experimental results

	Test	$T_{c,i}$ [°C]	$T_{c,o}$ [°C]	$T_{h,i}$ [°C]	$T_{h,o}$ [°C]
Type - I	1	14.34	22.18	42.76	39,77
	2	13.79	21.75	42.70	39,54
	3	15.56	22.97	42.95	39,97
	Avg.	14.56	22.30	42.80	39,76
	1	24.22	29.12	41.89	39,25
	2	24.46	29.07	41.80	39,09
	3	24.70	29.15	41.77	38,80
	Avg.	24.46	29.11	41.82	39,05
	1	31.74	34.29	41.59	40,54
	2	32.73	35.07	41.70	40,75
	3	32.59	34.96	41.74	40,76
	Avg.	32.35	34.78	41.68	40,68
Type - II	Test	$T_{c,i}$ [°C]	$T_{c,o}$ [°C]	$T_{h,i}$ [°C]	$T_{h,o}$ [°C]
	1	14.83	20.60	42.30	39.32
	2	14.66	20.50	42.28	39.25
	3	15.00	20.84	42.29	39.22
	Avg.	14.83	20.65	42.29	39.26
	1	24.11	27.67	42.79	40.31
	2	23.55	27.24	42.83	40.16
	3	23.73	27.32	42.83	39.93
	Avg.	23.80	27.41	42.82	40.14
	1	32.66	34.37	42.39	41.03
	2	32.83	34.45	42.41	41.05
	3	32.81	34.44	42.41	41.03
	Avg.	32.76	34.42	42.40	41.03

Fig. 6 is prepared for the comparison of *NTU* values of two types. *NTU* indicates the thermal size of the heat exchanger and provides a compound measure of the total heat transfer area *A*, the overall heat transfer coefficient *U* and the minimum heat capacity rate C_{min}. Since *U* is not constant, the definition of *NTU* should be considered as;

$$NTU = \frac{1}{C_{\min}} \int_A U dA \tag{10}$$

In this study, C_{min} is approximately constant. Therefore, the variations of *NTU* have similar trends with respect to the *UA* curves. Type-I has higher *NTU* than Type-II at any inlet temperature of the air as seen in Fig. 6.

Table 4. System parameters

	Test	$T_{c,i}$ [°C]	ΔT_m [°C]	Q [W]	Q_{max} [W]	UA [W/°C]	C_h [W/°C]	C_c [W/°C]	C_r	NTU	ε
Type - I	1	14.34	22.92	341.73	1346.62	14.91	104.50	47.383	0.453	0.315	0.250
	2	13.79	23.27	353.51	1369.81	15.19	104.50	47.382	0.453	0.321	0.254
	3	15.56	22.12	330.97	1297.85	14.95	104.50	47.384	0.453	0.316	0.250
	Avg.	14.56	22.77	342.07	1338.09	15.02	104.50	47.383	0.453	0.317	0.251
	1	24.22	13.87	239.72	837.58	17.28	104.50	47.401	0.454	0.365	0.281
	2	24.46	13.66	246.38	821.92	18.04	104.50	47.400	0.454	0.381	0.290
	3	24.70	13.35	251.29	809.10	18.83	104.50	47.399	0.454	0.397	0.300
	Avg.	24.46	13.62	245.80	822.86	18.05	104.50	47.400	0.454	0.381	0.290
	1	31.74	8.03	115.71	467.01	14.39	104.50	47.412	0.454	0.303	0.243
	2	32.73	7.30	105.07	425.30	14.36	104.50	47.414	0.454	0.303	0.242
	3	32.59	7.46	107.62	442.37	14.41	104.50	47.414	0.454	0.304	0.243
	Avg.	32.35	7.60	109.47	444.89	14.38	104.50	47.414	0.454	0.303	0.243
Type - II	Test	$T_{c,i}$ [°C]	ΔT_m [°C]	Q [W]	Q_{max} [W]	UA [W/°C]	C_{max} [W/°C]	C_{min} [W/°C]	C_r	NTU	ε
	1	14.83	23.06	292.22	1301.58	12.67	104.50	47.382	0.453	0.267	0.219
	2	14.66	23.15	297.12	1308.66	12.82	104.50	47.381	0.453	0.271	0.221
	3	15.00	22.81	298.42	1293.05	13.08	104.50	47.382	0.453	0.276	0.225
	Avg.	14.83	23.01	295.92	1301.09	12.86	104.50	47.382	0.453	0.271	0.221
	1	24.11	15.65	213.99	885.38	13.64	104.50	47.397	0.454	0.288	0.232
	2	23.55	16.10	227.13	913.79	14.10	104.50	47.396	0.454	0.297	0.239
	3	23.73	15.85	236.29	905.26	14.89	104.50	47.396	0.454	0.314	0.249
	Avg.	23.80	15.87	225.80	901.48	14.21	104.50	47.396	0.454	0.300	0.240
	1	32.66	8.20	111.85	461.33	13.61	104.50	47.413	0.454	0.287	0.232
	2	32.83	8.09	109.57	454.23	13.51	104.50	47.414	0.454	0.285	0.230
	3	32.81	8.09	110.75	455.17	13.68	104.50	47.414	0.454	0.289	0.233
	Avg.	32.76	8.13	110.72	456.91	13.60	104.50	47.414	0.454	0.287	0.232

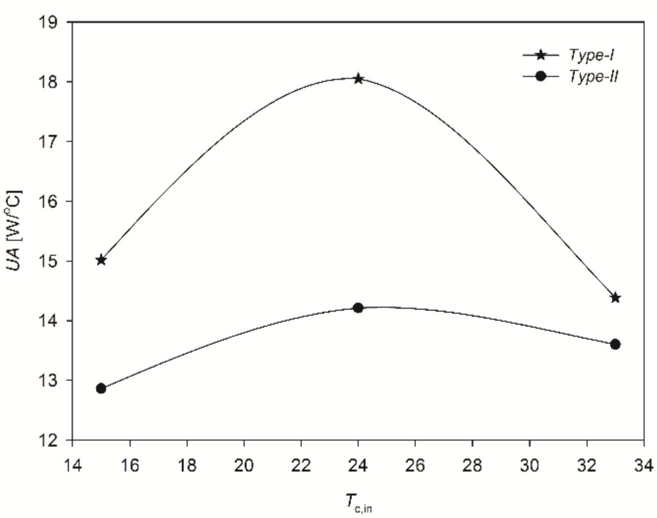

Fig. 5. Variation of the overall thermal conductance with respect to the inlet temperature of air

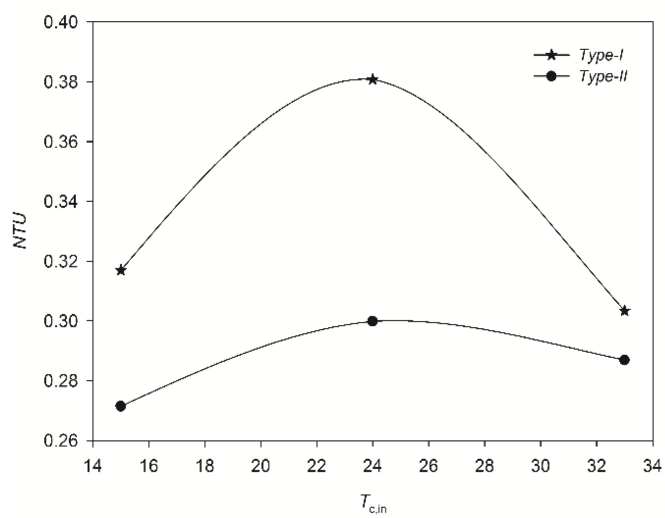

Fig. 6. Variation of the number of transfer units with respect to the inlet temperature of air

The effectiveness of a heat exchanger is important to learn the ratio of the actual heat transfer rate to the possible maximum heat transfer rate which depends on the overall effect of the operating conditions and design parameters. Fig. 7 presents the effectiveness changing with respect to the inlet temperature of the air. The maximum possible heat transfer changes with the inlet conditions although the flow rates stay constant. In this study, the actual heat transfer rate is considered as the average of the heat transfer rates of both the cold and hot sides.

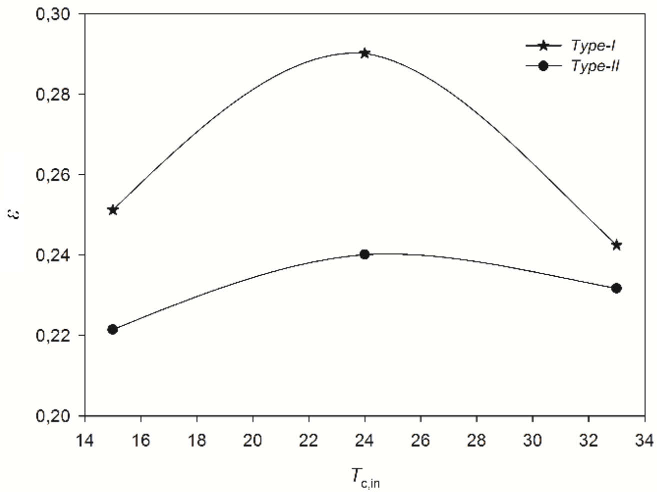

Fig. 7. Variation of effectiveness with respect to the inlet temperature of air

Additionally, the heat transfer rate inevitably includes the inlet temperature of the cold fluid. The actual and the maximum heat transfer rates under these conditions give rise to the change in the effectiveness seen in Fig. 7. The final evaluation of the effectiveness of two types is that the heat exchanger of the Type-I has higher effectiveness with respect to the effectiveness of Type-II due to the change of the geometry.

To get a clear comparison between the heat exchangers, it is quite valuable to use the normalized values of parameters like the overall thermal conductance (UA), number of transfer units (NTU), or effectiveness (ε) for different structures of the heat exchangers from the designer's point of view. Normalized values ($\overline{UA}, \overline{NTU}, \bar{\varepsilon}$) are obtained such that the average value of the selected parameter divided by the maximum value of the related results of both types. They are presented in Fig. 8.

The highest normalized thermal conductance is obtained in the Type-I for the inlet temperature of 24ºC of the air. At any temperature of the cold fluid, the difference between the types is remarkable and the Type-I is advantageous. The difference between the $\overline{UA}$ values of both types at the higher inlet temperature is less with respect to the differences at lowest inlet temperatures. The normalized number of transfer unit values are parallel to that of the thermal conductance due to the definition. As a design parameter $\overline{NTU}$ values also indicate the Type-I having larger heat transfer area. Note that the perfect heat exchanger requires higher effectiveness. When the $\bar{\varepsilon}$ values are compared, the Type-I becomes prominent in all cases.

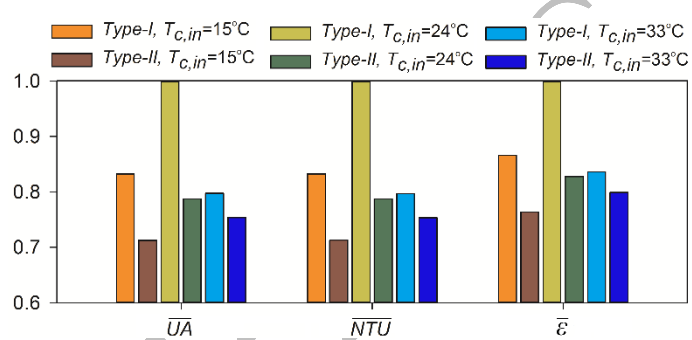

Fig. 8. Normalized parameters

CONCLUSION

The effect of the cold fluid inlet temperature is investigated experimentally using two different mini channel flat-tube heat exchangers with multi-louvered fins. Even though the heat exchangers have identical size, due to the different louvered fin row configuration they end up with different surface areas. The thermal performance of the heat exchangers is compared by using both *LMTD* and *effectiveness-NTU* method.

The heat exchanger named Type-I has higher thermal performance in terms of the overall thermal conductance, the number of transfer units and the effectiveness at any inlet temperature of the air. This may be a consequence of the total length of the serpentine flat-tube, despite the number of fin rows of the Type-I has less than the Type-II. Especially, at an inlet temperature of 15ºC and 24ºC, the thermal performance of the Type-I is greater than the Type-II. The difference is about 10% and 20% at an inlet temperature of 15ºC and 24ºC, respectively. Therefore, it is recommended from heat transfer performance point of view. At an inlet temperature of 33ºC, the difference between the thermal performances of the heat exchangers decreases about to 4%. Two types should be evaluated by economic constraints at higher inlet temperatures of air. Since the number of fins, the length of the flat tubes and the number of intermediate plates between the fin rows are different, the cost of both types will be different.

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NOMENCLATURE

2D	two-dimensional
3D	three-dimensional
A	air side heat transfer area, m^2
a	tube clearance for Type-I, mm
b	tube clearance for Type-II, mm
C_c	heat capacity rate of cold fluid, W/^{o}C
C_h	heat capacity rate of hot fluid, W/^{o}C
C_r	heat capacity ratio
$c_{p,c}$	specific heat of cold fluid, J/(kg ^{o}C)
$c_{p,h}$	specific heat of hot fluid, J/(kg ^{o}C)
F_d	flow depth, mm
F_h	fin height, mm
F_p	fin pitch, mm
h_c	heat transfer coefficient, W/(m^{2o}C)
L_h	louver height, mm
L_p	louver pitch, mm
L_α	louver angle, ^{o}
m_c	mass flow rate of cold fluid, kg/s
m_h	mass flow rate of hot fluid, kg/s
NTU	number of transfer unit
P	Pressure, Pa
P_t	intermediate plate thickness, mm
Q	average heat transfer rate, W
Q_c	cold fluid heat transfer rate, W
Q_h	hot fluid heat transfer rate, W
t	fin thickness, mm
$T_{c,in}$	inlet temperature of cold fluid, ^{o}C
$T_{c,out}$	outlet temperature of cold fluid, ^{o}C
T_d	tube depth, mm
$T_{h,in}$	inlet temperature of hot fluid, ^{o}C
$T_{h,out}$	outlet temperature of hot fluid, ^{o}C
T_p	tube pitch, mm
ΔT_m	logarithmic mean temperature difference, ^{o}C
UA	Overall thermal conductance, W/ ^{o}C

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