FANNING FRICTION (*f*) AND COLBURN (*j*) FACTORS OF A LOUVERED FIN AND FLAT TUBE COMPACT HEAT EXCHANGER

by

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In the present study, the heat transfer and pressure drop characteristics of air over the louvered fins in a compact heat exchanger, used as a radiator in the automobiles have been experimentally investigated. The experiments were conducted at various flow rates of air and the results showed a decrease in goodness factor of 22.7% with respect to increase in Reynolds number from 231 to 495. The experimental results were compared with the CFD results and the f and j factors from the CFD analysis are in good agreement with the experimental data. Also, the experimental f and j factors were compared with the predicted values from the available correlations in the literature for the louvered fin and tube compact heat exchangers. The large deviation of the predicted results revealed that the correlations are not reliable for the design of the compact heat exchanger. Hence, the CFD analysis is more advantageous for the optimal design of compact heat exchanger, which also reduces the experimentation time and cost.

Key words: *louvered fin, pressure drop, goodness factor, compact heat exchangers, CFD*

Introduction

Heat exchangers are one of the vital components of any energy systems in various industrial sectors like refrigeration, automotive, chemical, manufacturing and electronic cooling, *etc.*, to transfer heat from a hot fluid to cold fluid across an impermeable wall. The quantity of heat transfer depends on the temperature difference between two fluids, surface area, conductive resistance of the wall, and flow nature of the fluids. It is desired to design the heat exchangers with minimum volume and weight for transferring the required heat, particularly in the aviation and automobiles where the space is the major constrain. In order to achieve the previous task, the compact heat exchangers having surface density greater than 700 m^2/m^3 are widely used in which the flow passages of the fluids are small. Among these compact heat exchangers, the plate fin heat exchangers find the extensive variety of applications like radiator, evaporator, condenser, and oil cooler with air as one of the heat transfer fluid [1]. The major problem encountered in the compact heat exchanger is the predominant thermal resistance on

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the air side that accounts for nearly 80% of total thermal resistance in the heat exchangers [2, 3]. This lowers the overall performance of the heat exchanger, hence, it is required to enhance the air side heat transfer coefficient through the conventional techniques [4]. Among these, the use of finned surfaces on the air-side is normally used to enhance the overall thermal performance of the compact heat exchangers by providing the increase in surface area and inducing the turbulence mixing of air flow.

There are many types of fins such as plain fin, perforated fin, wavy fin, offset fin, and louvered fin. The louvered fin is more preferable for high interruption in the air flow and ability to create a series of thin boundary layers, in addition to its ease of manufacture and low cost. The researchers have focused on the thermal performance of the compact heat exchangers with louvered fins on different tube geometries. The literature pertaining to the previously mentioned are summarized. Beauvais [5] explored that the louvered fins act as the multiple flat plates in breaking the thermal boundary layer. Achaichia and Cowell [6] experimentally investigated the heat transfer and pressure drop characteristics of a flat tube and louvered plate fin surfaces. They reported the variation of Stanton number and friction factor as a function of Reynolds number and proposed heat transfer and friction correlations using the data bank. Wang *et al*. [7] proposed the general heat transfer and friction correlations for louver geometry having round tube configuration. The air side pressure drop in a multi louver flat tube heat exchanger was analyzed by Kim and Bullard [8] with both fluids unmixed conditions. The results revealed that the flow depth is one of the important parameters in influencing the pressure drop. Davenport [9] studied the characteristics of a non-standard variant of the flat tube and corrugated louvered fin and developed the correlations for (*f*) and (*j*) factors based on the experimental data. It was also reported that the flow alignment with the louvers resulted with the reduction in the thermal boundary layer thickness with respect to Reynolds number. In addition to the previously mentioned, several (*f*) and (*j*) correlations were developed by Dong *et al*. [10], Chang and Wang [11], Chang *et al*. [12], Li and Wang [13], and Sunden and Svantesson [14]. The effects of louver geometry and air flow on the thermal performance of the compact heat exchangers were analyzed by Vaisi *et al*. [15] and Khaled *et al*. [16], respectively. Recently, the thermo-hydraulic performance of a compact fin and tube heat exchanger with different tube configurations [17] and Titanium brazed plate fin heat exchanger [18] were analysed. Also, the researchers attempted to enhance the performance of compact heat exchanger through evaporation cooling and predicted the thermo-hydraulic performance by an artificial neural network model [19].

It is observed from the previous literature that the efficient design of heat exchangers involves several geometrical parameters such as fin pitch, transverse tube pitch, flow length, louver pitch, and louver angle, *etc.*, Considering the previously mentioned critical issues, the researchers have developed correlations for (*f*) and (*j*) factors based on their experimental and CFD results to save the experimentation time and cost. However, the predicted values from the available correlations showed considerable deviation for different geometrical configurations that makes more uncertainty on the applicability of the correlation towards the design of compact heat exchanger. Accordingly, the objective of the present research work is to evaluate the accuracy of the existing (*f*) and (*j*) correlations by conducting experimental and CFD analysis for a compact heat exchanger of particular configuration.

Experimental set-up

Figure 1 show the schematic arrangement of the experimental set-up and it consists of a compact heat exchanger (test radiator), hot water tank, centrifugal pump, blower, wind tunnel, and flow control valve. The test radiator is a cross flow type compact heat exchanger,

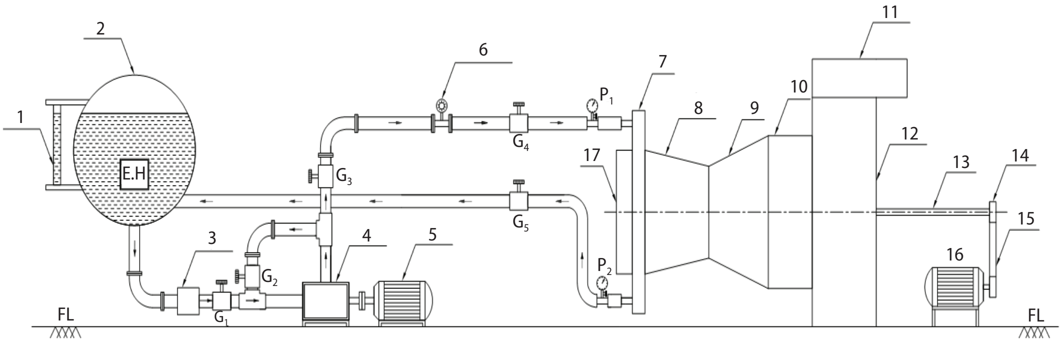

Figure 1. Schematic arrangement of the experimental set-up;
1 – water level indicator, 2 – hot water tank, 3 – mud box, 4 – pump, 5 – motor, 6 – flow control valve, 7 – radiator, 8 – tunnel, 9 – transition piece, 10 – circular passage, 11 – outlet duct, 12 – blower, 13 – shaft, 14 – pulley, 15 – belt, 16 – motor, 17 – rectangular duct, G1-G5 – gate valves, P1 and P2 – pressure gauges, EH – electrical heaters, and FL – floor level

in which water flows inside the tubes and air flows over the tubes through louvered fins. The radiator core is made of alternate layers of 75 numbers of louvered fins and 148 numbers of flat tubes with a core size of 810 × 717 × 52 mm. The louvers are trapezoidal in shape and each side of the fin has 27 louvers. The details of the louvered fin and flat tube geometry are shown in tab. 1. The hot water tank is fitted with twelve electrical heaters of capacity each 6 kW and the resistance temperature detector (RTD) of PT100 type with an accuracy of ±0.1 °C. The temperature of the water in the tank is maintained at a desired temperature of 90 °C throughout the experiment by controlling the power input to the heaters based on the temperature measured by the RTD. A centrifugal pump circulates hot water into the tube of the radiator and the flow rate is measured by a flow meter (MAGFLOW 5100W) with an accuracy of ±0.5%. The inlet of the tunnel is rectangular in the cross section with an area equal to the frontal area of the radiator core and the outlet of the tunnel is square in cross-section. The dampers are mounted at the outlet of the wind tunnel and they are arranged radially around the rotor in order to vary the frontal air velocity. A centrifugal blower sucks the air through the radiator core and the air temperature is continuously measured at the entry and exit of the radiator by using the RTD (PT100) with an accuracy of ±0.15 °C. The pressure transducers are used to measure the pressure drop of air and water across the heat exchanger.

Table 1. Specification of flat tube and louvered fin

Geometric parameters			
Flat tube		Louvered fin	
Length	822 mm	Fin pitch	1.5 mm
Width	23.45 mm	Fin thickness	0.1 mm
Wall thickness	0.32 mm	Plate spacing	7.6 mm
Diameter	2 mm	Louver pitch	1.2 mm
Transverse tube pitch	7.6 mm	Louver height	0.284 mm
Longitudinal tube pitch	28 mm	Louver angle	26°
Number of tubes	148	Flow passage hydraulic diameter	1.9 m
Number of longitudinal tube rows	2		

The radiator to be tested is fixed at the inlet of the wind tunnel in the experimental set-up, and it is ensured that there are no air leakages. The heaters in the hot water tank are switched on, and once the water in the boiler reaches the required temperature of 90 °C, both the centrifugal pump and the blower are made operative. The mass flow rate of water through the radiator is regulated with the aid of the flow control valve and the required frontal air velocity is achieved with the aid of a damper adjusting lever. The experiments were conducted for four different mass flow rates of water (1.25, 1.5, 1.83, and 2.25 kg/s) and for each mass flow rate of water, the air velocity was varied from 3.5-7.5 m/s with a step size of 1 m/s. The temperature and pressure drop of both the streams of fluids across the test radiator were continuously monitored and recorded using the data acquisition system, after the system attained the steady-state condition. The experiment trials were repeated thrice for each experimental condition to ensure the repeatability of the results.

Data analysis

In this section, the heat transfer and flow characteristics of the test radiator are presented in terms of the Colburn (*j*) factor and Fanning friction (*f*) factor with respect to Reynolds number. The equations employed in evaluation of the Fanning friction (*f*) factor and Colburn (*j*) factor are given. The hydraulic diameter of the louvered fin is calculated from:

$$D_h = \frac{4LA_{\min}}{A_s} \tag{1}$$

where D_h, L, $A_{\min}$, and A_s represent the hydraulic diameter, flow length or heat transfer matrix depth in the air flow direction, minimum free flow area, and the total area for heat transfer on the air side, respectively. The dimensionless Reynolds number based on louver pitch is calculated from:

$$\mathrm{Re}_{Lp} = \frac{GL_p}{\mu} \tag{2}$$

where L_p is louver pitch. The dimensionless Reynolds number based on hydraulic diameter is determined by using the equations:

$$\mathrm{Re}_{D_h} = \frac{GD_h}{\mu} \tag{3}$$

$$G = \frac{\rho A_f v}{A_{\min}} \tag{4}$$

where A_f and G represent the frontal area of the heat exchanger and the mass flux or mass velocity, respectively. Fanning friction (*f*) factor is calculated from:

$$f = \left(\frac{\Delta P}{2L}\right)\left(\frac{D_h}{\rho_a v^2}\right) \tag{5}$$

where ΔP, ρ_a, and v denote air-side pressure drop, density of air, and inlet (frontal) air velocity, respectively. The dimensionless Colburn (*j*) factor is evaluated using the equation:

$$j = \mathrm{StPr}^{2/3} = \left(\frac{D_h}{4L}\right)\ln\left(\frac{T_i - T_w}{T_o - T_w}\right)\mathrm{Pr}^{2/3} \tag{6}$$

where St, Pr, T_i, T_0, and T_w represent the Stanton number, Prandtl number, the inlet air, the outlet air temperatures, and the tube wall temperature, respectively. The results of the uncertainty analysis are given in tab. 2.

Table 2. Results of uncertainty analysis

Measured data		Derived data	
Temperature	±0.15 °C	Hydraulic diameter	±1.67%
Air velocity	±0.14%	Mass velocity	±1.6%
Air side pressure drop	±0.09%	Heat transfer coefficient	±3.6%
Water mass flow rate	±0.5%	Fanning friction factor (*f*)	±2%
		Colburn Factor (*j*)	±3.2%

Results and discussion

In order to ensure the accuracy and reliability of the experimental set-up, the experimental data such as air side temperature difference, pressure drop, average heat transfer co-efficient, (*f*) and (*j*) factors, and volume goodness factors are validated with the corresponding CFD results.

Airside temperature and pressure difference

Figure 2 illustrates the computational domain considered for the CFD analysis and the detailed procedure adopted for the CFD analysis was presented by Karthik *et al*. [20]. Figure 3

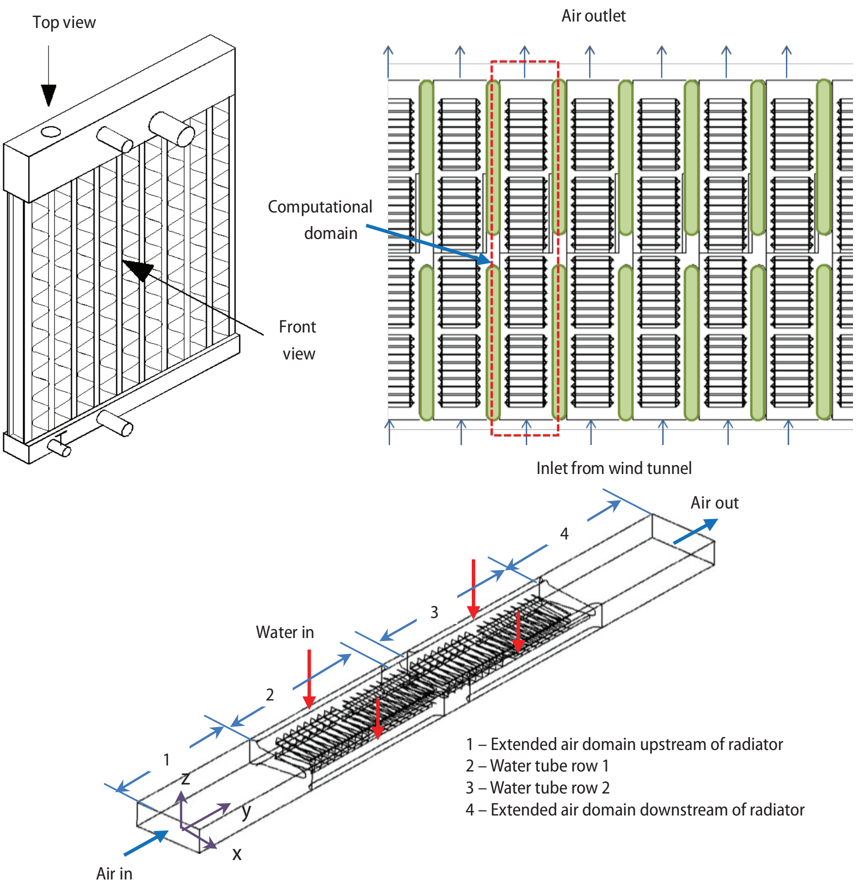

Figure 2. Details of computational domain

shows the comparison of the air side temperature difference obtained from the CFD analysis with the experimental data. The experimental results pertaining to three different conditions as shown in tab. 3 are validated with the CFD results. The percent deviations of the temperature values between the experimental and CFD results for three different validation cases (VC) are 11.05%, 14.28%, and 15.89%, respectively. The deviation could be due to the uncertainties in the experimental measurements and also by the numerical errors attributed to the turbulence model employed. Figure 4 compares the pressure drop across the heat exchanger for various inlet air velocities ranging from 3.5-7.5 m/s with the results obtained from the CFD analysis. It has been observed from the figure that the experimental results are in close agreement with the CFD results and the trend confirms the general characteristic curve of a typical compact heat exchanger. The pressure drop increases from 50-400 Pa for the variation in the inlet air velocity from 3.5-7.5 m/s. The increase in pressure drop is due to the presence of louvers and increase in the mass flow rate of air with respect to frontal air velocity, which in turn augment the air side pressure drop.

Table 3. Experimental data sets used for CFD validation

Validation cases	Air velocity [ms^{-1}]	Inlet air temperature, [K]	Water flow rate [kgs^{-1}]	Inlet water temperature, [K]
VC1	2.5	310	0.004215	363
VC2	5.6	308	0.006181	363
VC3	7.5	302	0.007586	363

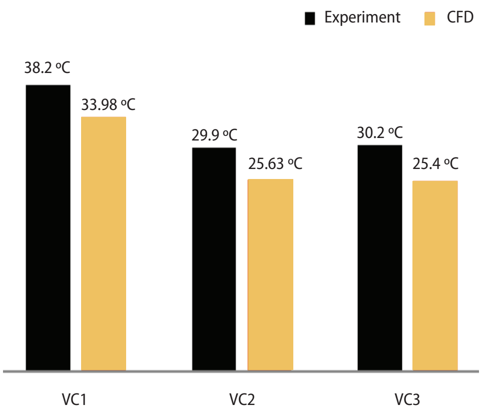

Figure 3. Comparison of the airside temperature

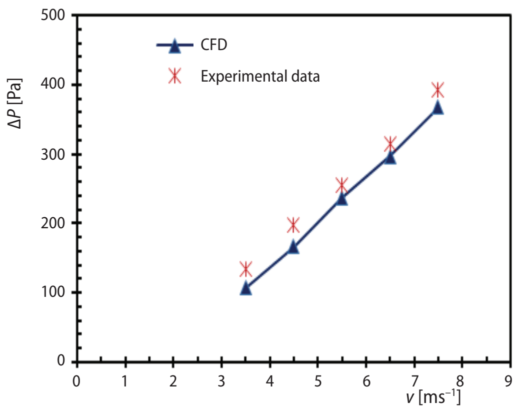

Figure 4. Comparison of air side pressure drop difference

Air side heat transfer coefficient and goodness factor

The average surface convective heat transfer coefficient of air side for various frontal air velocities ranging from 3.5-7.5 m/s based on the experimental and CFD analysis is presented in fig. 5. It has been observed that the convective heat transfer coefficient increases with respect to increase in Reynolds number which is due to the continuous disturbance of the thermal boundary layer with presence the of louvers. It has also been noticed that the experimentally determined values are higher than the results obtained from the CFD analysis at all air velocities. There is a deviation of 15% to 20% at the air velocities higher than 4 m/s. However, the discrepancies are high at the lower velocities (less than 4 m/s) owing to the uncertainties involved in the experimental measurements as well as the numerical error in the turbulence model employed.

Further, it is necessary to consider heat transfer and pressure drop simultaneously during the design and selection of any compact heat exchanger. In this regard, the heat exchanger designers normally evaluate the volume goodness factor *($j/f^{1/3}$)*, which can be used to predict the overall thermo-hydraulic performance in several practical applications where the entire heat exchanger volume should be taken into account in addition to the pressure drop. Figure 6 illustrates the variation of the volume goodness factor with respect to the Reynolds number. It has been seen from the figure that the goodness factor is higher at a lower Reynolds number, and tends to decreases with increase in the Reynolds number due to high pressure drop involved at higher velocities in the test radiator. It is found that there is a drop in goodness factor of 22.7% as Reynolds number increased from 231 to 495. It has also been noted that the experimental results have good agreement with the CFD results and at higher velocities the percentage deviation is negligible.

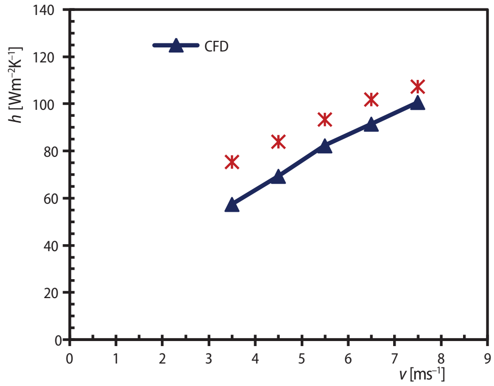

Figure 5. Comparison of heat transfer coefficient (*h*)

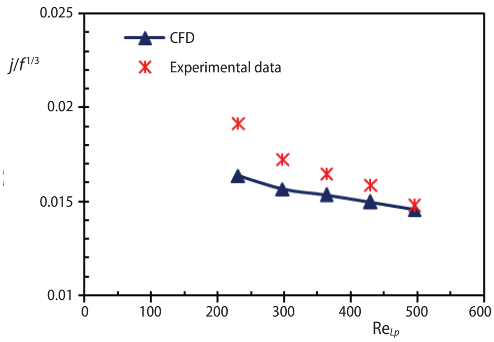

Figure 6. Comparison of volume goodness factor

Comparison of experimental (f) and (j) factors with the existing correlations

The experimentally determined (*f*) and (*j*) factors from the present investigation were compared with the values obtained from the correlations available in the literature that were developed for the louvered fin and flat tube compact heat exchangers. Figure 7 compares the variation of the experimental Fanning friction (*f*) factor with the four different existing correlations. It has been seen from figure that the values of (*f*) obtained from Dong *et al* [10], Chang *et al* [12], and Davenport [9] are much lower than the experimental values, whereas the predicted results from Li and Wang [13] are marginally higher (8-17%,) than the experimental values. This is mainly due to number of tube rows (single row), arrangement of louver regions and higher range of Reynolds number based on the louver pitch, when compared with the present investigation. It has also been noticed that the results of Davenport [9], Chang *et al* [12], and Dong *et al* [10] under predict the experimental data in the range of 66-72%, 61-66% and 41-55%, respectively. The possible reasons for the larger deviation in the (*f*) value could be due to considerable variations in the influencing geometrical parameters such as flow length, fin pitch, louver pitch, longitudinal and transverse tube pitch, louver angle, fin height, and louver height from the louvered fin considered in the present study. Hence, it is construed from the previous comparison that the correlation developed for a particular geometry may not be generalized for all such similar heat exchangers having much variation in fin and louver geometry.

Figure 8 shows the variation of the Colburn (*j*) factor obtained experimentally and the predicted values from five different existing correlations [9-11, 13, 14]. Though, the value of convective heat transfer coefficient increases with respect to increase in Reynolds number, the value of (*j*) tends to decrease with respect to increase in Reynolds number as shown in fig. 8. This is due to predominant effect of increase in frontal air velocity on the Stanton number that lowers the Colburn (*j*) factor. It has clearly been understood from the figure that the correlation by Dong *et al.* [10] predicts the experimental data within the acceptable limits in the lower Reynolds number and it over predicts the experimental data by 39.2% at $\mathrm{Re}_{Lp} = 495$. Further, the results of Sunden and Svantessan [14], Li and Wang [13], and Chang and Wang [11] over predict the experimental results and the over prediction is mainly due to the difference in the number of louver region and flow length from the present configuration. The previously mentioned parameters, number of louver region and flow length, play a vital role on the air side heat transfer coefficient that leads to considerable increase in (*j*) factor for a particular frontal air velocity. However, the results of the Davenport [9] highly under predict the experimental results due to larger fin pitch, lower louver angle and variation in louver fin geometry (triangular channel) compared to that of the louvered fin configuration used in the present investigation.

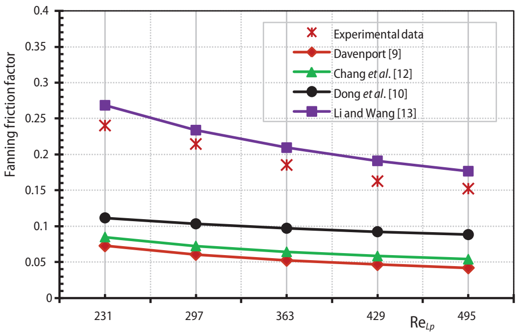

Figure 7. Comparison of the experimental friction (*j*) factor with the existing correlations

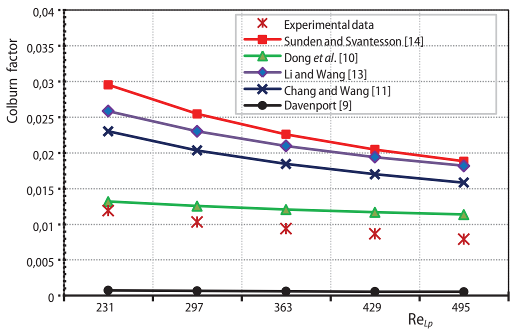

Figure 8. Comparison of experimental Colburn (*j*) factor with the existing correlations

Conclusions

The experimental investigation was carried out on the louvered fin and flat tube heat exchanger, and the experimental results were compared with the CFD results. Further, the experimental (*f*) and (*j*) factors were compared with the predicted values from the correlations available in the literature. The following conclusions were made from the present investigation.

- The presence of louvers appreciably increases the pressure drop from 50 Pa to 400 Pa for the variation in the inlet air velocity from 3.5 to 7.5 m/s and the convective heat transfer coefficient on the airside increases due to the continuous disturbance of the thermal boundary layer. However, there is a drop in volume goodness factor of 22.7% as Reynolds number increased from 231 to 495, owing to the predominant effect of increase in the frontal air velocity on Stanton number.
- The (*f*) and (*j*) factors from the CFD analysis are in good agreement with the experimental results and the variation of these values confirms the general characteristics curve of a typical compact heat exchanger, in which (*f*) and (*j*) factors decrease with increase in Reynolds number.
- The predicted (*f*) and (*j*) factors for the present configuration using different correlations available in the literature showed a considerable deviation from the experimental results.

Hence, it is construed that the existing correlations could not be used to predict (*f*) and (*j*) factors for all kinds of fin and tube configurations.

- The thermal analysis of compact heat exchangers using the recent features of CFD software will certainly make enormous techno-economic beneficial to the heat exchanger industries with appreciable saving in time towards the optimal design of the compact heat exchanger.

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Nomenclature

A_f – frontal area, [m^2]
$A_{\min}$ – minimum free flow area, [m^2]
A_s – surface area, [m^2]
D_h – hydraulic diameter, [m]
f – Fanning friction factor, [–]
G – mass flux or mass velocity, [kgm^{-2}s^{-1}]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
j – Colburn factor, [–]
L – length, [m]
L_p – louver pitch, [m]
ΔP – air side pressure drop, [Pa]
Pr – Prandtl number, [–]
Re – Reynolds number, [–]
St – Stanton number, [–]
T_i – air inlet temperature, [°C]
T_0 – air outlet temperature, [°C]
T_w – tube wall temperature, [°C]
v – frontal air velocity, [ms^{-1}]

Greek symbol

μ – dynamic viscosity, [Nsm^{-2}]
ρ – density, [kgm^{-3}]

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