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

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Abstract

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.

Keywords: Louvered fin, Pressure drop, Goodness factor, Compact heat exchangers, CFD.

1. 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 above 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 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 above are summarized as follows. Beavais [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 nonstandard 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 above, 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 [15] and Khaled [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 analyzed. 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 above 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 above critical issues, the researchers have developed correlations for ‘f’ and ‘j’ factors based on their experimental and computational fluid dynamics 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.
2. Experimental setup

Fig. 1 shows the schematic arrangement of the experimental setup 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, 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 louveres are trapezoidal in shape and each side of the fin has 27 louveres. The details of the louvered fin and flat tube geometry are shown in Table 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 above 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 RTDs (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

<table>
<thead>
<tr>
<th>Geometric Parameters</th>
<th>Flat tube</th>
<th>Louvered fin</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>822 mm</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Width</td>
<td>23.45 mm</td>
<td>0.1 mm</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>0.32 mm</td>
<td>7.6 mm</td>
</tr>
<tr>
<td>Diameter</td>
<td>2 mm</td>
<td>1.2 mm</td>
</tr>
<tr>
<td>Transverse tube pitch</td>
<td>7.6 mm</td>
<td>0.284 mm</td>
</tr>
<tr>
<td>Longitudinal tube pitch</td>
<td>28 mm</td>
<td>26°</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>148</td>
<td></td>
</tr>
<tr>
<td>Number of longitudinal tube rows</td>
<td>2</td>
<td>Flow passage hydraulic diameter</td>
</tr>
</tbody>
</table>

The radiator to be tested is fixed at the inlet of the wind tunnel in the experimental setup, 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 $3.5 \text{ m s}^{-1}$ to $7.5 \text{ m s}^{-1}$ with a step size of $1 \text{ m/s}$. 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.

![Fig.1 Schematic arrangement of the experimental setup](image)

**3. 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 below. The hydraulic diameter of the louvered fin is calculated from,

$$D_s = \frac{4L_{min}}{A_s}$$  \hspace{1cm} (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,

$$Re_{lp} = \frac{GL_p}{\mu},$$
where, \( L_p \) is louver pitch. The dimensionless Reynolds Number based on hydraulic diameter is determined by using the following equations:

\[
Re_{h} = \frac{GD_h}{\mu} \tag{3}
\]

\[
G = \frac{PA_f}{A_{min}} \tag{4}
\]

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,

\[
f = \left( \frac{\Delta P}{v^2} \right) \left( \frac{D_h}{\rho \alpha_v} \right) \tag{5}\]

where, \( \Delta P, \rho_a \) and \( v \) denote air-side pressure drop, density of air and inlet air velocity respectively. The dimensionless Colburn \( j \) factor is evaluated using the following equation

\[
j = St \times p_r^{\frac{3}{2}} = \left( \frac{D_h}{4L} \right) \times ln \left( \frac{T_i-T_o}{T_o-T_w} \right) \times p_r^{\frac{3}{2}} \tag{6}\]

where, \( St, Pr, T_i, T_o \) and \( T_w \) represent the Stanton number, Prandtl number, the inlet, outlet air temperatures and the tube wall temperature respectively.

**Table 2** Results of uncertainty analysis

<table>
<thead>
<tr>
<th>S. No</th>
<th>Measured data</th>
<th>S. No</th>
<th>Derived data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Temperature ( \pm 0.15 ) °C</td>
<td>1.</td>
<td>Hydraulic diameter ( \pm 1.67 ) %</td>
</tr>
<tr>
<td>2.</td>
<td>Air velocity ( \pm 0.14 ) %</td>
<td>2.</td>
<td>Mass velocity ( \pm 1.6 ) %</td>
</tr>
<tr>
<td>3.</td>
<td>Air side pressure drop ( \pm 0.09 ) %</td>
<td>3.</td>
<td>Heat transfer coefficient ( \pm 3.6 ) %</td>
</tr>
<tr>
<td>4.</td>
<td>Water mass flow rate ( \pm 0.5 ) %</td>
<td>4.</td>
<td>Fanning friction factor ( f ) ( \pm 2 ) %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5.</td>
<td>Colburn Factor ( j ) ( \pm 3.2 ) %</td>
</tr>
</tbody>
</table>

**4. Results and Discussion**

In order to ensure the accuracy and reliability of the experimental setup, 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.

**4.1 Airside temperature and pressure difference**

Fig. 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]. Fig. 3 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 Table 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 (VCs) 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. Fig. 4 compares the pressure drop across the heat exchanger for various inlet air velocities ranging from 3.5 to 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

<table>
<thead>
<tr>
<th>Validation Cases</th>
<th>Air velocity ($m,s^{-1}$)</th>
<th>Inlet air temperature ($K$)</th>
<th>Water flow rate ($kg,s^{-1}$)</th>
<th>Inlet water temperature ($K$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>VC1</td>
<td>2.5</td>
<td>310</td>
<td>0.004215</td>
<td>363</td>
</tr>
<tr>
<td>VC2</td>
<td>5.6</td>
<td>308</td>
<td>0.006181</td>
<td>363</td>
</tr>
<tr>
<td>VC3</td>
<td>7.5</td>
<td>302</td>
<td>0.007586</td>
<td>363</td>
</tr>
</tbody>
</table>

Fig. 3 Comparison of the airside temperature difference

Fig. 4 Comparison of airside pressure drop difference

### 4.2 Airside 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 to 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 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. Fig. 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.

4.3 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. Fig. 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 above 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.
Fig. 7 Comparison of the experimental friction ‘j’ factor with the existing correlations

Fig. 8 Comparison of experimental Colburn ‘j’ factor with the existing correlations

Fig. 8 shows the variation of the Colburn j factor obtained experimentally and the predicted values from five different existing correlations [9, 10, 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 ReLp = 495. Further, the results of Sunden and Svantesson [13], Li and Wang [12] 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 above 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.

6. 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.

Acknowledgement

The authors are grateful to M/s. Halgona Radiators Limited, Bangalore, India for providing the wind tunnel test facility.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$A_{min}$</td>
<td>minimum free flow area [$m^2$]</td>
</tr>
<tr>
<td>$A_s$</td>
<td>surface area [$m^2$]</td>
</tr>
<tr>
<td>$D_h$</td>
<td>hydraulic diameter [$m$]</td>
</tr>
<tr>
<td>$f$</td>
<td>Fanning friction factor [-]</td>
</tr>
<tr>
<td>$G$</td>
<td>mass flux or mass velocity ($kg \ m^{-2} \ s^{-1}$)</td>
</tr>
<tr>
<td>$T_w$</td>
<td>tube wall temperature ($^\circ C$)</td>
</tr>
<tr>
<td>$v$</td>
<td>frontal air velocity ($m \ s^{-1}$)</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>air side pressure drop ($Pa$)</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number [-]</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number [-]</td>
</tr>
</tbody>
</table>
\[ j \text{ Colburn factor } [-] \]
\[ L \text{ length } (m) \]
\[ T_i \text{ air inlet temperature } (^\circ C) \]
\[ T_o \text{ air outlet temperature } (^\circ C) \]
\[ L_p \text{ louver pitch } (m) \]
\[ St \text{ Stanton number } [-] \]

**Greek symbol**
\[ \rho \text{ density } (kg m^{-3}) \]
\[ \mu \text{ dynamic viscosity } (N s m^{-2}) \]

**References**


