A Numerical Investigation on the Performance of Finned Tube Heat Exchangers Having Alternate Arrangements of Flat and Circular Tubes

To cite this article: Mohd Zeeshan et al 2019 J. Phys.: Conf. Ser. 1172 012057

View the article online for updates and enhancements.
A Numerical Investigation on the Performance of Finned Tube Heat Exchangers Having Alternate Arrangements of Flat and Circular Tubes

Mohd Zeeshan1, a), Virendra Kumar1, Sujit Nath1 and Dipankar Bhanja1

1Department of Mechanical Engineering, National Institute of Technology Silchar, Assam, India, Pin: 788010
a) Corresponding author: nits.zeeshan@gmail.com

Abstract. The thermal hydraulic behaviour of finned tube heat exchangers (FTHEs) has been studied numerically by alternating the tube arrangements. Area goodness factor ($j/f$) and heat transfer rate per unit fan power consumption ($Q/P_f$) have been incorporated to evaluate the performance of the FTHEs. Performance ranking of the different configurations of FTHEs has also been obtained by the MOORA method. Heat transfer coefficient of air ($h$), $j/f$ factor and $Q/P_f$ factor are taken as beneficial attributes whereas, pressure drop penalty is assumed to be a non-beneficial attribute. All attributes own same importance factor. It is found in the investigations that finned tube heat exchanger having the flat tube in first row, provides increased $h$ and reduces pressure drop penalty compared to the baseline case.

1. Introduction

FTHEs are widely used in various applications, e.g. ventilating and air-conditioning systems, refrigeration, and car radiators etc. as they own high performance and low space requirement. The shape of the tube and the spatial arrangements of the tube bundles significantly affect the performance of the FTHEs. The aerodynamic shape of the flat tube provides the better performance as it provides less pressure drop penalty, increases the separation delay and reducing the wake zone behind the tube that result an enhancement in the thermal performance.

A number of literature is available which assures the ability of the flat tubes in performance augmentation of the FTHEs. Experimental as well as numerical studies have been done to evaluate the performance enhancement by the flat tubes with various spatial arrangements. Fiebig et al. [1] experimentally found that flat tubes can enhance the heat transfer rate up to 100%. At the same time Jacobi and Shah [2] reviewed a number of investigation on the heat transfer augmentation which also confirms the flat tube contribution in the performance enhancement of FTHEs. Yoo et al. [3] experimentally investigated that flat tube heat exchanger with vortex generators (VGs) has better heat transfer rate compared to the round tube configurations. Junqi et al. [4], Chu et al. [5], Cuevas et al. [6], Huisseune et al. [7], and Du et al. [8] incorporated enhanced fin surfaces and VGs etc. with flat tubes to augment the thermal hydraulic performance of the FTHEs. Zeeshan et al. [11] numerically investigated that if the flatness increases performance is also increased, due to separation delay and less friction factor. Kumar and Jayavel [12] found that placing oval tubes in the first row increases the performance compared to placing a circular one in the first row of the tube bundle.

The preceding literature review concluded that the better performance of FTHEs can be achieved by incorporating tubes having aerodynamic shapes (i.e. flat and oval tubes) in place of round tubes. It
is evident from the literature survey that effect of combinations of flat and circular tube on the performance of FTHE is lagging. In the present work two combinations of the flat and circular tubes are studied and explored the thermal hydraulic behaviour of FTHEs. The $j/f$ factor (i.e. Colburn factor/friction factor) and $Q/Pf$ factor are taken as performance evaluation criteria. Moreover, MOORA method [11, 13-14] is also applied to have the performance ranking of FTHEs.

2. Methodology
In the current numerical investigation, the evaporator geometry has been taken from Joardar and Jacobi [9] with the same geometrical and thermo-physical parameters. The geometrical parameters of the flat tube have been taken from Zeeshan et al. [11] on the basis of providing the better thermal hydraulic performance. The geometrical description of the considered circular and flat tube has been shown in Fig.1. The flat tube and the circular tube have equal perimeter. The transverse tube pitch ($P_s$) and longitudinal tube pitch ($P_l$), are of same dimensions ($P_s=P_l=25.4$ mm). Two different combinations of circular and flat tube has been introduced to study the airside thermal hydraulic behaviour of FTHEs. The combinations of circular and flat tubes are depicted in Fig. 2.

3. Numerical Approach
The features of symmetry and periodicity in the geometry of evaporator enables to model the only (1/8)th part of fin plate with tubes to predict the thermal hydraulic behaviour of FTHEs. The computational model of the considered evaporator segment has been modelled in Design Modeler ANSYS 14.0. Workbench ANSYS 14.0 (ICEM CFD) is used to generate the mesh with structured and non-structured grid systems. The governing equations (mass, momentum and energy conservation) along with the boundary conditions has been solved by FLUENT ANSYS 14.0 [11] in the computational domain. SIMPLE algorithm is applied to couple the velocity and pressure. The discretization of the convective terms has been done with the second order upwind scheme. The considered computation domain is prolonged by 5H and 30H at the inlet and outlet respectively.
11 and 13-14] for the uniformity of the inlet velocity and a recirculation-free flow. A 3-D model of the computational domain with boundary conditions has been depicted in Fig. 3. Furthermore explanation of governing equations and boundary condition can be seen in He et al. [10] and Zeeshan et al. [11].

![3-D model of the considered computational domain with boundary conditions](image)

**FIGURE 3:** 3-D model of the considered computational domain with boundary conditions

### 4. Validation of Numerical Results

The simulation parameters are: Reynolds number $Re$: 400 to 900; Inlet temperature: 310.6 K; Tube wall temperature: 291.77 K. Three grid systems having 850950, 1179341 and 1681899 elements are been investigated for the baseline case i.e., FTHE with circular tubes. The predicted $h$ (heat transfer coefficient of air) is depicted in Fig. 4 for the three grid systems. 1179341 elements is finally chosen for the numerical analysis. In the current study the numerical results for $h$ and $\Delta p$ are compared with existed experimental results given by Joardar and Jacobi [9] to validate the numerical model. There is a little difference between the results i.e., for airside heat transfer coefficient ($h$) is 12.01% (at $Re = 900$) to $-12.18\%$ (at $Re = 900$) and that for pressure drop ($\Delta p$) is 17.43% (at $Re = 400$) to $-15.04\%$ (at $Re = 900$) as shown in Fig.5. This difference exists mainly due to the closeness of the numerical solution to the ideal conditions whereas in experimentations leakage and contact resistance are inevitable. The grid independency together with fair agreement of the numerical predictions with the experimental results assures the trustworthiness of the numerical model for the further predictions of thermal hydraulic behaviour of FTHEs.

![Variation of $h$ with $Re$ for different grid systems](image)

**FIGURE 4:** Variation of $h$ with $Re$ for different grid systems

![Variation of $h$ and $\Delta p$ with $Re$](image)

**FIGURE 5:** Variation of $h$ and $\Delta p$ with $Re$
5. Results and Discussion
In the current numerical analysis, airside thermal hydraulic performances of the FTHEs, having alternating tube arrangements have been evaluated and matched with the baseline case i.e. finned circular tube heat exchanger. In the first alternating arrangement (1F-1C), a flat tube is placed at the first row and in second alternating arrangement (1C-1F), a circular tube is placed at the first row. Figure 6 shows the distribution of the temperature and velocity at \( Re = 900 \) over the considered fluid domain. It can be appreciated that the wake zone behind the circular tube is more in comparison to the flat tubes which causes the reduction in heat transfer rate. In contrast, the wake zone behind the flat tube is lesser than the circular tube as the separation point is closer to the rear end of the tube. Smaller wake region and lower form drag are achieved by implementation of the flat tubes which significantly reduces the pressure drop penalty and increases the heat transfer rate. Figure 7 represents the variation of \( h \) and \( \Delta p \) over the considered \( Re \) range 400-900. For the combination 1F-1C having flat tube in the first row and circular tubes alternatively as shown in Fig. 6, the incoming flow faced less blockage as the frontal exposed area compared to circular one is less, which reduces the pressure drop penalty.

Figure 6: Distribution of (a) temperature and (b) velocity in the considered domain at \( Re = 900 \)

Figure 7: Variation of the predicted \( h \) and \( \Delta p \) with \( Re \)

Moreover, the separation delay occurs in the flat tube case which significantly reduces the wake zone behind the flat tubes results increase in \( h \). In second arrangement (1C-1F), as the circular tube is placed at the first row, separation takes place earlier compared to the flat tubes which enlarges the wake zone behind the tubes. This results in the larger \( \Delta p \) and the smaller \( h \) as compared to the flat tubes. It can be concluded from the variation of \( h \) and \( \Delta p \) that both the alternating arrangements provide a higher \( h \) with low \( \Delta p \) compared to the baseline case. Variation of the \( j/f \) factor over the considered \( Re \) range is shown in the Fig. 7. It can be seen that both the arrangements significantly increase the performance of FTHE. The alternating arrangement 1F-1C, provides higher values of the \( j/f \) factor which clearly shows the higher performance with less frontal area requirement of the FTHE.
compared to the other considered cases. The finned tube heat exchanger having the tube arrangement 1F-1C, provides 4.09% and 1.88% increased heat transfer coefficient ($h$) at $Re=400$ and $Re=900$, respectively and reduces pressure drop penalty ($\Delta P$) by 21.48% at $Re=400$ and 18.82% at $Re=900$, compared to the baseline case.

| $Re$  | Assessment values ($\gamma$) | 1F-1C | 1C-1F | Baseline Case |
|-------|-------------------------------|-------|-------|---------------|
| 400   | 0.2004                        | 0.1693| 0.1302|               |
| 500   | 0.2040                        | 0.1699| 0.1259|               |
| 600   | 0.2137                        | 0.1547| 0.1259|               |
| 700   | 0.2011                        | 0.1577| 0.1411|               |
| 800   | 0.2034                        | 0.1551| 0.1414|               |
| 900   | 0.2010                        | 0.1565| 0.1424|               |

It is the point worthy to note that the finned tube bundles having alternating arrangements of circular and flat tubes provide a noticeable reduction in the pressure drop whereas the heat transfer rate is increased in a little amount compared to the baseline case. Both the pressure drop reduction and heat transfer augmentation is more when the flat tube is placed in the first row (1F-1C) compared to 1C-1F. It can be established from Fig.8 that the alternating arrangement of tube bundle 1F-1C shows higher $Q/P_f$ value which signifies the higher heat transfer with less expense of fan power. The assessment values obtained by the MOORA method have been shown in Table 1. The performance ranking obtained by MOORA also assures the better thermal hydraulic performance of 1F-1C configuration, which indicates that the 1F-1C provides higher heat transfer rate on the expense of low fan power and pressure drop penalty and requires less frontal area.

6. Conclusions
In the present numerical study, two alternating arrangements of the flat and circular tubes are introduced namely 1F-1C (having the flat tube at first row) and 1C-1F (having the circular tube at first row). The thermal hydraulic performance of enhanced FTHEs are evaluated and compared with baseline case (where all tubes are circular tubes). The main conclusions of the present study can be summarized as follows:

- Introducing an alternate arrangement of flat and circular tube in the finned circular tube bundles affects the performance significantly as $\Delta P$ is reduced due to weaker wake zone behind the tubes. A little increment in the $h$ is also achieved due to separation delay in the enhanced cases.
- The alternating arrangement 1F-1C provides the higher $h$ and lesser $\Delta P$ compared to the other considered cases. A significant reduction of pressure drop penalty ($\Delta P$) by 21.48% at $Re=400$ and 18.82% at $Re=900$ has been noticed. Whereas, $h$ is increased by 4.09% at $Re=400$ and 1.88% at $Re=900$.
- On the basis of the $j/f$ factor and $Q/P_f$ factor, the FTHE having alternating arrangement 1F-1C, provides the better thermal hydraulic performance compared to other considered cases. It implies that 1F-1C requires less frontal area i.e., compactness and provide higher heat transfer rate per unit fan power consumption compared to the other considered cases.
- Moreover, the assessment values of the MOORA method also assures that the tubes arrangement 1F-1C provides the better thermal hydraulic performance and can be selected as an optimal configuration.

7. References
[1] M. Fiebig, A Valencia, N.K.Mitra, Local heat transfer and flow losses in fin-and-tube heat exchangers with vortex generators: a comparison of round and flat tubes, Exp Therm Fluid Sci, 8 (1994) 35–45.
[2] A.M Jacobi, R.K.Shah, Heat Transfer Surface Enhancement through the Use of Longitudinal Vortices: A Review of Recent Progress, Exp. Therm. Fl. Sci., 11(1995), 295-309.

[3] S. Y.Yoo, D.S.Park, M.H Chung., Heat Transfer Enhancement for Fin-Tube Heat Exchanger Using Vortex Generators, KSME Int. J., 16(2002)109-115.

[4] D. Junqi, C.Jiangping,., C.Zhiju, Z. Yimin, Z. Wenfeng, Heat transfer and pressure drop correlations for the wavy fin and flat tube heat exchangers, App. Therm. Eng., 27(2007),2066–2073.

[5] P. Chu,., Y. L. He, Lei, L T Tian, and R. Li, Three-dimensional numerical study on fin-and-oval-tube heat exchanger with longitudinal vortex generators, App. Therm. Eng., 29 (2009) 859–876.

[6] C. Cuevas, D. Makaire, L. Dardenne, P. Ngendakumana, Thermo-hydraulic characterization of a louvered fin and flat tube heat exchanger, Exp. Therm. Fl. Sci., 35 (2001) 154–164.

[7] H. Huisseune, C.T. Joen, P.D. Jaeger, B. Ameel, S.D. Schamphleire, M.D. Paepe, Influence of the louver and delta winglet geometry on the thermal hydraulic performance of a compound heat exchanger Int. J. of H. M. Trans, 57 (2013) 58–72.

[8] X. Du, L. Feng, L. Li, L. Yang, Y. Yang, Heat transfer enhancement of wavy finned flat tube by punched longitudinal vortex generators, Int. J. H. M. Trans., 75 (2014) 368–380.

[9] M. Zeeshan, S. Nath, D. Bhanja, Numerical study to predict optimal configuration of fin and tube compact heat exchanger with various tube shapes and spatial arrangements, En. Conv. Manag., 148 (2017) 737-752.

[10] A. Joardar, and A.M. Jacobi, Heat transfer enhancement by winglet-type vortex generator arrays in compact plain-fin-and-tube heat exchangers, Int. J. ref., 31 (2008). 87–97.

[11] Y.L. He, P. Chu, W.Q. Tao, Y.W. Zhang, T. Xie, Analysis of heat transfer and pressure drop for fin-and-tube heat exchangers with rectangular winglet-type vortex generators, App. Therm. Engg., 61 (2013) 770-783.

[12] R.D. Kumar, S.Jayavel, Air side performance of finned-tube heat exchanger with combination of circular and elliptical tubes, App. Therm. Eng., 119 (2017) 360–372.

[13] M. Zeeshan, S. Nath, D. Bhanja, A. Das, Numerical investigation for the optimal placements of rectangular vortex generators for improved thermal performance of fin-and-tube heat exchangers, App. Therm. Eng., 136 (2018) 589–601.

[14] M. Zeeshan, S. A. Hazarika, S. Nath, and D. Bhanja, Numerical investigation on the performance of fin and tube heat exchangers using rectangular vortex generators, AIP Conference Proceedings (2017) 1859, 020011.

Acknowledgments
Authors thank TEQIP-III (N.I.T. Silchar, Assam) for their financial support. We also thankful to the CFD lab, M.E. Department (N.I.T. Silchar, Assam), for providing the computational facility in carrying out the present numerical investigation.