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Three-dimensional performance analysis of wavy fin tube heat exchangers in laminar flow regime

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ABSTRACT

The current study is focused on two fin configurations, the wavy-fin in-line and the wavy-fin staggered. These two fin configurations are numerically investigated in both staggered and in-lined tube layouts. Heat transfer and pressure drop characteristics of the heat exchanger are investigated for Reynolds numbers ranging from 200 to 2000. Model validation is carried out by comparing the simulated case friction factor (f) and Colburn factor (j) with the experimental data of Bhuiyan et al. [2013]. In this study, the effect of geometrical parameters such as fin pitch, longitudinal pitch and transverse pitch of tube spacing are studied. Results are presented in the form of friction factor (f) and Colburn factor (j). This study reveals that the flow distinction between plain and wavy fin has a profound influence on the heat transfer and flow friction performance of these configurations when compared on the basis of tube layouts. For laminar flow conditions heat transfer and friction factor decrease with the increase of longitudinal and transverse pitches of tube spacing whereas they increase with fin pitches for both in-line and staggered configurations.

Keywords— *Heat transfer, Fluid flow, Laminar flow, Heat exchanger*

1. INTRODUCTION

The fin geometry has become as increasingly important factor in the design of a plate-and-fin heat exchanger. The high performance offset strip, wavy and louver fins provide quite high heat transfer coefficients for gases and two-phase applications. It offers significant advantages like lower gas pressure drop than circular tube designs and the ability to have the fins normal to the gas flow over the full gas flow depth over the traditional fin-and-round tube geometry. Enhanced surface geometries are widely used with liquids for cooling electronic equipment. The typically extended surfaces used for the plate-and-fin heat exchangers are: plain fin, wavy fin, offset strip fin, louvered fin, perforated fin, etc. Based on the tube arrangement, these types of heat exchangers can further be divided into two different groups such as staggered and inclined arrangement. Figure 1 shows some typical finned-tube heat exchanger designs, especially for plain and wavy structure.

Plate fin-and-tube heat exchangers of plain fin pattern are commonly used in the process and HVAC&R (Heating, Ventilating, air conditioning, and refrigeration) industries. The plain plate fin configuration is the most popular fin pattern, owing to its simplicity, durability and versatility in application. The plain fin-and-tube heat exchangers usually consist of mechanically or hydraulically expanded round tubes in a block of parallel continuous fins and, depending on the application, the heat exchangers can be produced with one or more rows. During the past few decades, many efforts have been devoted to heating transfer and friction characteristics of plate fin-and tube heat exchangers. Among the entire extended fin surfaces, plain fin represents the simplest geometry. Though lower heat transfer performance is observed for plain fins as compared to the specially configured fin surfaces, these fin types are still widely used where pressure drop characteristics low are desired. When the fins have periodic corrugations in their geometry in the form of a wave, then it is called a wavy fin. The wavy pattern may be smooth or of a herringbone pattern. These periodic corrugations having a definite angle of corrugation that helps in better mixing of flow, thereby providing higher heat transfer. These corrugations in a wavy fin help in increasing the flow length in a limited space than that of the plain fin. This type of geometry is being widely studied and used these days due to its attractive heat transfer performance as demonstrated in their studies conducted by Nishimura et al. [8] and Wang et al. [9]. The important parameters in the study of the wavy fin are the wavy angle and the wavy height, fin pitch, fin length, fin thickness, longitudinal pitch, transverse pitch, waviness amplitude, colburn factor, friction factor, and pressure drop which will be explored in detail in this study.

2. GOVERNING EQUATIONS

In this section, the laws governing the problem are explained in detail. The governing equations are subjected to the boundary conditions of the problem to formulate a solution. These expressions would then be discretized using the finite volume method to estimate the solution. The present problem involved thermal transport with convective heat transfer. Working fluid used is air with Prandtl number set to 0.736.

Assumptions about the fluid and the analysis are as follows:
1. The fluid is Newtonian, incompressible with constant physical properties.
2. The flow is assumed to be three-dimensional and steady state.
3. Viscous dissipation and viscous work are neglected.
4. No body forces.

Based on the above assumptions, the problem is defined by the laws of mass, momentum and energy which are stated in the following sections. The presented study stretches from the laminar range flow ($4000 \le Re \le 1200$) to the transitional range flow ($1300 \le Re \le 2000$). Equations that govern the problem are that of a laminar model for the flow in the laminar range. While for the solution of the flow in the transitional range turbulence models are used, and the equations that govern the problem in the transitional range are that of the turbulence models.

Three different turbulence models were tested in the present study for the flow in the transitional range namely, k-ε model, RNG k-ε model and Baseline (BSL) k-ω model. The commercial CFD code CFX-5 was used to carry out the computations.

3. LAMINAR MODEL

The flow in the laminar range ($4000 \le Re \le 1200$) was described by the conservation laws for mass (continuity), momentum (Navier-Stokes), and by the energy equation. The basic equations describing the three-dimensional flow are as follows:-

The Continuity Equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

The Momentum Equation:
x- Component:

$$\rho\left(u\frac{\partial}{\partial x}u + v\frac{\partial}{\partial y}u + w\frac{\partial}{\partial z}u \right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)$$

y- Component:

$$\rho\left(u\frac{\partial}{\partial x}v + v\frac{\partial}{\partial y}v + w\frac{\partial}{\partial z}v \right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)$$

z- Component:

$$\rho\left(u\frac{\partial}{\partial x}w + v\frac{\partial}{\partial y}w + w\frac{\partial}{\partial z}w \right) = -\frac{\partial p}{\partial z} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)$$

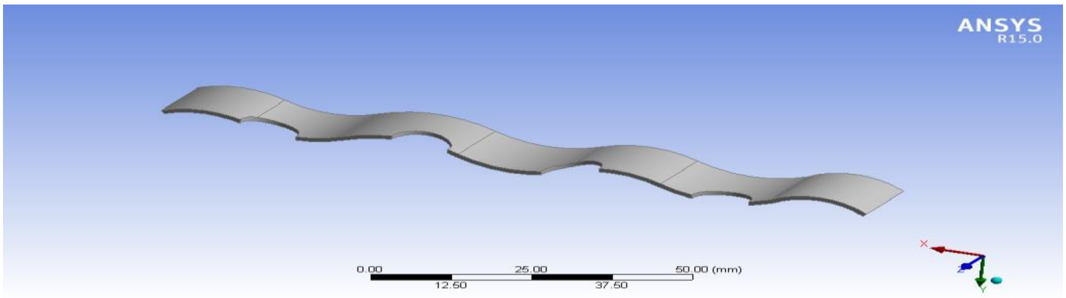
The Energy Equation:

$$\rho C_p\left(u\frac{\partial}{\partial x}T + v\frac{\partial}{\partial y}T + w\frac{\partial}{\partial z}T \right) = \lambda\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right)$$

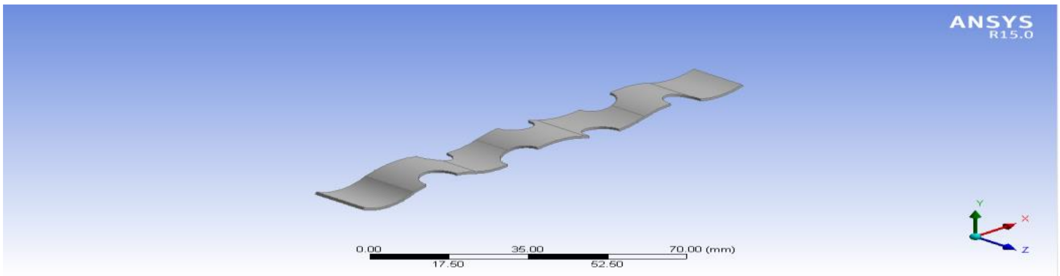
4. THE CFD MODEL

Geometry considered for the present investigation is plain fin in-lined and staggered configuration shown in Figure 1. The z-direction is perpendicular to the paper. Assuming symmetry condition on the mid plane between the two fins, the bottom and the top boundaries simulate the fin and the mid-plane respectively. The detailed geometry (A. A. Bhuiyan et al 2013) of the numerically examined heat exchanger was defined by the following dimensions:

Tube diameter (D) 9.525 mm
Longitudinal tube pitch (L_l) 19.05 mm
Transverse tube pitch (L_t) 25.4 mm
Fin Pitch (Fp) 3.53 mm
The number of tube row (N) 4



(a) Inline wavy- fin configuration



(b) Staggered wavy- fin configuration

Fig. 1: Geometry considered for the present investigation

5. BOUNDARY CONDITIONS

This section describes boundary conditions used in the present study with reference to Figure 1. Assuming symmetry conditions on the mid-plane between two fins, the bottom and top boundaries simulate the fin and the mid-plane respectively. This symmetry of the problem was used to model only one half of the domain for computational purposes.

The boundary conditions used are as follows:

- Between the two fins, symmetry conditions are considered.
- At the inlet uniform flow with constant velocity uin and constant temperature Tin = 25 ºC are assumed.
- At the outlet, stream wise gradient (Neumann boundary conditions) for all the variables are set to zero.
- No-slip boundary condition is used at the fins and the tube surfaces. These surfaces are assumed to be a solid wall with no slip boundary condition and constant wall temperature T_{wall} = 100 ºC.
- The fins and tubes are assumed to be made of aluminium.

The average Nusselt number Nu, defined as:

$$Nu = hH/K$$

The friction factor f and the Colburn factor j are defined as

$$f= (Pin-P)H/1/2 \text{ x } \rho \text{ x } u^2_{in} \text{ x } 4L$$

$$j = Nu/Re_H \text{ x } Pr^{1/3}$$

6. RESULTS AND DISCUSSION

The accuracy of the present simulation study was validated by comparing the results for wavy-fin inline and staggered configuration with the published experimental results by Bhuyian et al. 2013. Once the accuracy of the method used was established, the computations were extended for the cases with wavy-fin in-line configuration and wavy-fin staggered configurations as well as for those cases for which no previous studies have been reported.

The validation of the of the numerical simulation presented in this work was done by comparing the published values for the Colburn factor (j) and the friction factor (f) by Bhuyian et al. 2013 with the numerical results for laminar flow range. Same geometrical parameters are used in the current numerical simulation study for the validation of the analysis here: L_l = 22 mm, L_t = 25.4 mm, F_p = 3.00 mm, F_t = 0.13 mm and D = 9.5 mm.

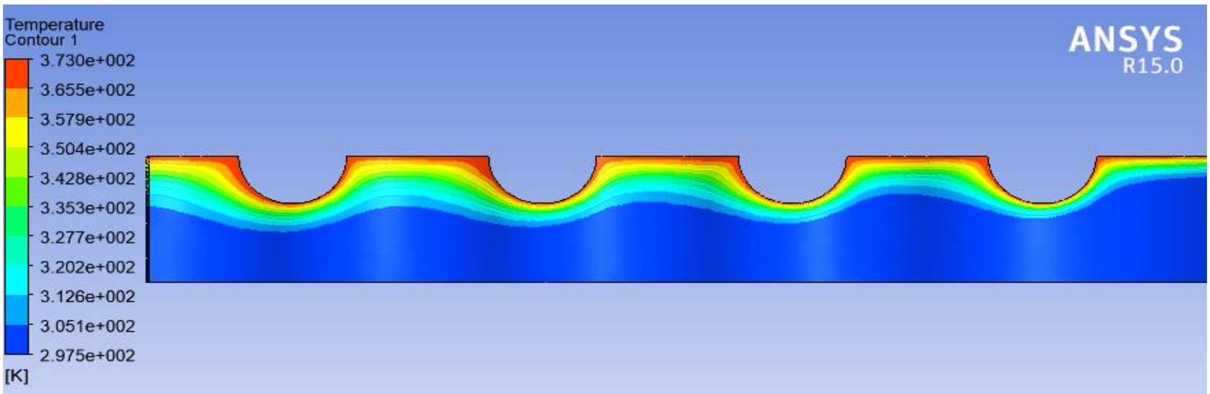


Fig. 2: Temperature contour at Re = 200 for wavy-fin inline configuration

6.1 Effect of longitudinal tube pitch

The effects of the change in longitudinal pitch (L_l) on the heat transfer and the pressure drop characteristics for the four tube row domain of the wavy-fin in-line and wavy-fin staggered configuration was studied by running test cases for each of these fin arrangements. In each case, the longitudinal tube pitch (L_l) is changed keeping all other geometrical parameters constant so that the effects of the longitudinal pitch over the heat exchanger performance can be evaluated.

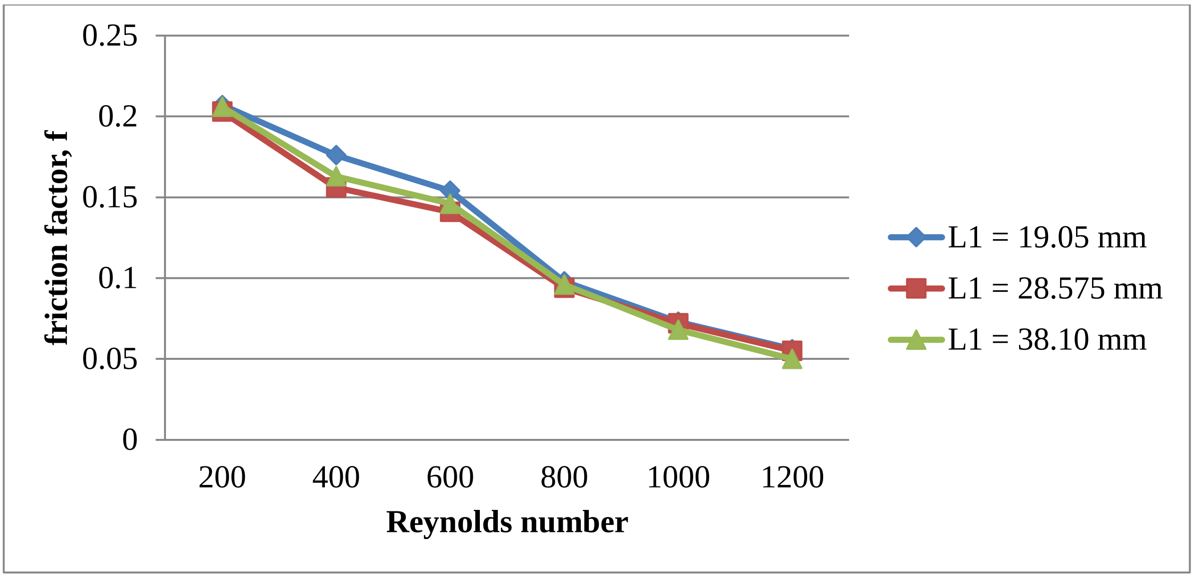


Fig. 3: Effect of varying longitudinal tube pitch on friction factor variation for wavy fin in-line configuration

It can be seen from figure 3 that the friction factor (f) decreases with the increase in the longitudinal tube pitch. This observation again contradicts the expectation that an increase in the fin surface area would increase the flow friction characteristics. This phenomenon, again, can be explained on the fact that lower the longitudinal pitch (Ll), more restricted and dense the airflow is, due to the better flow mixing caused by the obstruction to the flow by the tightly spaced tubes. Increasing longitudinal tube pitch, even though increases the fin surface area, the effect is minor when compared with the less restriction to flow caused by the farther spacing of the tubes for the range of the longitudinal tube pitches tested.

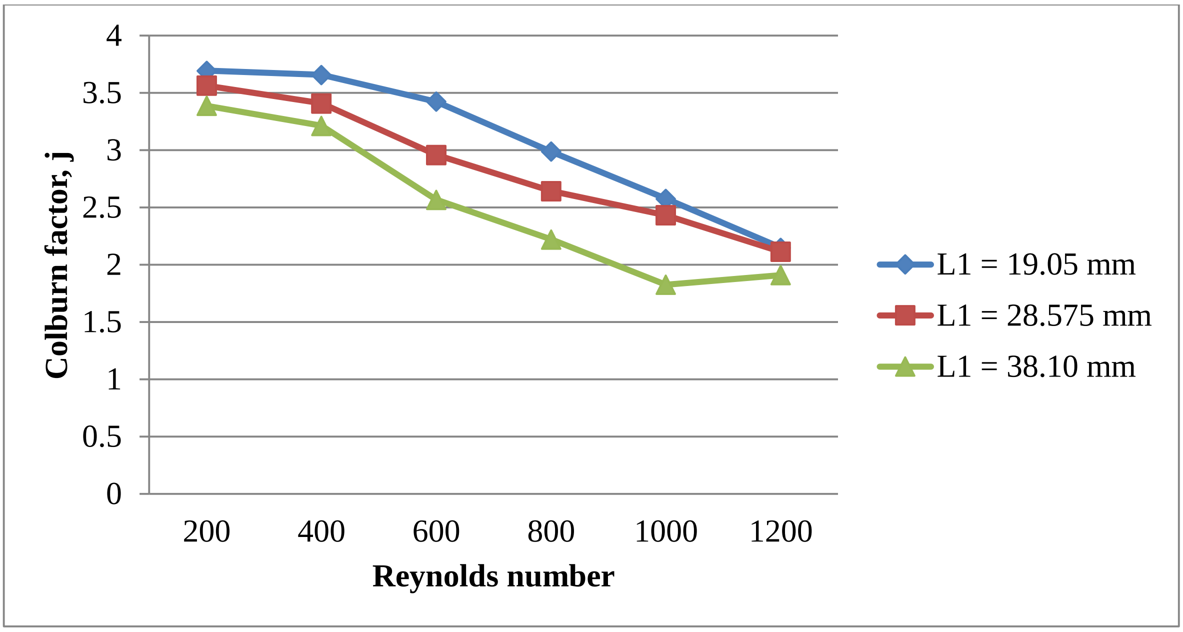


Fig. 4: Effect of varying longitudinal tube pitch on Colburn factor variation for Wavy fin In-line Configuration

Figure 4 shows the variation of the Colburn factor (j) against the Reynolds number (Re) for the three longitudinal pitch test cases. It can be seen from Figure 5.10 that the Colburn factor (j) decreases with the increase in the longitudinal tube pitch (Ll). This observation again confirms the observation stated before that the increase in the longitudinal tube pitch (Ll) makes the flow more simplistic thereby decreasing the heat transfer performance for the fin-configuration.

6.2 Effects of transverse tube pitch

After investigating the effects of the longitudinal pitch on the heat exchanger performance next step in this study was to explore the effects of the transverse pitch. The effects of the change in transverse pitch (Lt) on the heat transfer and the pressure drop characteristics for the four tube row domain of the wavy-fin in-line configuration was studied by running three test cases for each of these fin arrangements

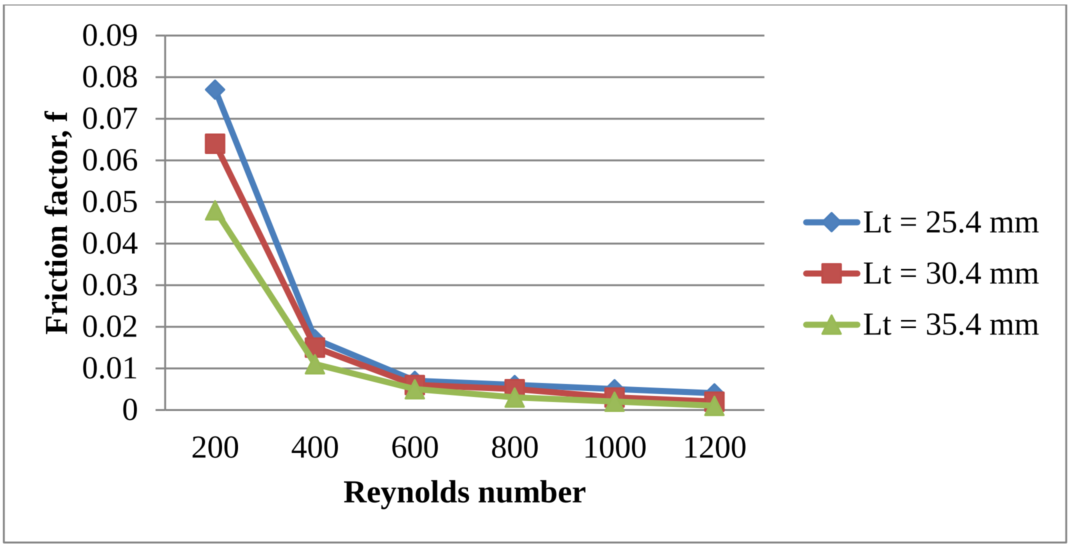


Fig. 5: Effect of varying transverse tube pitch on friction factor variation for Wavy fin In-line Configuration

Figure 5 shows the variation of the friction factor (f) against the Reynolds number (Re) for the three transverse pitches (Lt) test cases. It can be seen from figure 5 that the friction factor (f) decreases with the increase in the transverse tube pitch.

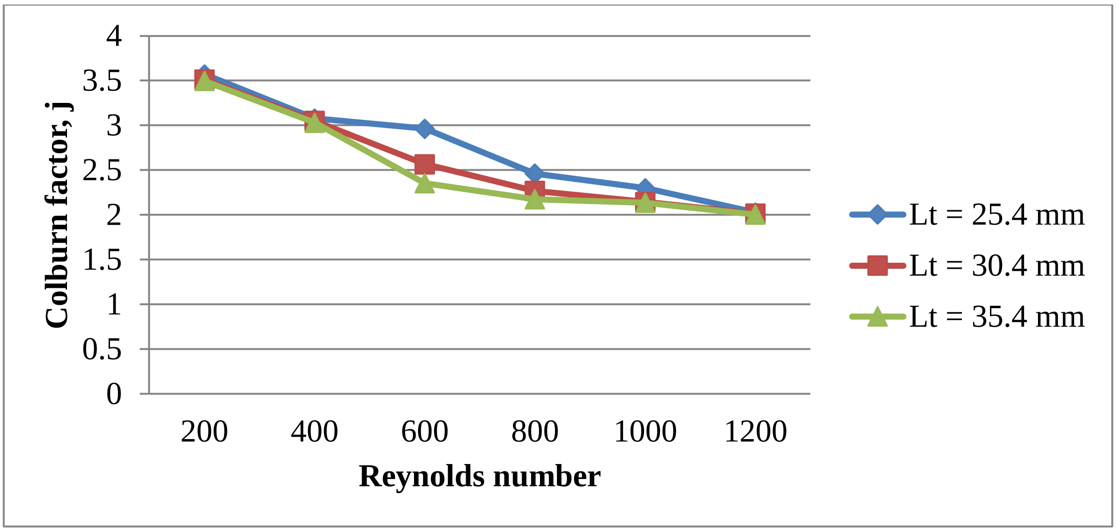


Fig. 6: Effect of varying transverse tube pitch on colburn factor variation for Wavy fin In-line Configuration

It can be seen from figure 6 that the Colburn factor (j) decreases with the increase in the transverse tube pitch (Lt), verifying the observation stated before that the increase in the transverse tube pitch (Lt) makes the flow more simplistic thereby decreasing the heat transfer performance for the fin-configuration.

6.3 Effect of fin pitch

The effects of the change in fin pitch (Fp) on the heat transfer and the pressure drop characteristics for the four tube row domain of the wavy-fin in- line configuration was studied by running three test cases for each of these fin arrangements. In each case, the fin pitch (Fp) is changed keeping all other geometrical parameters constant so that the effects of the longitudinal pitch over the heat exchanger performance can be analyzed.

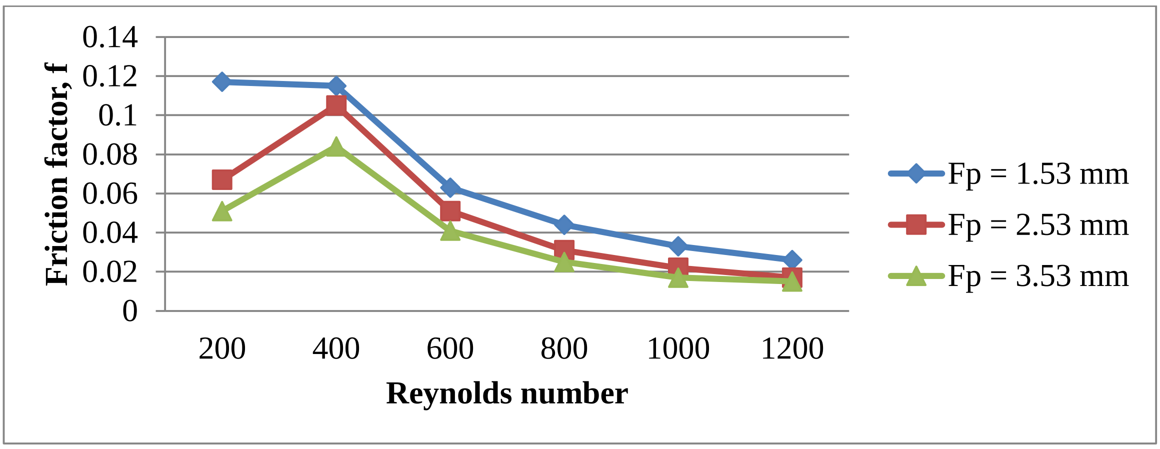


Fig. 7: Effect of varying fin pitch on friction factor variation for Wavy fin In-line Configuration

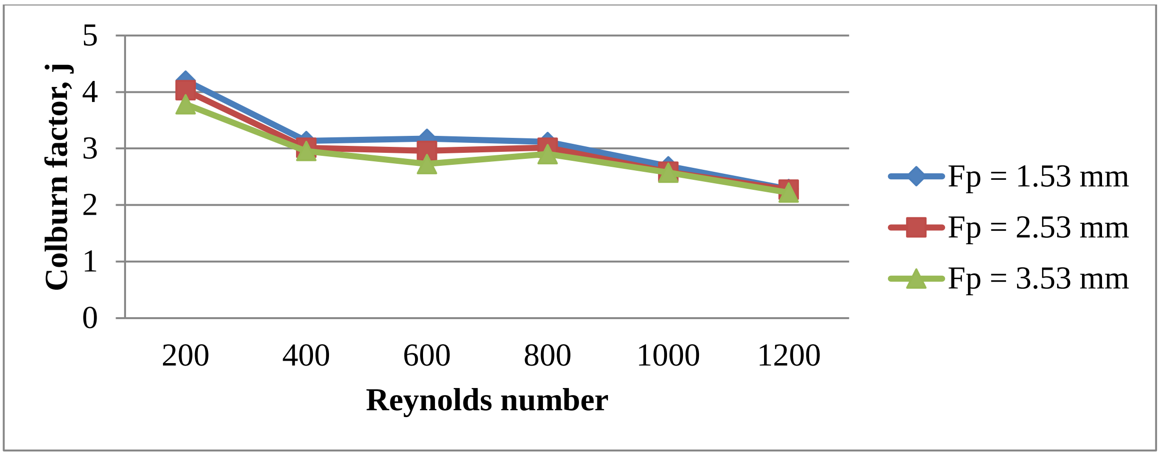


Fig. 8: Effect of varying fin pitch on colburn factor variation for Wavy fin In-line Configuration

It can be seen from figure 8 that the Colburn factor (j) decreases with the decrease in the fin pitch (Fp). This observation can be explained by the fact that keeping the longitudinal tube pitch (Ll) and the transverse tube pitch (Lp) constant, when fin pitch (Fp) is reduced, the flow becomes more streamlined. This flow streamlining caused by a reduction in the fin pitch (Fp), simplifies the flow and hence decreases the turbulence level and the better flow mixing. Also, the available heat transfer area from the tube surfaces reduces from decreased fin pitch (Fp) which affects the Colburn factor (j). As a result, the Colburn factor (j) decreases with the decrease in the fin pitch (Fp).

7. CONCLUSION

Based on the numerical results obtained in this investigation, the following conclusions are made.

- The flow distinction between the plain fin and the wavy fin was found to have a significant effect on the heat transfer and the flow friction characteristics of these two fin configurations.
- The flow structure for the plain-fin configuration is marked by the flow recirculation zones found in the trailing edge of the tubes as the flow passes over the tubes. On the other hand, for the wavy-fin configurations, no such flow recirculation zones were obtained since the flow is guided by the wavy corrugations and it is repeatedly re-oriented due to the wavy structure of the fin.
- The wavy-fin staggered and in-lined configurations were found to be less dependent on the tube layout for their thermal and hydraulic characteristics since a better flow mixing is achieved as the flow passes over the wavy corrugations. Hence wavy-fin was also found to show a much larger heat transfer performance as indicated by the higher Colburn factor (j).
- However, the pressure penalty is also high in comparison with plain-fin counterpart as indicated by the higher friction factor (f).
- The study on the four tube row domains for wavy fin staggered configurations revealed that longitudinal (Ll) and transverse (Lt) tube pitches affect the overall heat exchanger performance. It was demonstrated that the increase in the longitudinal and the transverse pitch for the wavy fin staggered configuration causes a decrease in the heat transfer performance because the flow becomes less compact with the increase in the longitudinal and transverse tube pitch. This decrease in heat transfer performance is also accompanied by the corresponding decrease in the pressure drop characteristics.
- The friction factor (f) decreases with the decrease in the fin pitch. This observation again can be explained on the basis of the flow streamlining and the flow simplification caused by the reduction of fin pitch. This flow simplification reduces the flow mixing which thereby reduces the flow friction resistance. Also reduction in the fin pitch (Fp) reduces the tube surface area reducing the flow friction, which certainly has an augmenting effect in reducing the flow friction factor (f).
- The Colburn factor (j) decreases with the decrease in the fin pitch (Fp). This observation can be explained by the fact that keeping the longitudinal tube pitch (Ll) and the transverse tube pitch (Lp) constant, when fin pitch (Fp) is reduced, the flow becomes more streamlined. This flow streamlining caused by a reduction in the fin pitch (Fp), simplifies the flow and hence decreases the turbulence level and the better flow mixing. Also, the available heat transfer area from the tube surfaces reduces from decreased fin pitch (Fp) which affects the Colburn factor (j). As a result, the Colburn factor (j) decreases with the decrease in the fin pitch (Fp).
- Overall staggered configuration is recommended over in-line configuration

8. REFERENCES

[1] Arafat A. Bhuiyan Three-dimensional performance analysis of plain fin tube heat exchangers in transitional regime, Applied Thermal Engineering, 50, 445-454, 2013.
[2] M. Khoshvaght- Aliabadi, F. Hormozi, A. Zamzamian, Experimental analysis of the thermal–hydraulic performance of copper–water nano-fluid flow in different plate-fin channels Experimental Thermal and Fluid Science 52, pp. 248–258, 2014.
[3] W.S. Kuo, Y.M. Lie, Y.Y. Hsieh, T.F. Lin, Condensation heat transfer and pressure drop of refrigerant R -410A flow in a vertical plate heat exchanger, International Journal of heat and mass transfer, 48, pp.5205-5220, 2005.
[4] S.M. Pesteei , P.M.V. Subbarao, R.S. Agarwal, Experimental study of the effect of winglet location on heat transfer enhancement and pressure drop in fin-tube heat exchangers, Applied Thermal Engineering, 25, pp.1684– 1696, 2005.
[5] Ting Ma, Min Zeng, Yanpeng Ji, Haibin Zhu, Qiuwang Wang, Investigation of a novel bayonet tube high temperature heat exchanger with inner and outer fins, International Journal of Hydrogen Energy, 36, pp. 3757- 3763, 2011.
[6] Jay J. Bhavsar, V K. Matawala, S. Dixit, Design and experimental analysis of spiral tube heat exchanger, International Journal of Mechanical and Production Engineering, Volume-1, Issue-1, pp. 2320-2092, 2013.

[7] M. Thirumarimurugan, T. Kannadasan, S. Gopalakrishnan, Performance analysis of cross flow plate fin heat exchanger for an immiscible system using ANN, International Journal of Chemical and Environmental Engineering, Volume-1, No.1, 2010.

[8] A.Vaisi, M. Esmaeilpour and H. Taherian, Experimental investigation of geometry effects on the performance of a compact louvered heat exchanger, Applied Thermal Engineering 31, pp. 3337-3346, (2011).

[9] Mao-Yu Wena, Ching-Yen Ho, Heat-transfer enhancement in the fin-and-tube heat exchanger with improved fin design, Applied Thermal Engineering 29, pp.1050–1057, (2009).

[10] M. Khoshvaght-Aliabadi, F. Hormozi, A. Zamzamian "Role of channel shape on the performance of plate-fin heat exchangers: Experimental assessment" International Journal of Thermal Sciences 79, pp. 183-193, (2014).

[11] Giovanni lozza and Umberto Merlo. ―An experimental investigation of heat transfer and friction losses of interrupted and wauy fins for fin and tube heat exchanger, International Journal of refrigeration 24, pp.409-416, (2001).

[12] H.Iwai, S Kawakami, K.Suzuki, J.Sujii, T.Abiko, Performance evaluation of wire spring fin for compact plate fin heat exchangers, Enhanced Compact and Ultra-Compact Heat Exchangers, Science, Engineering and Technology, CHE2005-21 (2005).

[13] Yonghan Kim and Yongchan Kim, Heat transfer characteristics of flat plate finned-tube heat exchangers with large fin pitch, International Journal of Refrigeration 28, pp. 851–858, (2005).

[14] Praveen Pandey, Rozeena Praveen, S. N. Mishra ―Experimental investigation on the effect of fin pitch on the performance of plate type fins, IJRET, Volume: 1 Issue: 3, pp. 382-388. (2012).

