Three-Dimensional Performance Analysis of Wavy Fin Tube Heat Exchangers

Sujay Kumar Sheel[1], Vikas Kesharwani[2]
[1]*M.Tech Scholar, Department of Mechanical Engg. TIT&R, Bhopal*
[2]*Prof., Department of Mechanical Engg. TIT&R, Bhopal*

***Abstract*- In general, a lot of work on heat exchanger is successfully done; still a lot of work has to be done. In this proposed research work done to study the effects of Reynolds number, fin pitch and tube pitches on the overall Heat transfer and friction factor for wavy fin tube heat exchangers with large tube diameter. To compare the simulated case friction factor (f) and Colburn factor (j) with the experimental data of Bhuyian et al.2013.To compare in-line and staggered configurations.**

***Index terms*- Heat Exchanger, friction factor, Colburn factor, Reynolds number**

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 typical 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 in two different groups such as staggered and inclined arrangement. Fig. 1 shows some typical finned-tube heat exchanger designs especially for plain and wavy structure.

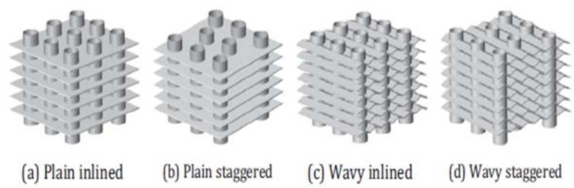


Fig. 1: Different fin and tube arrangement in plain and wavy finned-tube heat exchanger

2. METHODOLOGY

2.1 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:

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

2.2 TURBULENCE MODELS

This section briefly describes basic concepts and classification of different turbulence models. The three turbulence models tested in this study were k-ε model, RNG k-ε model and Baseline (BSL) k-ω model. The fluid flow was governed by the equations described in the following sections named after the above three turbulence models for transitional range flow ($1300 \leq Re \leq 2000$).

Turbulence consists of small scale fluctuations in the flow characteristics over time. It is a complex process, mainly because it is three dimensional, unsteady and chaotic, and it can have a significant effect on the characteristics of the flow. Turbulence occurs when the inertia forces in the fluid become significant compared to viscous forces, and is characterized by a high Reynolds Number.

In principle, the Navier-Stokes equations describe both laminar and turbulent flows without the need for additional information. However, turbulent flows at realistic Reynolds numbers span a large range of turbulent length and time scales and would generally involve length scales much smaller than the smallest finite volume mesh which can be practically used in a numerical analysis. The Direct Numerical Simulation (DNS) of these flows would require computing power which is many orders of magnitude higher than available in the foreseeable future.

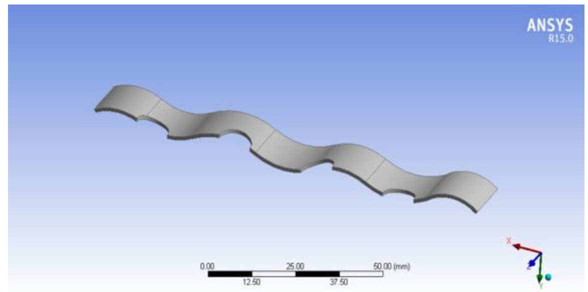
To enable the effects of turbulence to be predicted, a large volume of CFD research has concentrated on methods which make use of turbulence models. Turbulence models have been specifically developed to account for the effects of turbulence without recourse to a prohibitively fine mesh and Direct Numerical Simulation.

2.3 THE CFD MODEL

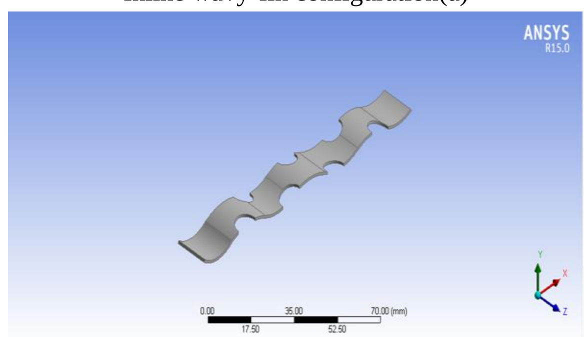
Geometry considered for the present investigation is plain finin-lined and staggered configuration shown in the Fig. 4.1. The z-direction is perpendicular to the paper. Assuming symmetry condition on the midplane between the two fins, the bottom and the top boundaries simulate the fin and the mid-plane respectively. Fig. 4.2 shows the nomenclature used for longitudinal tube pitch (Ll), transverse tube pitch (Lt), fin pitch (Fp), fin thickness (Ft), and tube diameter (D).

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 (Ll) 19.05 mm
Transverse tube pitch (Lt) 25.4 mm
Fin Pitch (Fp) 3.53 mm
The number of tube row (N) 4



Inline wavy-fin configuration(a)

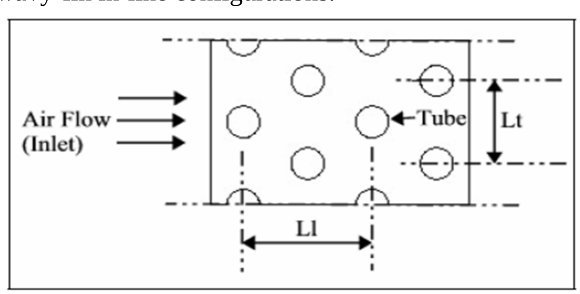


Staggered wavy-fin configuration(b)

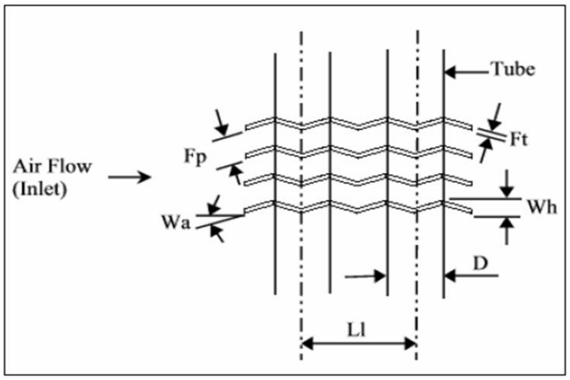
Fig.2: Geometry considered for the present investigation

2.4 NOMENCLATURE

Figure 3 shows the nomenclature used for the wavy-fin staggered configurations used in the present study. Figure 3 (a) explains the nomenclature (the tubes are in staggered order) Note that even though different fin configurations are used in these figures the nomenclature is the same for all two models used in the present study, i.e. wavy fin staggered and wavy-fin in-line configurations.



(a)



(b)

Fig. 3: Nomenclature used with respect to inline wavy-fin and wavy-fin staggered configuration

Figure 3(a) shows the nomenclature used for longitudinal tube pitch (Ll) and transverse tube pitch (Lt). In the present study three different combinations of longitudinal tube pitches (Ll) and transverse tube pitches (Lt) are investigated for their effects on the heat transfer and pressure drop for the plain and wavy fin configurations. The fin thickness (Ft) and the tube diameter (D) are kept constant in all the simulations.

3. RESULT DISCUSSION

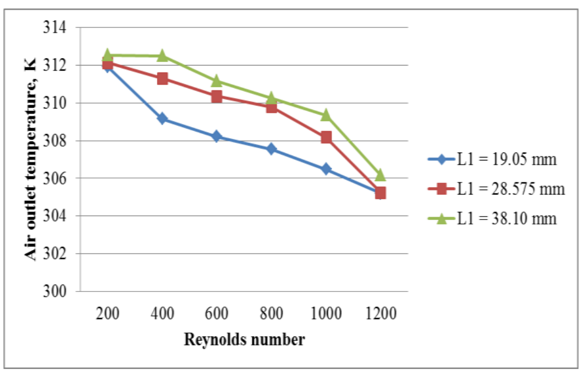


Fig. 4: Effect of varying longitudinal tube pitch on air outlet temperature variation for Wavy fin In-line Configuration

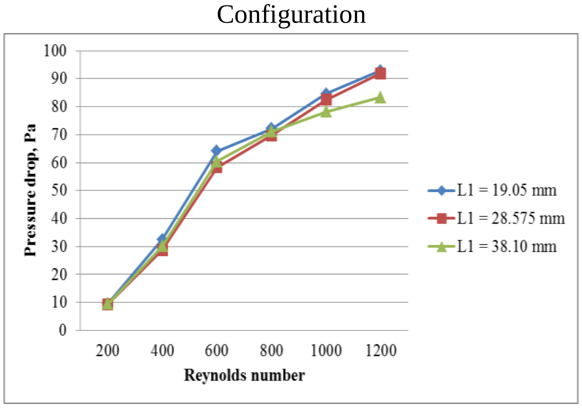


Fig. 5: Effect of varying longitudinal tube pitch on pressure drop variation for Wavy fin In-line Configuration

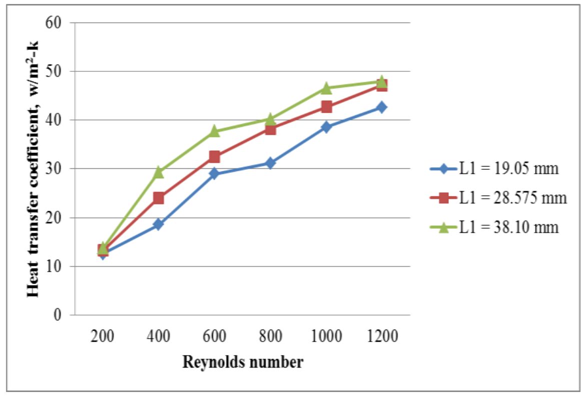


Fig. 6: Effect of varying longitudinal tube pitch on heat transfer coefficient variation for Wavy fin In-line Configuration

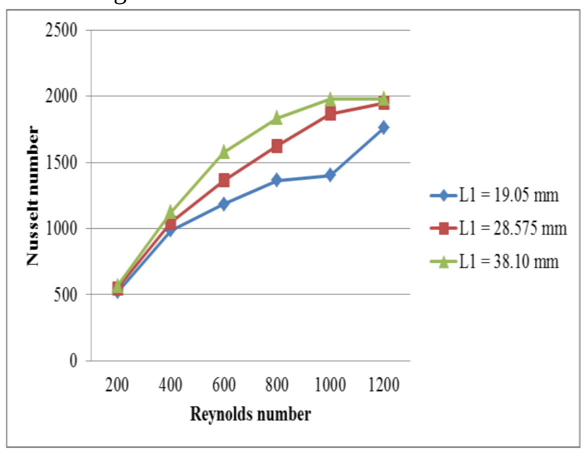


Fig. 7: Effect of varying longitudinal tube pitch on Nusselt number variation for Wavy fin In-line Configuration

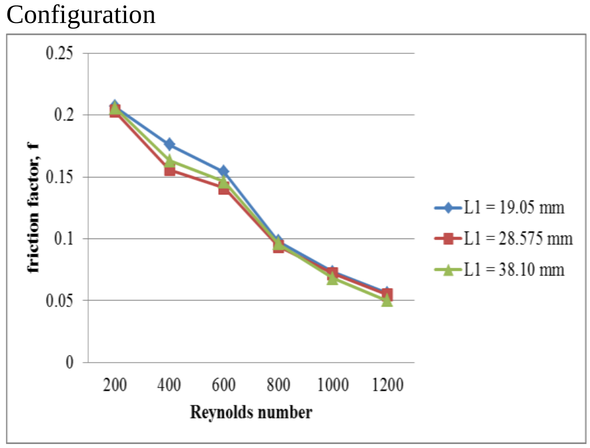


Fig. 8: Effect of varying longitudinal tube pitch on friction factor variation for Wavy fin In-line Configuration

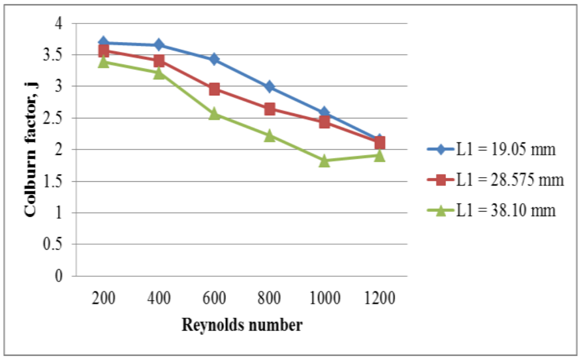


Fig. 9: Effect of varying longitudinal tube pitch on Colburn factor variation for Wavy fin In-line Configuration

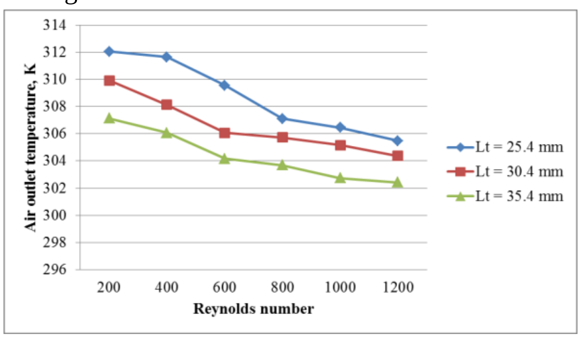


Fig. 10: Effect of varying transverse tube pitch on air outlet temperature variation for Wavy fin In-line Configuration

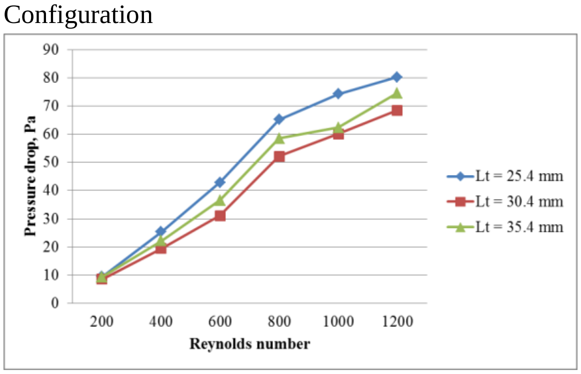


Fig. 11: Effect of varying transverse tube pitch on pressure drop variation for Wavy fin In-line Configuration

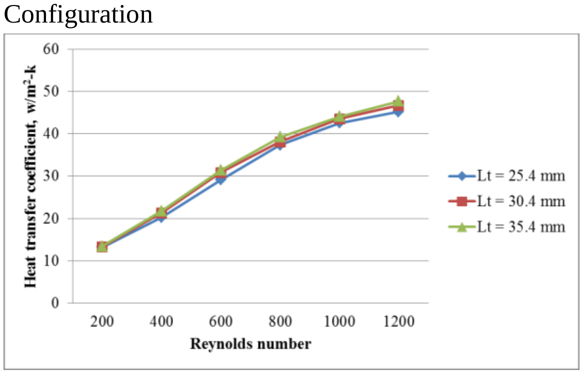


Fig. 12: Effect of varying transverse tube pitch on heat transfer coefficient variation for Wavy fin In-line Configuration

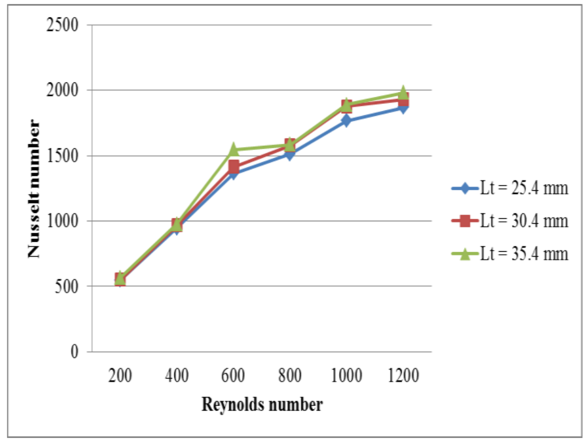


Fig. 13: Effect of varying transverse tube pitch on Nusselt number variation for Wavy fin In-line Configuration

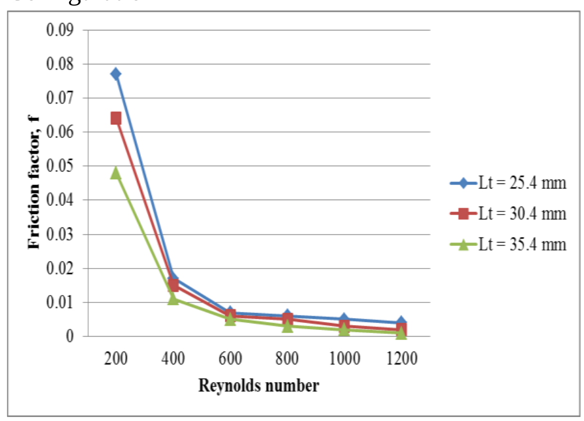


Fig. 14: Effect of varying transverse tube pitch on friction factor variation for Wavy fin In-line Configuration

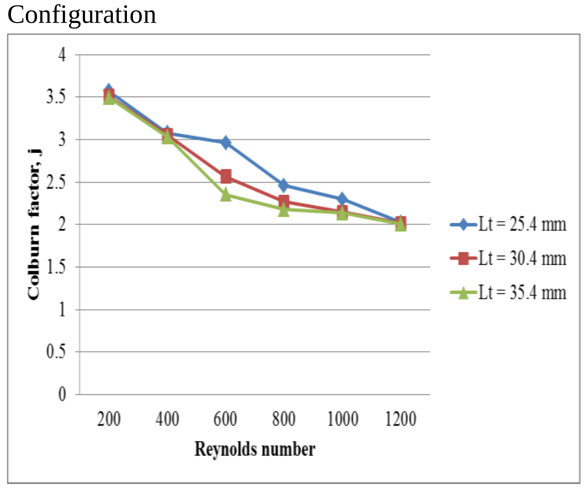


Fig. 15: Effect of varying transverse tube pitch on Colburn factor variation for Wavy fin In-line Configuration

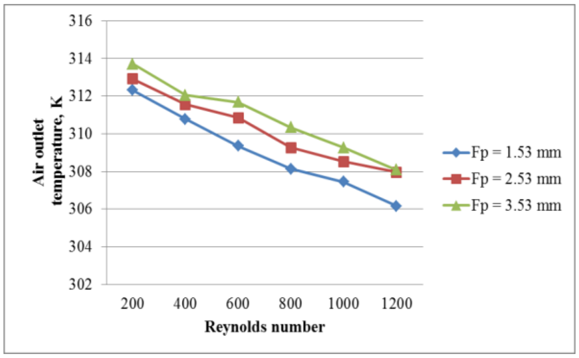


Fig. 16: Effect of varying fin pitch on air outlet temperature variation for Wavy fin In-line Configuration

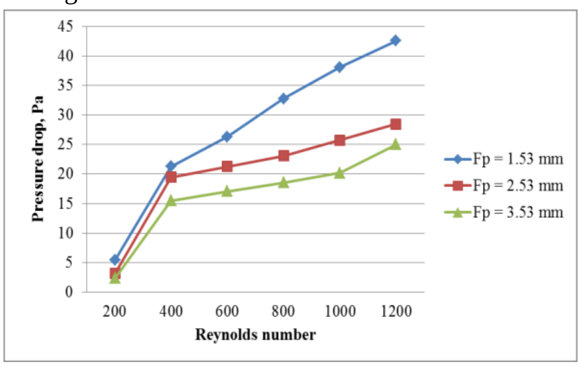


Fig. 17: Effect of varying fin pitch on pressure drop variation for Wavy fin In-line Configuration

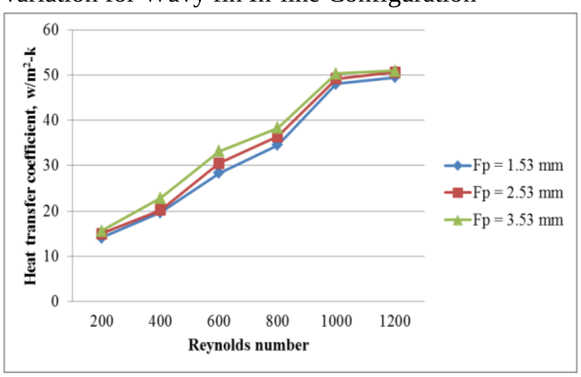


Fig. 18: Effect of varying fin pitch on heat transfer coefficient variation for Wavy fin In-line Configuration

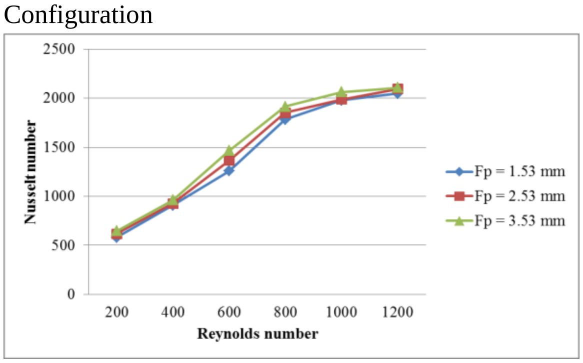


Fig. 19: Effect of varying fin pitch on Nusselt number variation for Wavy fin In-line Configuration

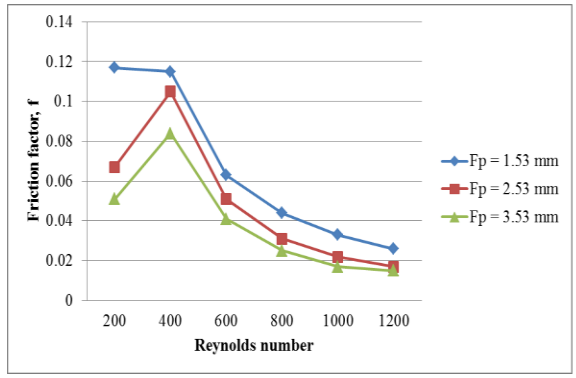


Fig. 20: Effect of varying fin pitch on friction factor variation for Wavy fin In-line Configuration

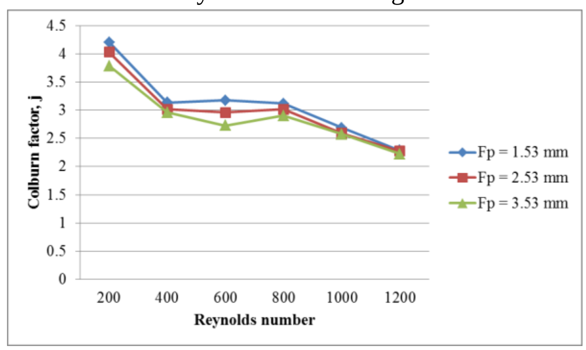


Fig. 21: Effect of varying fin pitch on Colburn factor variation for Wavy fin In-line Configuration

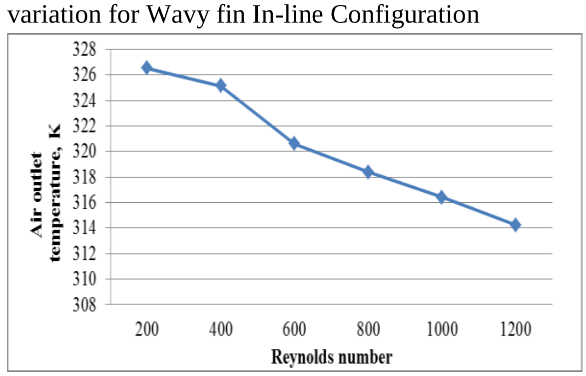


Fig. 22: Air outlet temperature variation for wavy fin staggered configuration

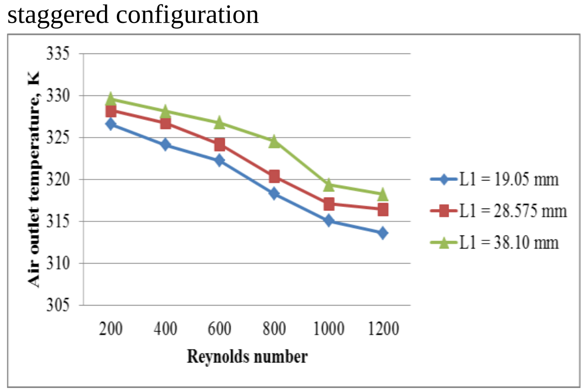


Fig. 23: Effect of varying longitudinal tube pitch on air outlet temperature variation for wavy fin staggered configuration

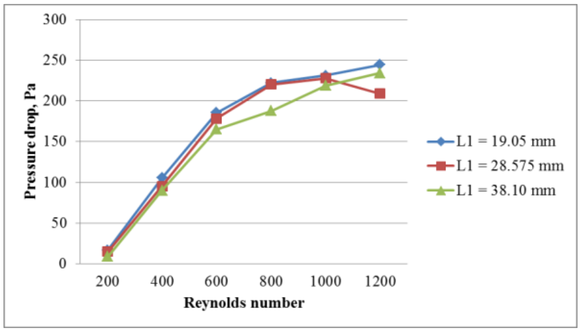


Fig. 24: Effect of varying longitudinal tube pitch on pressure drop variation for wavy fin staggered configuration

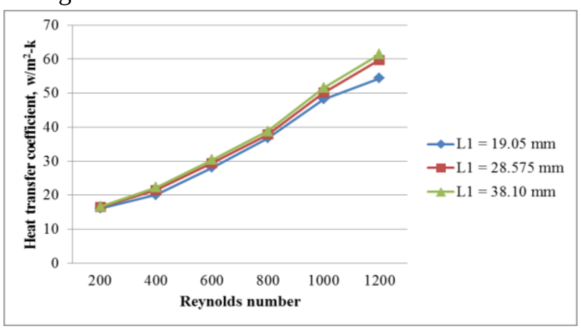


Fig. 25: Effect of varying longitudinal tube pitch on heat transfer coefficient variation for wavy fin staggered configuration

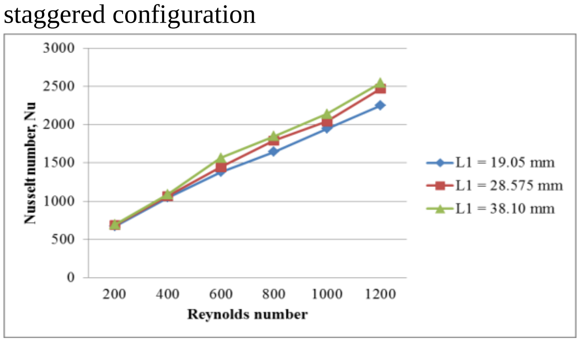


Fig. 26: Effect of varying longitudinal tube pitch on Nusselt number variation for wavy fin staggered configuration

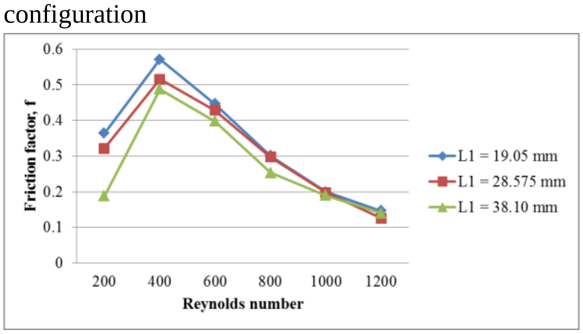


Fig. 27: Effect of varying longitudinal tube pitch on friction factor variation for wavy fin staggered configuration

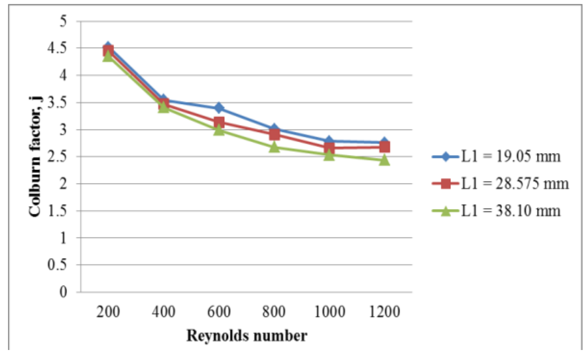


Fig. 28: Effect of varying longitudinal tube pitch on Colburn factor variation for wavy fin staggered configuration

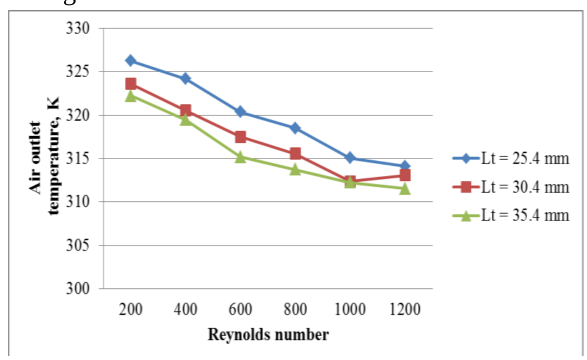


Fig. 29: Effect of varying longitudinal tube pitch on air outlet temperature for wavy fin staggered configuration

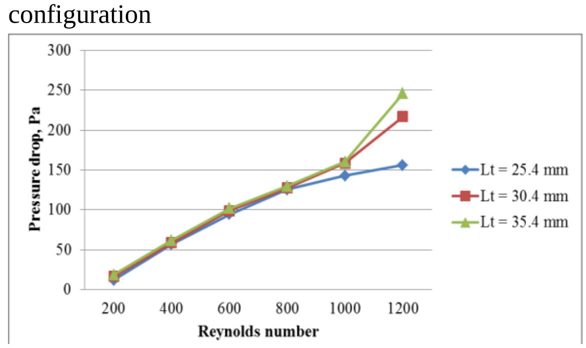


Fig. 30: Effect of varying longitudinal tube pitch on pressure drop for wavy fin staggered configuration

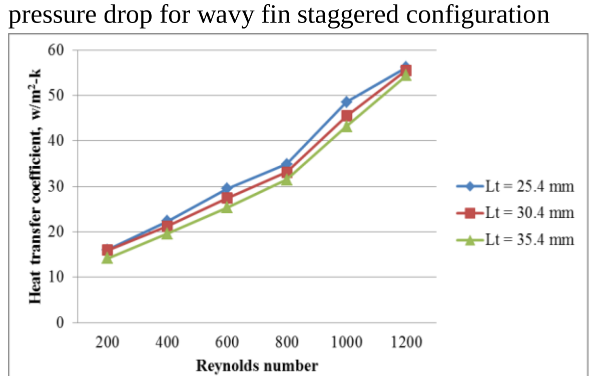


Fig. 31: Effect of varying longitudinal tube pitch on heat transfer coefficient for wavy fin staggered configuration

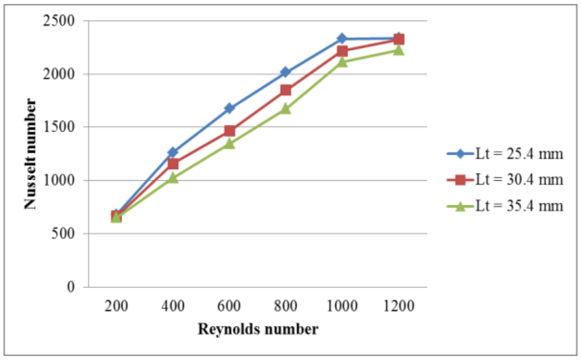


Fig. 32: Effect of varying longitudinal tube pitch on Nusselt number for wavy fin staggered configuration

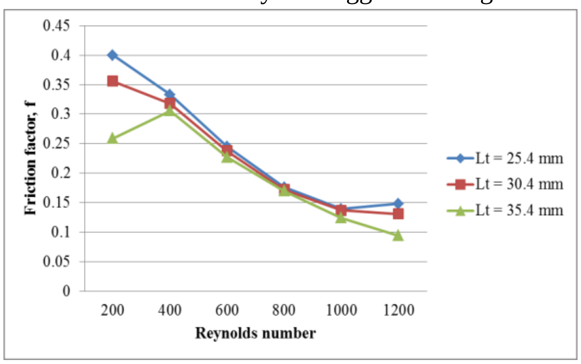


Fig. 33: Effect of varying longitudinal tube pitch on friction factor for wavy fin staggered configuration

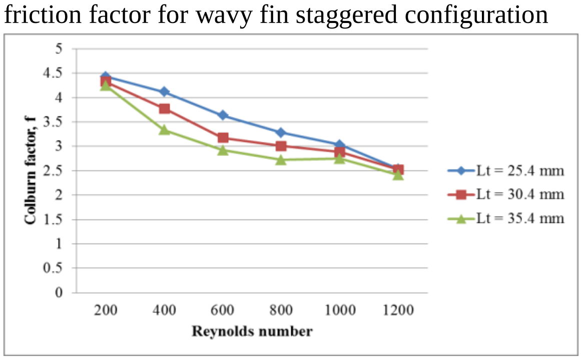


Fig. 34: Effect of varying longitudinal tube pitch on Colburn factor for wavy fin staggered configuration

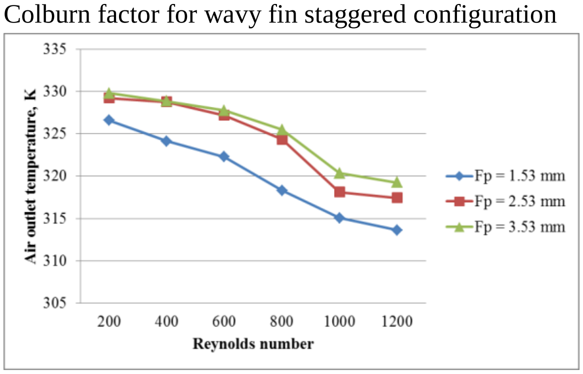


Fig. 35: Effect of varying fin tube pitch on air outlet temperature variation for wavy fin staggered configuration

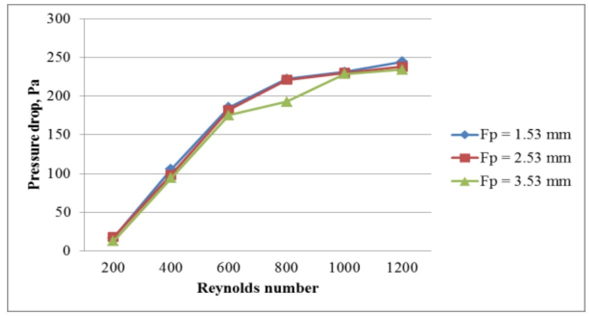


Fig. 36: Effect of varying fin tube pitch on pressure drop variation for wavy fin staggered configuration

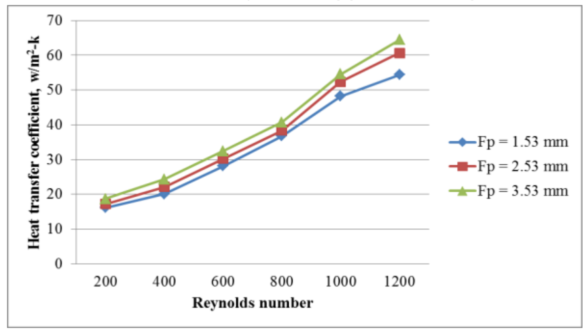


Fig. 37: Effect of varying longitudinal tube pitch on heat transfer coefficient variation for wavy fin staggered configuration

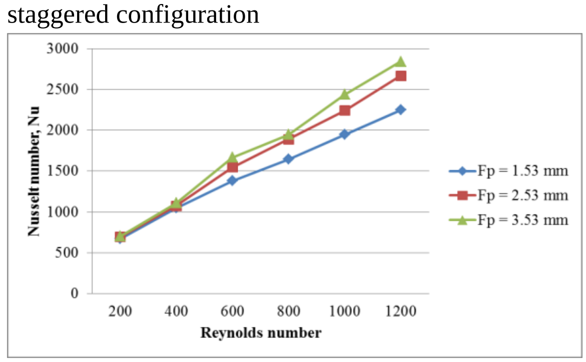


Fig. 38: Effect of varying fin tube pitch on Nusselt number variation for wavy fin staggered configuration

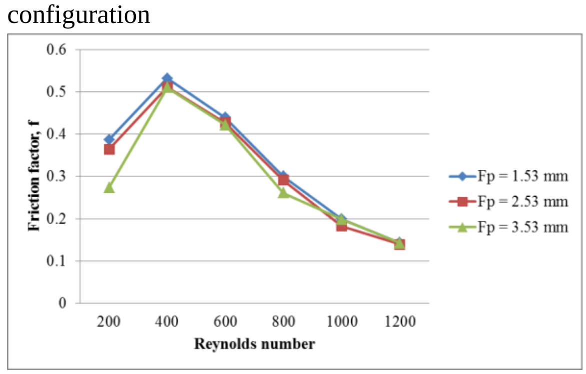


Fig. 39: Effect of varying longitudinal tube pitch on

friction factor variation for wavy fin staggered configuration

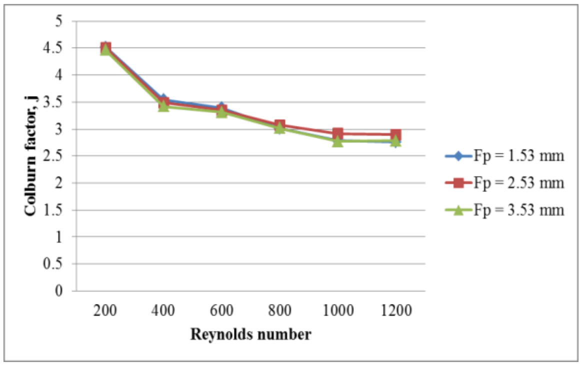


Fig. 40: Effect of varying longitudinal tube pitch on colburn factor variation for wavy fin staggered configuration

4. CONCLUSION

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

1. 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.
2. 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 flow is guided by the wavy corrugations and it is repeatedly re-oriented due to the wavy structure of the fin.
3. 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).
4. However, the pressure penalty is also high in comparison with plain-fin counterpart as indicated by the higher friction factor (f).
5. 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 the heat transfer performance is also accompanied by the corresponding decrease in the pressure drop characteristics.
6. 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).
7. 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 (Lt) constant, when fin pitch (Fp) is reduced, the flow becomes more streamlined. This flow streamlining caused by 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).
8. Overall staggered configuration is recommended over in-line configuration.

REFERENCE

[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 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, YanpengJi, 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 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 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 performance of plate-fin heat exchangers: Experimental assessment" International Journal of Thermal Sciences 79, pp. 183-193, (2014).