Innovative Systems Design and Engineering www.iiste.org
ISSN 2222-1727 (Paper) ISSN 2222-2871 (Online)
Vol 2, No 6, 2011

Numerical Prediction of Laminar Characteristics of Fluid Flow and Heat Transfer in Finned-Tube Heat Exchangers

Arafat A. Bhuiyan, A K M Sadrul Islam

Department of Mechanical Engineering, Islamic University of Technology (IUT)

Board Bazar, Gazipur-1704, Dhaka, Bangladesh

Tel: +88-02-9291254-59, E-mail: arafat@iut-dhaka.edu

M. Ruhul Amin

Department of Mechanical and Industrial Engineering , Montana State University

220 Roberts Hall, Bozeman, MT 59717-3800, USA

Tel: (406) 994-6295, E-mail: ramin@me.montana.edu

Abstract

This study is performed to determine the characteristics of fin and tube heat exchanger for the plain and wavy fin configurations considering in lined and staggered tube arrangements for laminar flow regime. This analysis has been conducted using Commercial Computational Fluid Dynamics Code ANSYS CFX-11. The results are expressed in terms of friction factor (f) and the Colburn factor (j) and efficiency index (j/f).The code was validated by comparing the results obtained with the previously investigated experimental data. The effects of different geometrical parameter such as Longitudinal pitch, Transverse Pitch, Fin Pitch, Wavy angle on the heat transfer and the pressure drop were investigated. This study reveals that the flow distinction between plain and wavy fin has a profound influence on the heat transfer and pressure drop performance. It was observed that, increasing the longitudinal and transverse pitch causes a decrease in the thermal and hydraulic performance of the heat exchanger for low Reynolds Number problem. The result for the fin pitch study indicated that decrease in the fin pitch causes a decrease in both heat transfer and friction characteristics significantly.

Keywords: Friction factor, Colburn factor, Efficiency index, longitudinal pitch, transverse pitch, fin pitch.

1. Introduction

Plate fin-and-tube heat exchangers are employed in a wide variety of engineering applications for instance, in air-conditioning equipment, process gas heaters, and coolers. They are quite compact, light weight, and characterized by a relatively low cost fabrication. The heat exchanger consists of mechanically or hydraulically expanded plurality of equally spaced parallel tubes through which a heat transfer medium such as water, oil, or refrigerant is forced to flow while a second heat transfer medium such as air is directed across the tubes in a block of parallel fins. In such type of heat exchangers , continuous and plain or specially configured fins are used on the outside of the array of the round tubes of staggered or in-lined arrangement passing perpendicularly through the plates to improve the heat transfer coefficient on the gas side. The heat transfer between the gas, fins and the tube surfaces is determined by the flow structure which is in most case three-dimensional. In realistic applications the governing thermal resistance for an air-cooled heat exchanger is usually on the air side which may account for 85% or more of the total resistance (C. C. Wang 1997). As a result to effectively improve the thermal performance and to significantly reduce the size and weight of air cooled heat exchangers that is to improve the overall heat transfer performance, the use of enhanced surfaces is very popular in air cooled heat exchangers, although a continuous plain fin is still a commonly used configuration where low pressure drop characteristics are desired. Wavy or corrugated fin are very popular fin patterns that are developed to improve the heat transfer performance .The wavy surface can lengthen the flow path of the airflow and cause better air flow mixing.

Therefore, higher heat transfer performance is expected compared to the plain plate fin surface. However the higher heat transfer performance of the wavy fin surface is accompanied by the higher pressure drop as compared to the plain fin type.

There have been a number of studies on the pressure drop and heat transfer characteristics of bare tube banks in cross flow. Most of the earlier studies were experimental in nature. The overviews of different researchers have been discussed considering the pattern of the heat exchanger. 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. Available experimental information on the plate fin and tube heat exchangers has been presented reviewed and correlated in the literature (Wang et al. 1996, R. Seshimo and Fujii 1991, Kundu et al. 1992). The experimental data available up to 1994 have been reviewed in the book by McQuiston and Parker (1994). Wang et al. (1996) reported airside performance for 15 samples of plain fin-and-tube heat exchangers. They examined the effects of several geometrical parameters, including the number of tube rows, fin spacing and fin thickness. Wang et al. (1996) argued that the occurrence of "maximum phenomenon" for the Colburn j factors at a large number of tube rows and small fin spacing may not be associated with the experimental uncertainties. Recently, Kim et al. (1999) and Wang et al. (2000) proposed correlations for heat transfer and friction characteristics for several geometric parameters on the thermal and hydraulic performance of a number of plain-fin heat exchangers. Kim and Song (2002) studied the effect of distance between the plates for a single tube row in the range $114 \leqslant Re \leqslant 2660$ and found high heat and mass transfer coefficients in the front of the tube due to the existence of a horseshoe vortex observed in case of a plain-fin.

There are also a number of numerical studies for plate fin-and-tube heat exchangers in the literature. Most of the earlier researchers used two-dimensional (2-D) and laminar flow conditions in their numerical calculations (Kundu 1991). As there is a complicated flow structure between the fins, the three-dimensional (3-D) numerical studies tend to be difficult. Few researchers have reported 3-D modeling for plain-fin configuration in their numerical studies (Bastini et al. 1991, Zdravistch et al. 1994, Mendez et al. 2000). Jang et al. (1996) performed numerical studies over a 3-D multi-row plate¬ fin heat exchanger. Tutar and Akkoca (2004) reported a 3-D transient numerical study which investigates the time-dependent modeling of the unsteady laminar flow and the heat transfer over multi-row (1-5 rows) of plate fin-and-tube heat exchanger. Tutar et al. (2001) reported a three-dimensional numerical investigation which studies the effect of fin spacing and Reynolds number over a single row tube domain for a Reynolds number range of $l200 \leqslant Re \leqslant 2000$.

The first comprehensive study related to the wavy fin pattern was done by Beecher and Fagan (1987). This study measured the effect of air velocity and fin pattern on the airside heat transfer in wavy fin-and-tube heat exchangers using single channel experimental test models. Kim et al. (1996) proposed correlations for predicting the Colburn factor (j) and the friction factor (f) based on the Beecher and Fagan (1987) test data. Several researchers (Ramdhyani 1992, Snyder et al. 1993, Webb 1990, Mirth and Ramadhyani 1994, Wang et al. 1995 and 1997, Wang et al. 1998, Wang et al. 1999, Yan and Sheen 2000, Wang et al. 2002) have conducted the experimental studies on wavy fin and tube heat exchangers.

Few researchers (Patel et al. 1991, Rutledge and Sleicher 1994, Yang et al. 1997, McNab et al. 1998, Jang and Chen 1997) have presented 2D numerical studies on the thermal and hydraulic characteristics for a wavy corrugated channel flow. Since wavy fin heat exchangers are widely used in the industry, the ability of numerical codes to predict the thermal/hydraulic performance of these surfaces is of considerable interest. The literature review shows that few researchers have reported 3-D numerical investigations for the thermal and hydraulic performance of the plain and wavy fin configurations (Tutar et al. 2001, Panse 2005).

2. Governing Equations

The present study was performed considering thermal transport with convective heat transfer. Air is used as working fluid assuming constant properties (k=0.0261W/mK, μ=1.831x10-05 Ns/m^2, Pr=0.736, ρ=1.185 kgm^{-3}). Assuming a steady three dimensional incompressible flow with no viscous dissipation and viscous work, laminar flow conditions are considered. The flow in the laminar range described by the conservation laws for mass (continuity), momentum (Navier-Stokes) and by the energy equations are as follows:

$$\left[\frac{\partial u_i}{\partial x_i} = 0\right] \tag{1}$$

$$\rho\left(\frac{\partial u_i}{\partial t} + u_j\frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_T)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \rho g_i \tag{2}$$

$$\rho C_P\left(\frac{\partial T}{\partial t} + u_j\frac{\partial T}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\lambda + \frac{\mu_T C_P}{Pr_T}\right)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

In Equations (2) and (3) μ_T and Pr_T are turbulent viscosity and turbulent Prandtl number respectively. $Pr_T = 0.9$ was used in the current study. The value of μ_T is determined based on the specific turbulence model that is being used. In k-ω turbulent model the μ_T is linked to the turbulence kinetic energy (k) and turbulence frequency (ω) via the following relation:

$$\mu_T = \rho\frac{k}{\omega} \tag{4}$$

The transport equations for k and ω were first developed by Wilcox (1986) and later it was modified by Menter (1994), can be expressed as:

$$\rho\left(\frac{\partial k}{\partial t} + u_j\frac{\partial k}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{k3}}\right)\frac{\partial k}{\partial x_j}\right] + P_k - \beta'\rho k\omega \tag{5}$$

$$\rho\left(\frac{\partial \omega}{\partial t} + u_j\frac{\partial \omega}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{\omega 3}}\right)\frac{\partial \omega}{\partial x_j}\right] + 2(1 - F_1)\rho\frac{1}{\sigma_{\omega 2}\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j} + \alpha_3\frac{\omega}{k}P_k - \beta_3\rho\omega^2 \tag{6}$$

In equation (6), F1 is a blending function and its value is a function of the wall distance. $F_1 = 1$ and 0 near the surface and inside the boundary layer respectively. The constants of this model (ϕ_3) are calculated from the constants ϕ1 and ϕ2 based on the following general equation.

$$\Phi_3 = F_1\Phi_1 + (1 - F_1)\Phi_2 \tag{7}$$

The model constants are given as $\alpha_1 = 5/9$, β'=0.09, β_1 =0.075, $\sigma_{\kappa 1}$= 2, $\sigma_{\omega 1}$= 2, α_2 = 0.44, β_2 = 0.0828, $\sigma_{\kappa 2}$ =1, $\sigma_{\omega 2}$ =1/0.856. In Equations (2) and (3) if we eliminate the terms containing µT then it becomes the equations for the laminar flow.

3. Computational Details

3.1 Geometry and Coordinate System

Geometry considered for the present investigation is plain and wavy fin in-lined and staggered configuration shown in the Fig. 1. The x-y coordinate system is shown in Fig. 2. 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.

3.2 Boundary Conditions

All numerical simulations are carried out using a finite-volume method. The boundaries of the computational domain consist of inlet and outlet, symmetry planes and solid walls. Symmetry boundary conditions at the centre plane, tube centre plane, top symmetry and bottom symmetry are considered. Uniform flow with constant velocity uin and constant temperature Tin, boundary conditions at the inlet flow is used to trigger the flow unsteadiness in the flow passage. Other velocity components are assumed to be zero. A constant temperature of 25˚C is set at the flow inlet to meet the room air conditions. 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 solid wall with no slip boundary condition and constant wall temperature T_{wall} set to 100˚C. The fins and tubes are assumed to be made of aluminium. The heat exchanger model with its extended volume is illustrated in Figure, while the actual area of interest for the heat exchanger simulation is as a computational domain with the coordinates system shown in figure.

3.3 Grid Sensitivity Test

As the accurateness of the numerical results depends stalwartly on the mesh resolutions, a number of trial simulations were carried out with different mesh resolutions. The grids chosen for the computational domain of the plain fin staggered arrangement had 91821, 141700 and 191480 nodes and wavy fin staggered arrangement had 121574, 183223 and 263318 nodes. The maximum difference in the friction factor and the Colburn factor of the plain fin with 91821 and 141700 nodes were 22.081% and 32.632% respectively, while with 141700 and 191480 nodes the differences were 2.567% and 7.40% respectively. The grid with 141700 nodes for plain fin was chosen for the current investigation with minimum computational time and acceptable accuracy. Also, it was observed that an unstructured mesh system with triangular mesh containing 183223 nodes with 451683 elements is considered for wavy fin to be fine enough to resolve the flow features in all simulations. The graphical representation of grid analysis for wavy fin staggered arrangement is shown in fig. 3.

3.4 Code Validation

To ensure the numerical results are unswerving, calculations were first prepared to scrutinize the recital of fin geometry having 4 rows staggered circular tube configuration with the experimental data by Wang et al. (1997).The detailed geometry of the examined heat exchanger is same as Wang et al. The accuracy of the study was established by comparing the values for friction factor (f) and Colburn factor (j) by Wang et al. for laminar flow. The maximum difference in friction factor and the Colburn factor between the numerical results and the experimental results were found to be 10.33% and 9.81% respectively for the laminar flow range. The graphical presentation is shown in fig. 4 for plain fin staggered arrangements.

4. Result and Discussion

The current investigation is performed for the laminar ($400 \leqslant Re_H \leqslant 1200$) flow range to determine the flow distinction among the plain and wavy fin staggered and in lined configurations using CFD code ANSYS CFX-11. The laminar flow range is selected for the present study because the flow remains in this range for most of the fin and tube heat exchanger as it is recommended by the experimental studies of several researchers.

For the plain fin, flow is fairly straightforward and it gets interrupted only by the tubes. Because of the flow interruption by the tubes, a flow recirculation zone is observed at the trailing edge of the tubes as the flow passes over the tubes. In case of wavy fin, the flow is guided by the corrugations as it gets re-oriented each time it passes over a wavy corrugation. This is distinctive flow pattern compared to plain fin arrangements. For this reason, less flow recirculation is observed in the wake of the tubes. The difference between plain fin staggered and plain fin in-lined configuration can be observed from the streamline and velocity vectors of these arrangements as shown in figs. 5 and 6. For the plain fin staggered configurations flow interruption takes place on both sides of the domain. Because of the repeated interruption of the flow due to staggered tube on both sides of the domain for this fin configuration, a smaller recirculation zone is observed in the trailing edge of the domain. For the plain fin in-lined arrangements flow is blocked only on one side of the domain due to the in-lined arrays of the tubes. As all the tubes for the plain fin in-lined configurations lie on one side of the domain, flow gets separated into two forms as seen from the figure. These two regions can be free flow regions where there is no tube and the stagnant flow regions, in the trailing edge of the tubes. This effect satisfied the answer that higher recirculation zone for the plain fin in-lined configuration as compared with the plain fin staggered configurations. Thus in case of plain fin staggered configurations the flow gets interrupted on both sides of the domain at a regular interval because of the tube layouts resulting in smaller recirculation zone.

Fig. 7 shows the temperature contour for the plain fin staggered , plain fin in-lined , wavy fin staggered and wavy fin in-lined configurations taken on the X-Y Planes at Z=1.765 mm for Re_H=1200 in Laminar flow. Fig. 8 shows the pressure distribution for the same configuration as temperature distributions. For the present analysis inlet air temperature was kept constant at 25˚C and the fin and the tube surfaces were kept as wall boundary with a constant temperature of 100˚C and outlet pressure was assumed to be zero. The temperature profile study provides the same behaviour as the streamline pattern and the velocity vector. Much larger recirculation zones are observed in the trailing edge of the tubes for plain fin in-lined arrangements as compared to the plain fin staggered configurations. But for the wavy fin staggered and in-lined arrangements less recirculation was found. From the temperature profile of plain fin in-lined arrangements, it is found that there is larger high temperature zones in the trailing edge of the tubes can be called warm zones because of the recirculation flow which stretches between two adjacent tubes.

The special effects of different geometrical parameters such as Ll, Lt and Fp on the heat transfer and the pressure drop characteristics are investigated. As mentioned earlier that Laminar flow model is considered for the flow range of $400<Re_H<1200$.

The variation of friction factor (f), Colburn factor (j) and efficiency index (j/f) against the Reynolds number (Re_H) for the laminar flow range for the plain and wavy staggered and in lined configurations are shown in fig. 9. The friction factor (f) and Colburn factor (j) for the plain fin staggered configuration varies from 28.41% to 43.39% and from 21.25% to 36.14% respectively higher than the corresponding plain fin in-lined arrangements. However, the efficiency index (j/f) for the plain fin staggered configuration decreases from 4.7% to 5.5% compared to that of the plain fin in-lined arrangements. On the other hand, the variation in f and j in case of wavy staggered configuration is 13.17%-15.88% and 8.07%-10.98% higher than that of the in-lined arrangements respectively. But the efficiency index (j/f) shows the opposite behaviour as the f and j, as it decreases about 4.02%-4.7% from the in-lined to staggered configuration. It is clear that the tube arrangements play a vital role in the heat transfer and pressure drop characteristics. In staggered arrangements, better flow mixing is observed due to staggered tube layouts and thus provides higher heat transfer and pressure drop characteristics than the in-lined arrangements. So staggered configuration will be considered to investigate the effect of different geometrical parameter in the laminar range.

Fig. 10 shows the variation of the friction factor (f), Colburn factor (j) and efficiency index (j/f) against the Reynolds number (ReH) for the three longitudinal pitch (Ll) cases for the laminar flow range of plain staggered arrangements. This figure indicates that the friction factor (f) decreases with the increase in the longitudinal pitch (Ll). For a Reynolds number of $Re_H=1000$, the friction factor (f) decreases with the increase of longitudinal tube pitch (Ll) from 19.05 to 27.875 mm and 19.05 to 38.10 mm by 12.70% and 24.19% for the plain staggered arrangement. This is because increase in longitudinal pitch decreases flow restriction and as such lower friction factor. The Colburn factor (j) decreases with the increase in the longitudinal pitch (Ll). For example, at ReH=1000, the Colburn factor (j) decreases with the increase of longitudinal tube pitch (Ll) from 19.05 to 27.875 mm and 19.05 to38.10 mm by 8.55% and 15.39% for the staggered arrangement. The reason for this decrease is the same as before, that is, increased pitch decreases flow restriction and hence lowers heat transfer. It can be seen from the figure that efficiency index goes up with the increase of longitudinal (Ll) pitch. Say, at $Re_H=1000$, the efficiency index (j/f) increases with the increase of longitudinal tube pitch (Ll) from 19.05 to 28.575 mm and 19.05 to 38.10 mm by 8.55% and 15.39% for the plain staggered arrangement .

Variation of Ll has significant effects in f, j and j/f with the increase in ReH as illustrated in fig. 11 for wavy staggered configuration. The graphical presentation indicates that f decreases with the increase in Ll. For, the velocity at $Re_H=1000$, f decreases with the increase in Ll from 19.05 to 28.575 mm and 19.05 to 38.10 mm by 11.67% and 20.97% respectively for wavy staggered arrangements. This is because increase in pitch decreases flow restriction and as such lower pressure drop. The same behaviour as f is found for j. For example, at ReH=1000, j decreases with the increase of Ll as f by 5.48% and 9.43% respectively. But the percentage in decrease is different from f. The reason for this decrease is the same as before, that is, increased pitch decreases flow restriction and hence lowers heat transfer. Though f and j decrease with the increase in Ll but j/f goes up with the increase of Ll. For a particular Reynolds Number of $Re_H=1000$ in the laminar range, j/f increase with the increase of Ll as f and j by 7.01% and 14.60% respectively. This can be explained as the percentage increase in f is high compared to j. That's why j/f increases with the increase in Re_H.

The effects of transverse tube pitch (Lt) on the heat transfer, pressure drop and efficiency index for the plain staggered arrangement is shown in Fig. 12. This figure indicates that the friction factor (f) decreases with the increase in the transverse tube pitch (Lt). For a particular Reynolds number of Re_H=1000, the friction factor (f) decreases with the increase of transverse tube pitch (Lt) from 25.4 to 30.4 mm and 25.4 to 35.4 mm by 10.56% and 16.95% for the staggered condition. Colburn factor (j) decreases with the increase of transverse tube pitch (Lt) from 25.4 to 30.4 mm and 25.4 to 35.4 mm by 7.88% and 15.93% for the staggered arrangement .It can be seen from the figure that efficiency index goes up with the increase of transverse tube pitch (Lt). The efficiency index (j/f) increase with the increase of transverse tube pitch (Lt) for the staggered arrangement from 25.4 to 30.4 mm and 25.4 to 35.4 mm are 7.88% and 15.93% respectively.

The effects of transverse pitch (Lt) on the pressure drop and heat transfer and efficiency for the staggered arrangements are shown in fig. 13 for wavy staggered arrangement. These figures indicate that f and j decreases with the increase in Lt. For a particular Reynolds number of Re_H=1000, for the increase of Lt from 25.4 to 30.4 mm and 25.4 to 35.4 mm, the decrease in f is 14.99% and 26.51% and for j, it is about 7.72% and 13.65% respectively. It can also be seen from the fig. 12 that efficiency goes up with the increase of Lt as Ll. It increases about 8.55% and 17.50 % for the same change in Lt as f and j respectively. This can be explained that, efficiency of the heat exchanger depends on good heat transfer between the fluid and the fin and the low pressure drop of the flow. j decreases with the increases of Lt. At the same time, f decreases with the increases of Lt. But the efficiency increases as the percentage decrease in f is more than j. That means as the flow becomes free with the increases of Ll the change of pressure is more significant as compared to heat transfer.

Fig. 14 shows the variation of the friction factor (f), Colburn factor (j) and efficiency index (j/f) against the Reynolds number (ReH) for the three fins pitch (Fp) cases. It shows that the friction factor (f) decreases with the decrease in the fin pitch (Fp). For example, considering laminar flow at Re_H=1000, the friction factor (f) decrease with the decrease of fin pitch (Fp) from 3.53 to 2.53 mm and 3.53 to 1.53 mm by 11.7% and 20.89% for the staggered arrangement. This is because when fin pitch is reduced the flow becomes more streamlined resulting in better flow mixing. Also reduction in the fin pitch (Fp) reduces the tube surface area which affects the friction factor (f). The Colburn factor (j) decreases with the decrease in the fin pitch (Fp). For a particular Reynolds number of Re_H=1000, the Colburn factor (j) decreases with the decrease of fin pitch (Fp) from 3.53 to 2.53 mm and 3.53 to 1.53mm by 5.91 % and 10.59 % for the staggered arrangement . This observation can also be explained on the basis of flow streamlining and the flow simplification keeping constant the longitudinal pitch (Ll) and transverse tube pitch (Lt) while the fin pitch is reduced. The flow simplification reduces the heat transfer performance. The reductions in fin pitch (Fp) affects the heat transfer area and thus minimize the Colburn factor (j). It can be seen from the figure that efficiency index increases with the decreases in fin pitch (Fp). For a particular Reynolds number of Re_H=1000, the efficiency index(j/f) increase with the decrease of Fin pitch(Fp) from 3.53 to 2.53 mm and 3.53 to 1.53 mm are 6.55% and 13.02% for the staggered. This observation suggests that even though the heat transfer performance (j) decrease with the decrease in fin pitch (Fp), the efficiency for the surface goes up due to the corresponding decrease in the friction factor (f). As the pressure drop and the heat transfer both decreases with the decrease in the fin pitches (Fp). This is because the percentage decrease in the heat transfer performance is lower than the percentage decrease in the pressure drop. As a result the efficiency index (j/f) increases with the decrease in the fin pitch (Fp).

Fig. 15 represents the effects of fin pitch (Fp) on the f, j and j/f for wavy staggered arrangement. The effect of this is totally different compared to Ll and Lt. It shows that f decreases with the decrease in Fp. For the decrease in fin pitch (Fp) from 3.53 to 2.53 mm and 3.53 to 1.53 mm, f decreases by 14.23 % and 25.89 % respectively. This is drastic change compared to other pitch. This is because when Fp is reduced the flow becomes more streamlined resulting in better flow mixing. Also reduction in the Fp reduces the tube surface area which affects the pressure drop performance. Heat transfer performance is quite similar as the pressure drop. This surveillance can also be explained on the basis of flow streamlining and the flow simplification keeping constant Ll and Lt, while Fp is reduced. The flow simplification reduces the heat transfer performance. Variation of j/f for different Fp cases point out the reverse effect between j/f and Re_H. In the turbulent range, j/f increases with the decrease in Fp same as f and j by 7.52% and 15.24% respectively.

5. Conclusion

The numerical investigations of heat transfer and pressure drop for the Plain and wavy fin and tube Heat exchanger for laminar flow regime was carried out in this study. ANSYS CFX-11 was used to perform the numerical simulation. The effects of different geometrical parameter such as Longitudinal pitch (Ll), Transverse Pitch (Lt), Fin Pitch (Fp), on the heat transfer and the pressure drop were investigated for the laminar flow range for the four fin configurations. The effect of flow distinction between plain fin and wavy fin is significant. Recirculation zone is the differentiating factor for these two fin configurations. The area of the recirculation zone for the plain fin is more than the wavy fin configurations. On the other hand, in case of in lined arrangements, the recirculation zone is larger than the staggered arrangements in between two adjacent tubes, since the flow is obstructed only on one side of the domain. Wavy fin show larger heat transfer performance as indicated by higher Colburn factor (j). It was found that the increase in the longitudinal pitch (Ll) cause a decrease in the heat transfer and pressure drop performance as the flow becomes free and less compact with the increase in the tube pitch. As the pressure drop decrease is more significant than heat transfer, so the efficiency goes high with the increase in tube pitch. It was found like the effects of longitudinal pitch (Ll) that with the increase in the Transverse pitch (Lt) cause a decrease in the heat transfer and pressure drop performance. As the flow becomes less compact with the increase in the transverse pitch, that's why f and j decrease. The effect of fin pitch (Fp) on the heat exchanger performance demonstrates that decrease in the fin pitch shows opposite performance as the longitudinal and transverse pitches. As the fin pitch decrease, the flow becomes more streamlined. It affects the heat transfer performance as well as pressure drop characteristics. The efficiency index goes up with the decrease in the fin.

References

Ali, M. M. and Ramadhyani, S. (1992), "Experiments on convective heat transfer in corrugated channels", *Experimental Heat Transfer*, Vol. 5, Number 3, pp. 175-193.

Bardina, J.E., Huang, P.G. and Coakley, T.J. (1997), "Turbulence Modeling Validation Testing and Development", *NASA Technical Memorandum* 110446.

Bastini, A., Fiebig, M. and Mitra, N. K. (1991), "Numerical studies of a compact fin-tube heat exchanger", Proceedings of EUROTHERM Seminar No. 18, *Design and Operation of Heat Exchangers*, pp. 154-163, Hamburg, Germany.

Cui, J., and Tafti, D. K. (2002), "Computations of flow and heat transfer in a three dimensional multi-louvered fin geometry", *International Journal of Heat and mass Transfer*, 45(25), pp.-5007-5023.

Dejon, N.C., and Jacobi, A. M. (1999), "Flow, Heat transfer and Pressure drop Interactions in Louvered fin arrays", *ACRC*, Tr-146.

Ergin, S., Ota, M., Yamaguchi, H. and Sakamoto, M. (1996), "Numerical study of the effect of interwall spacing on turbulent flow in a corrugated duct", American Society of Mechanical Engineers, Heat Transfer Division, Vol. 333, Number 2, pp. 47-54, *Proceedings of the ASME Heat Transfer Division*.

Groijans, H. and Menter, F. R. (1998), "Wall functions for general application CFD codes", ECCOMAS 98 *Proceedings of the Fourth European Computational Fluid Dynamics Conference*, pp. 1112-1117.

Guo, D. and Tafti, D. K. (2003), "Effect of Inlet Flow Angle on Performance of Multi-louvered Fin Heat Exchangers", *American Society of Mechanical Engineers, Heat Transfer Division, (Publication) HTD*, Vol. 374, Number 1, pp. 103-109.

Jang, Y. J., Wu, M. C. and Chang, W. J. (1996), "Numerical and experimental studies of three dimensional plate-fin and tube heat exchangers", *International Journal of Heat and Mass Transfer*, Vol. 39, Number 14, pp. 3057-3066.

Kayansayan, N. (1993), "Heat transfer characterization of flat plain fins and round tube heat exchangers", *Experimental Thermal and Fluid Science*, Vol. 6, Number 3, pp. 263-272.

Kim, J. Y., and Song, T. H.(2002), "Microscopic phenomena and macroscopic evaluation of heat transfer from plate fins/circular tube assembly using naphthalene sublimation technique", *International Journal of Heat and Mass Transfer,* Vol. 45, Number 16, pp. 3397-3404.

Kim, N. H., Youn, B. and Webb, R. L.(1999), "Air-side heat transfer and friction correlations for plain fin-and-tube heat exchangers with staggered tube arrangements", *Journal of Heat Transfer*, Transactions ASME, Vol. 121, Number 3, pp. 662-667.

Kim, N. H., Youn, 1. H. and Webb, R. L. (1996), "Heat transfer and friction correlations for wavy plate fin-and-tube heat exchangers", *American Society of Mechanical Engineers, Heat Transfer Division, (Publication) HTD*, Vol. 330, Number 8, pp. 43-52.

Kundu, D., Skeikh, A. H. and Lou, D. Y. S. (1992), "Heat transfer in cross-flow over cylinders between two parallel plates", *Journal of Heat Transfer, Transactions ASME*, Vol. 114, Number 3, pp. 558-564.

Kundu, D., Skeikh, A. H. and Lou, D. Y. S. (1991), "Heat transfer predictions in cross flow over cylinders between two parallel plates", *Numerical Heat Transfer: An International Journal of Computation and Methodology; Part A: Applications*, Vol. 9, Number 3, pp. 361-377.

Kundu, D., Skeikh, A. H. and Lou, D. Y. S. (1991), "Pressure and heat transfer in cross flow over cylinders between two parallel plates", *Numerical Heat Transfer: An International Journal of Computation and Methodology; Part A: Applications*, Vol. 9, Number 3, pp. 345-360.

McNab, C. M., Atkinson, K. N., Heikal, M. R. and Taylor, N.(1996), "Numerical modeling of heat transfer and fluid flow over herringbone corrugated fins" *Proceedings of the 11th International Heat Transfer Conference*, Kyongiu, Korea, Vol. 6, pp. 119-124.

McQuiston, F. C.,and Parker, J.D. (1994), “Heating, Ventilating and Air Conditioning analysis and Design”*John Wiley*, New York.

Mendez, R. R., Sen, M., Yang, K. T. and McClain, R. (2000), "Effect of fin spacing on convection in a plate fin and tube heat exchanger", *International Journal of Heat and Mass Transfer*, Vol. 43, Number 1, pp. 39-51.

Menter, F. R. (1994), "Two-equation eddy-viscosity turbulence models for engineering applications", *AIAA Journal*, Vol. 32, Number 8, pp. 1598-1605.

Mirth, D. R. and Ramadhyani, S.(1994), "Correlations for predicting the air-side nusselt numbers and friction factors in chilled-water cooling coils", *Experimental Heat Transfer*, Vol. 7, Number 2, pp. 143-162.

Panse, S. P.(2005), “A numerical Investigation of thermal and hydraulic characteristics in three dimensional plate and wavy fin-tube heat exchangers for Laminar and transitional flow regimes”, *M. Sc. Thesis*, Montana State University, Bozeman , Montana, USA.

Patel, V. c., Chon, J. T. and Yoon, J. Y.(1991), "Laminar flow over wavy walls", *Journal of Fluids Engineering, Transactions of the ASME*, Vol. 113, Number 4, pp. 574-578.

Patel, V. C., Chon, J. T. and Yoon, J. Y.(1991), "Turbulent flow in a channel with a wavy wall", *Journal of Fluids Engineering, Transactions of the ASME*, Vol. 113, Number 4, pp. 579-586.

Rutledge, J. and Sleicher, C. A. (1994), "Direct simulation of turbulent flow and heat transfer in a channel Part II: A Green's function technique for wavy walls", *Communications in Numerical Methods in Engineering*, Vol. 10, Number 6, pp. 489-496.

Seshimo, Y., and Fujii, M. (1991), "Experimental study on the performance of plate fin and tube heat exchangers at low Reynolds numbers", *ASME/JSME Thermal Engineering Joint Conference*, Vol. 4, pp. 449-454.

Snyder, B., Li, K. T. and Wirtz, R. A.(1993), "Heat transfer enhancement in a serpentine channel", *International Journal of Heat and Mass Transfer*, Vol. 36, Number 12, pp. 2965-2976.

Suga, K. and Aoki, H.(1991), “Numerical study on Heat transfer and pressure drop in Multi-louvered fins”, *ASME/JSME Thermal engineering Proceedings*, Vol. 4, pp.361-368.

Tafti, D. K., Zhang, L. W., and Wang, G. (1999), “A time dependent calculation procedure for fully developed and developing flow and heat transfer in Louvered fin geometries”, *Numerical Heat Transfer* A,35(3), pp. 225-249.

Tutar, M. and Akkoca, A. (2004), "Numerical analysis of fluid flow and heat transfer characteristics in three-dimensional plate fin-and-tube heat exchangers", *Numerical Heat Transfer; Part A: Applications*, Vol. 46, Number 3, pp. 301-321.

Tutar, M., Akkoca, A. and Serdar, 0. (2001), "A numerical study of heat transfer and fluid flow in a plate fin- and- tube heat exchanger", *American Society of Mechanical Engineers, Pressure Vessels and Piping Division)(Publication) PVP*, Vol. 431, Number 3, pp. 77-84.

Wang, C. C. and Chi, K. Y.(2000), "Heat transfer and friction characteristics of plain fm/ tube heat exchangers, Part I: New experimental data", International Journal of Heat and Mass Transfer, Vol. 43, Number 15, pp. 2681-2691.

Wang, C. C., Chang, Y. 1., Hsieh, Y. C. and Lin, Y. T. (1996), "Sensible heat and friction I characteristics of plate fin-and-tube heat exchangers having plane fins", *International Journal of Refrigeration*, Vol. 19, Number 4, pp. 223-230.

Wang, C. C., Chi, K. Y. and Chang, C. 1.(1997), "Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, Part II: Correlation", *International Journal of Heat and Mass Transfer*, Vol. 43, Number 15, pp. 2693-2700.

Wang, C. C., Fu, W. L. and Chang, C. T. (1997), "Heat transfer and friction characteristics of typical wavy fin-and-tube heat exchangers", *Experimental Thermal and Fluid Science*, l' Vol. 14, Number 2, pp. 174-186.

Wang, C. C., Hwang, Y. M. and Lin, Y. T. (2002), "Empirical correlations for heat transfer and (flow friction characteristics of herringbone wavy fin-and-tube heat exchangers", International Journal of Refrigeration, Vol. 25, Number 5, pp. 673-680.

Wang, C. C., Lee, W. S. and Sheu, W. J.(2001), "A comparative study of compact enhanced finned-tube heat exchangers", *International Journal of Heat and Mass Transfer*, Vol. 44, Number 18, pp. 3565-3573.

Wang, C. C.,Jang., J. Y., Chiou,. N. F.(1999), “A heat transfer and friction correlation for wavy fin and tube heat exchangers”, *International Journal of Heat and mass transfer*; 42(10):1919-24.

Wang, C. C., Lin, Y. T., Lee, C. 1., Chang, Y. 1.(1998), "Investigation of wavy fin-and-tube heat exchangers: A contribution to databank", *Experimental Heat Transfer*, Vol. 12, Number 1, pp. 73-89.

Wang, C. C., Tsi, Y. M. and Lu, D. C.(1998), "Comprehensive study of convex-louver and wavy fin-and-tube heat exchangers" *Journal of Thermo-physics and Heat Transfer*, Vo1.l2, Number 3, pp. 423-430.

Wang, C. C., Wu, M. C. and Chang, C. T. (1995), "Finned tube heat exchangers: Wavy fin geometry", *Proceedings of Asia Pacific Conference on Built Environment*, Singapore, Vol. 1, pp. 194-204.

Yan, W. M., Sheen, P. 1. (2000), "Friction heating analysis of piston rod of shock absorber and the correlation between temperature and hardness", *International Journal of Heat and Mass Transfer*, Vol. 43, Number 9, pp. 1651-1659.

Yang, L. C., Asako, Y., Yamaguchi, Y. and Faghri, M. (1997), "Numerical prediction of transitional characteristics of flow and heat transfer in a corrugated duct", *ASME Journal of Heat Transfer*, Vol. 119, pp. 62-69.

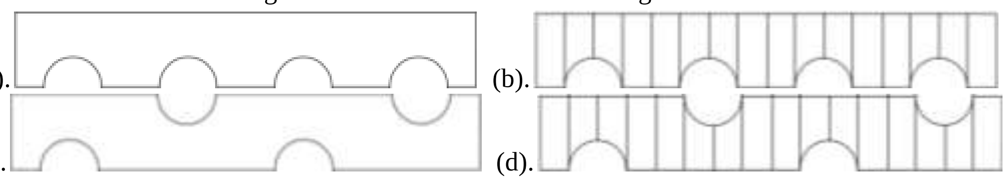
Zdravistch, F., Fletcher, C. A 1. and Behnia, M. (1994), "Laminar and turbulent heat transfer Predictions in tube banks in cross flow", Proceedings of the International Conference On Fluid and Thermal Energy

Convention, Bali, Indonesia, pp. 12-16.

Wilcox, D. C., (1986), "Multi-scale model for turbulent flows", *AIAA 24th Aerospace Sciences Meeting, American Institute of Aeronautics and Astronautics.*

(a). Plain In lined (b) Plain Staggered (c) Wavy in lined (d) Wavy Staggered

Figure 1: Different fin and tube arrangement



(a). (b). (c). (d).

Figure 2: Coordinate system of fin configuration.

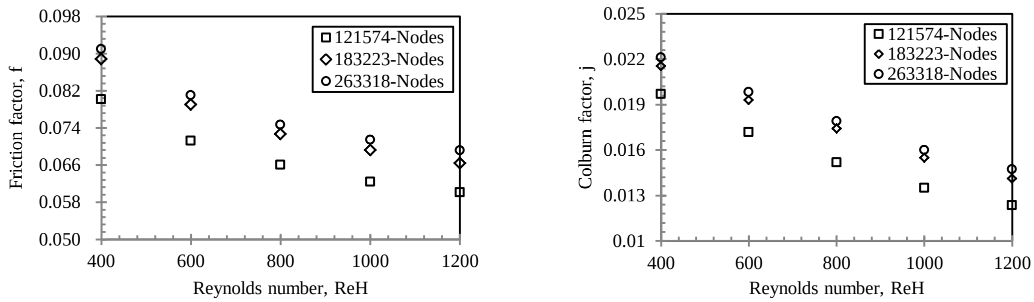


Figure 3: Friction factor (f) and Colburn factor (j) for different grid resolution for wavy fin

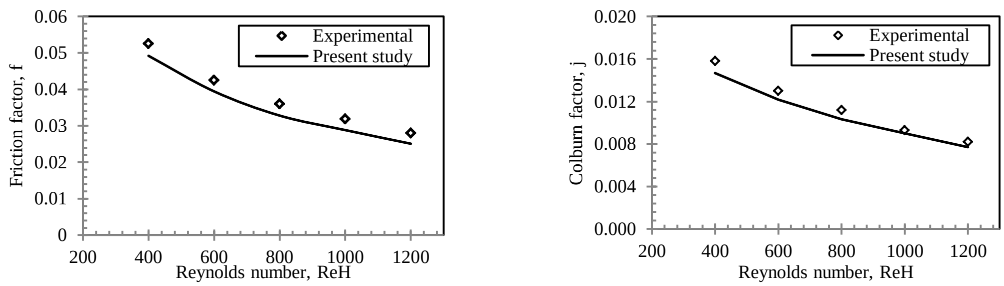
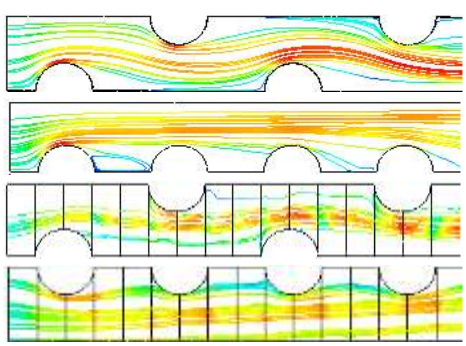


Figure 4: Friction factor (f) and Colburn factor (j) compared with the experimental data of Wang et al (1997).



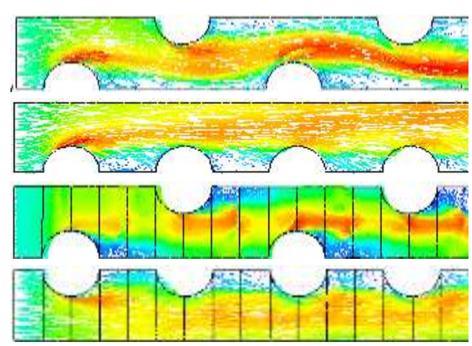
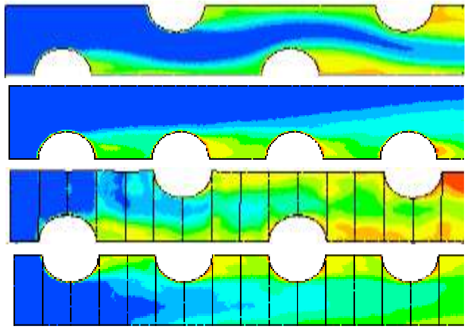


Figure 5: Streamline pattern of fin configuration Figure 6: Velocity Vectors of fin configuration



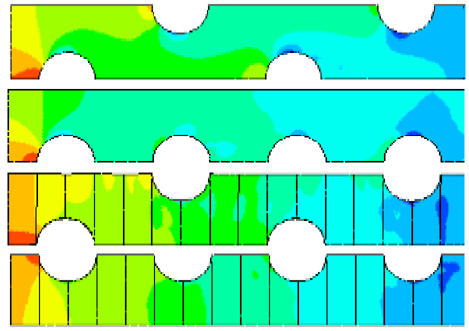


Figure 7: Temperature contour of fin configuration Figure 8: Pressure distribution of fin configuration

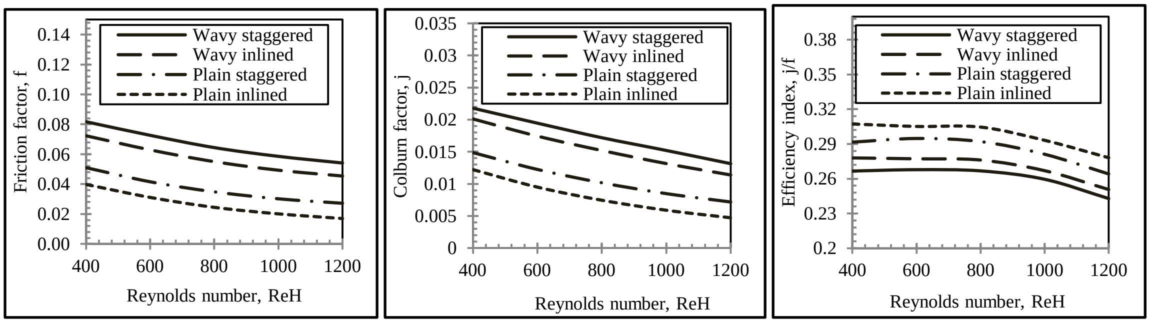


Figure 9: Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for plain and wavy fin and tube heat exchangers considering in lined and staggered arrangements

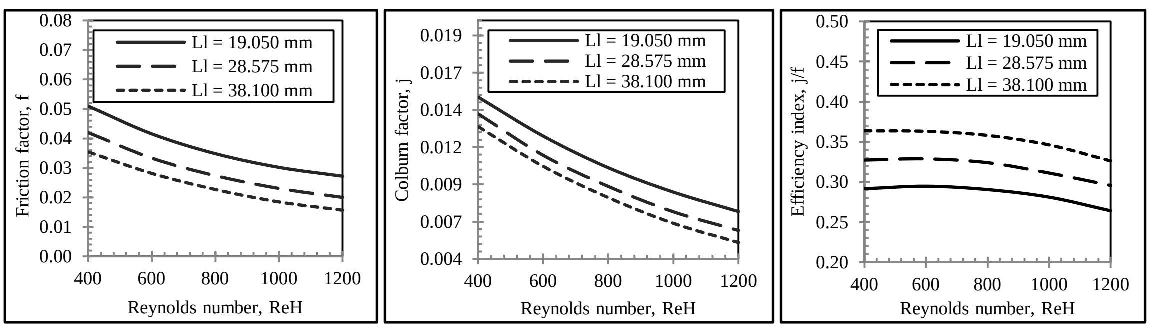


Figure 10: Variation of Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for different Longitudinal pitch (Ll) of plain fin staggered arrangement.

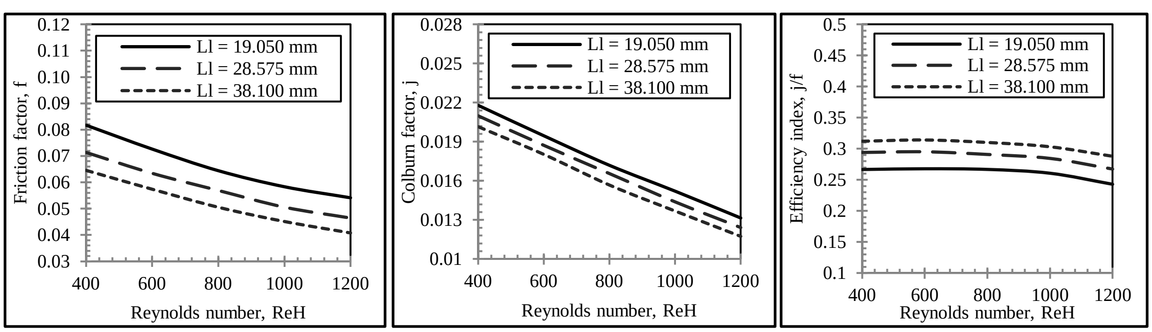


Figure 11: Variation of Friction factor (f), Colburn factor (j), and Efficiency index (j/f) for different Longitudinal pitch (Ll) of wavy fin staggered arrangement.

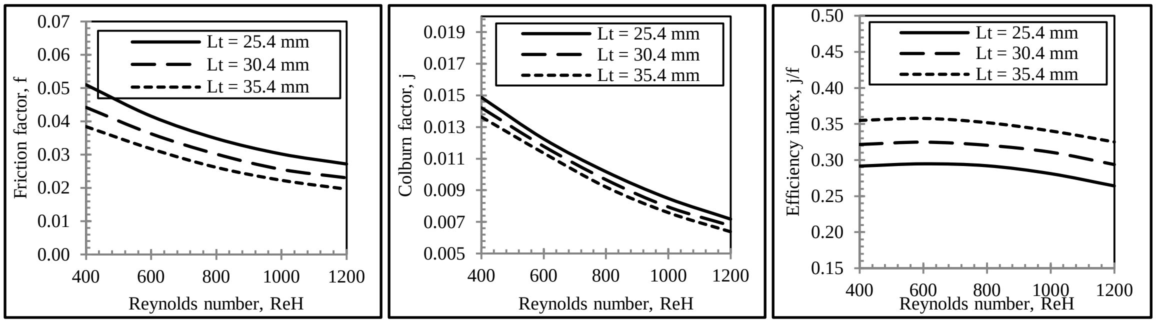


Figure 12: Variation of Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for different Transverse pitch (Lt) of plain fin staggered arrangement.

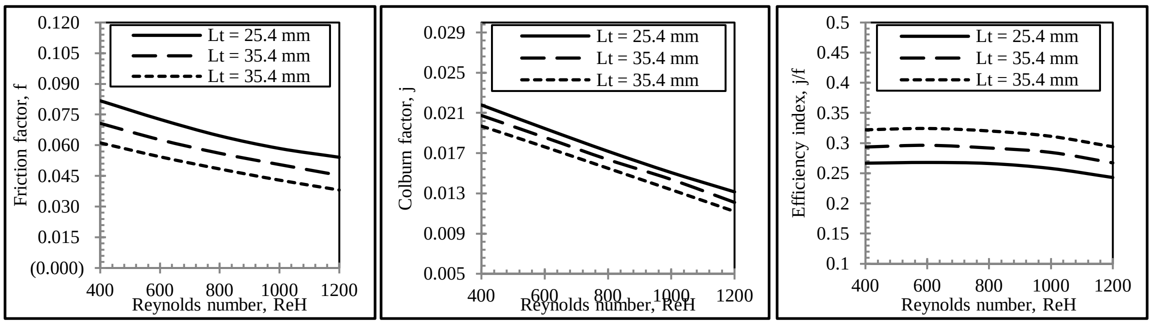


Figure 13: Variation of Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for different Transverse pitch (Lt) of wavy fin staggered arrangement.

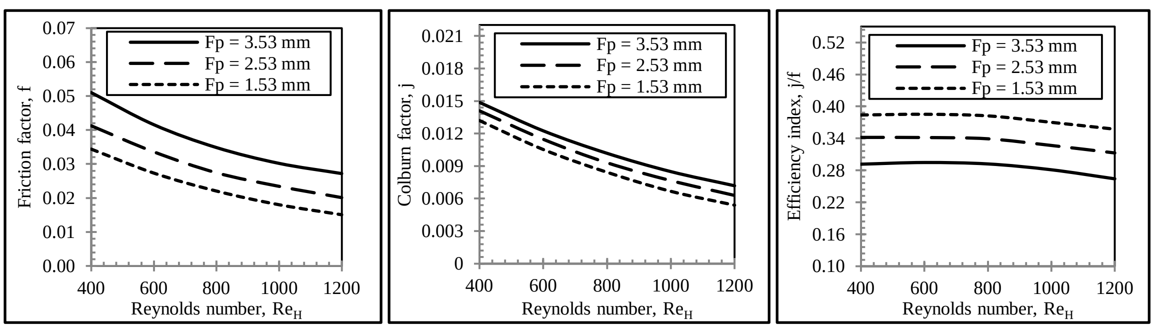


Figure 14: Variation of Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for different Fin pitch (Ft) of plain fin staggered arrangement.

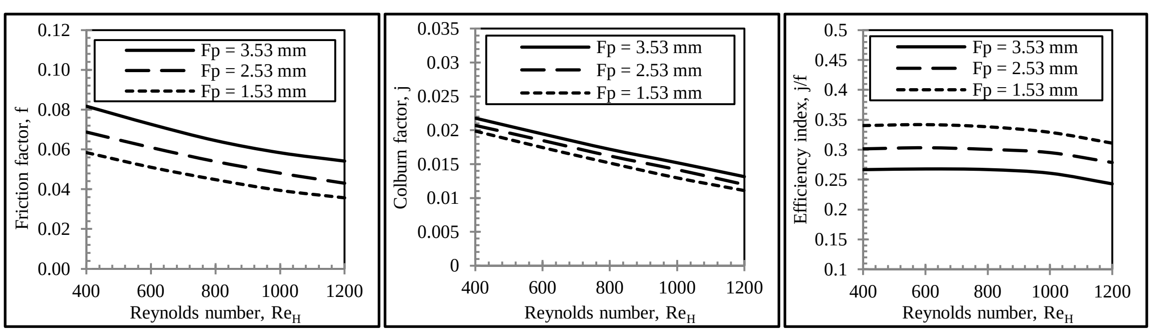


Figure 15: Variation of Friction factor (f), Colburn factor (j) and Efficiency index (j/f) for different Fin pitch (Ft) of wavy fin staggered arrangement.