



A NUMERICAL INVESTIGATION ON HEAT TRANSFER AND PRESSURE DROP PERFORMANCE OF TYPICAL WAVY FIN-AND-TUBE HEAT EXCHANGER

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ABSTRACT

This numerical analysis presents the airside performance of a wavy fin-and-tube heat exchanger having 4 row configurations considering steady, incompressible and 3D flow. Results are presented in the form of friction factor (f), Colburn factor (j) and efficiency index (j/f). The effects of longitudinal pitch (Ll), transverse pitch (Lt), Fin Pitch (Fp), wavy angle (Wa) and inlet flow angle (α) on wavy fin-and-tube heat exchanger are studied. Different inlet flow angles such as 0° , $\pm 17.5^\circ$ and $\pm 35^\circ$ are employed normal to the face of the heat exchanger. There is a strong correlation between the response of the flow efficiency, pressure drop and heat transfer performance to these parameters.

Keywords: Wavy fin-tube heat exchanger, friction factor (f), Colburn factor (j), efficiency index (j/f), longitudinal pitch (Ll), transverse pitch (Lt), fin pitch (Fp), wavy angle (Wa) and inlet flow angle (α).

1. INTRODUCTION

Fin-and-tube heat exchangers are usually used in the process and HVAC&R industries which consist of a group of fins set parallel to one another at prearranged spacing. In sensible applications the principal 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 [Wang, 1997]. As a result to efficiently get better thermal performance and also 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 well-liked in air cooled heat exchangers. Kays and London, 1984] provide the most comprehensive design database though it is containing relatively old data. Wavy or corrugated fins 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 other plate fin surface. In the present study wavy fin is considered for the determination of the performance in the fin-and tube heat exchangers.



For wavy fin having circular tube configurations, extensive experimental data were reported in the literature on the fluid flow and heat transfer characteristics. Correlations were also proposed for those characteristics by the researchers. Beecher et al. has done the first comprehensive study [Beecher and Fagan, 1987] and projected correlations for predicting the Colburn factor (j) and the friction factor (f) based on this study was proposed by Kim et al. [Kim, 1996]. Several experiments were performed for the convective heat transfer in 2D channel as mentioned in the literature [Ali and Ramadhyani, 1992, Snyder and Wirtz, 1993]. Also a lot of experiments were performed to settle on the heat transfer and pressure drop characteristics of 3D wavy fin-and-tube heat exchangers as documented in [Wang, 1997, Wang, 1995, Wang, 1998, Wang, 1999, Yan, 2000]. They also gave correlations for use in describing the Colburn factor and friction factor. Wang et al. [2002] proposed the most updated heat transfer coefficient and friction factor correlations wavy fin-and-tube heat exchangers.

In addition to the experimental study, there are also some numerical investigations for the wavy fin heat exchangers. These investigations provided valuable information of the fundamental phenomenon of the fin geometry. 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 substantial attention. But most of previous numerical studies were 2D in nature and assume laminar flow for the wavy conduit. [Asako, 1987, Amano, 1985, Nishimura, 1987, Xin, 1988, Patel, 1991, Rutledge, 1994, Yang, 1997]. However only a few researchers have used 3-D simulations when side wall effects are important, as in tube banks with fins [McNab, 1996, Jang, 1997]. Few experimental data is available which explores the effects of Wavy angle (Wa) though this parameter is important which determine the heat transfer and pressure drop characteristics. The numerical study by Jang et al. [1997] explores the effect of these parameters on the performance of the wavy-fin configuration. Panse [2005] explores the effects of wavy angle and wavy height for the wavy fin staggered configuration for the transitional flow range.

Even though wavy fins have been studied extensively in the past two decades, little attention has been given to the low Reynolds number flow regime. Most of the preceding numerical investigations presume that the inlet flow is normal to the heat exchanger face. Guo and Tafti [2003] carried out the study experimentally to investigate the effect of inlet flow angle in multi-louvered fin. His study revealed that there is a strong correlation between the response of the flow efficiency and the heat transfer coefficient to the inlet flow angle. Also to the best of authors' knowledge, no studies have critically evaluated the effect of inlet flow angle on performance –all studies assume that the flow is normal to the wavy fin-and-tube heat exchanger face.

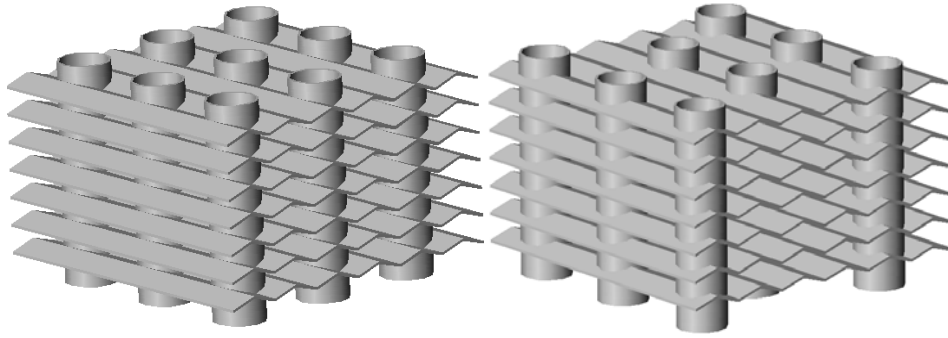


Figure 1: In-lined and staggered tube arrangement of wavy fin-and -tube heat exchanger

2. PHYSICAL MODEL AND GOVERNING EQUATIONS

The present study is based on thermal transport with convective heat transfer for the wavy fin staggered and in-lined arrangements as shown in fig. 1 considering steady 3D incompressible flow with no viscous dissipation and viscous work. Air is assumed as the working fluid with constant properties. Flow is considered turbulent and three turbulent models are used namely standard k- ϵ , k- ω and RNG k- ϵ model. The flow is described by the conservation laws for mass (continuity), momentum (Navier-Stokes) and by the energy equations are as follows:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

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

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

Where, μ_T and Pr_T are turbulent viscosity and turbulent Prandtl number respectively.

As suggested by Yuan [2000], $Pr_T = 0.9$ 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:

$$\mu_T = \rho \frac{k}{\omega} \quad (4)$$

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:

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

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



In equation (6), F_1 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 ϕ_1 and ϕ_2 based on the following general equation.

$$\phi_3 = F_1 \phi_1 + (1 - F_1) \phi_2 \quad (7)$$

The model constants are given as $\alpha_1 = 5/9$, $\beta' = 0.09$, $\beta_1 = 0.075$, $\sigma_{\kappa 1} = 2$, $\sigma_{\omega 1} = 2$, $\alpha_2 = 0.44$, $\beta_2 = 0.0828$, $\sigma_{\kappa 2} = 1$, $\sigma_{\omega 2} = 1/0.856$. Details of these different turbulent models are documented in literature [Panse, 2005, Wilcox, 1986, Menter, 1994, Reddy and Gartling, 1994].

The average Nusselt number Nu , defined as:

$$Nu = \frac{h.H}{\lambda} \quad (8)$$

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

$$f = \frac{(P_{in} - P)}{1/2 \cdot \rho \cdot u_{in}^2} \frac{H}{4L}, \quad j = \frac{Nu}{Re_H \cdot Pr^{1/3}} \quad (9)$$

3. COMPUTATIONAL DETAILS

3.1 Domain configuration

The computational domain for the present study is defined by $0 \leq X \leq 16.16D$, $0 \leq Y \leq 2.66D$ and $0 \leq Z \leq 0.37D$, where tube diameter, $D = 9.525\text{mm}$. The computational domain extends farther than the heat exchangers as seven times to downstream of it, to reduce the numerical oscillations [Mendez et al., 2000]. The nomenclature considered in the present analysis are longitudinal pitch (L_l), transverse pitch (L_t), fin pitch (F_p) and Wavy angle (Wa) as shown in fig. 2.

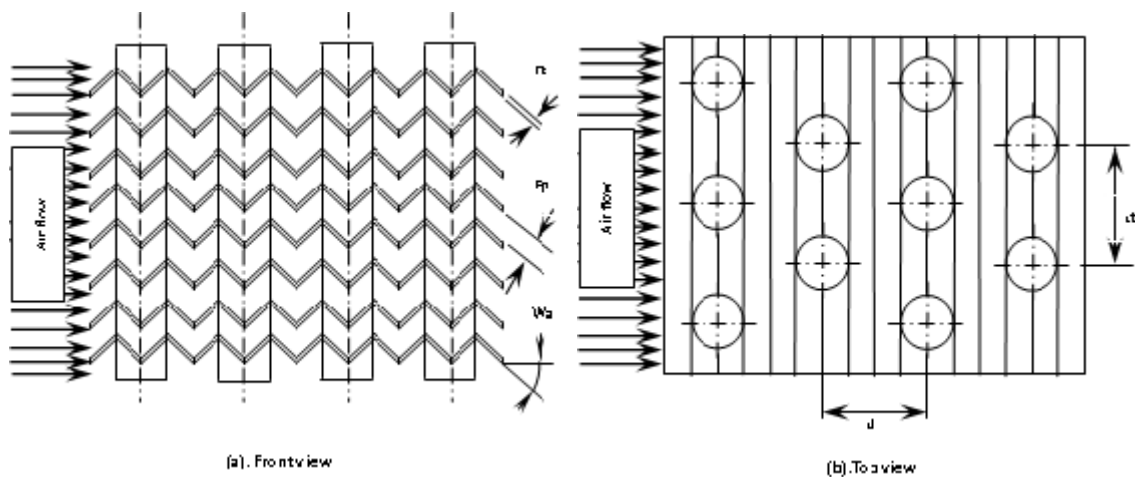


Figure 2: Nomenclature with respect to wavy fin staggered configuration

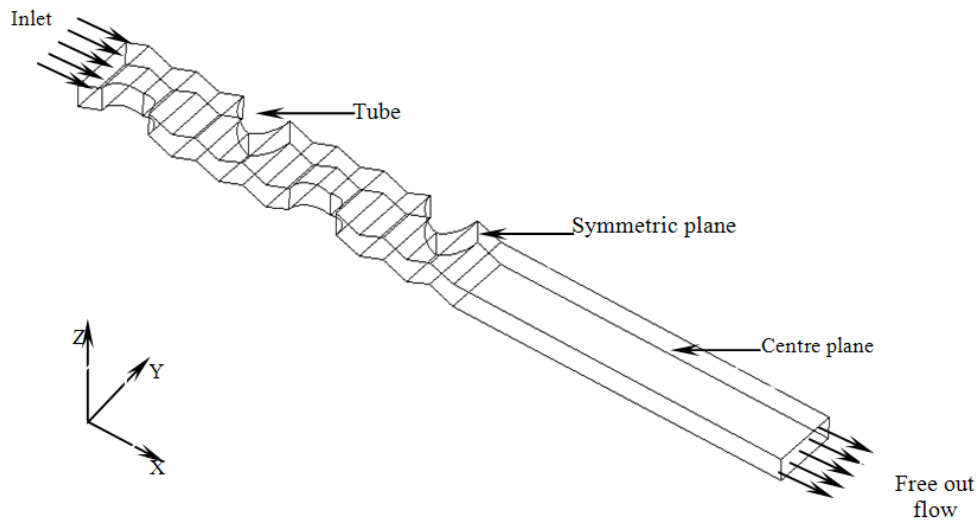


Figure 3: Boundary conditions of the wavy fin-and-tube heat exchanger of staggered type

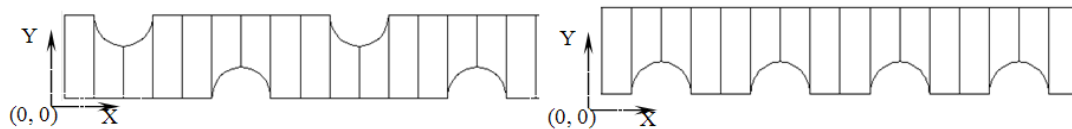


Figure 4: Computational domain and the Coordinate system

3.2. Coordinate system and 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 as illustrated in fig. 3. Symmetry boundary conditions at the centre plane, tube centre plane, top symmetry and bottom symmetry are considered. Uniform flow with constant velocity u_{in} and constant temperature T_{in} , 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 aluminum. Computational domain and the coordinates system is shown in fig. 4.

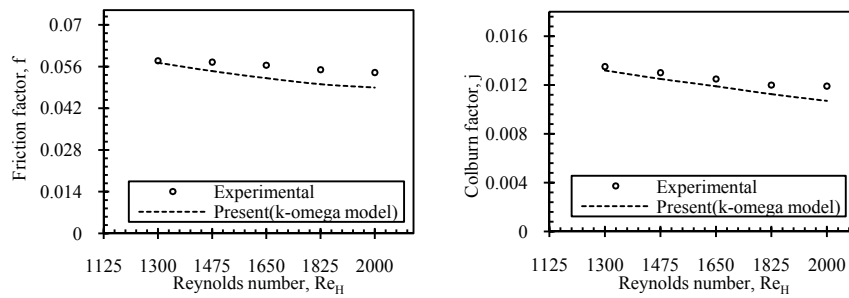


Figure 6: Comparison between obtained numerical and the experimental data.

3.3 Grid Independence

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 three different grid size chosen are 121574, 183223 and 263318 nodes. The variation of friction factor and Colburn factor of the domain are investigated to attain the independence of f and j for a given number of nodes. It was observed that an unstructured mesh system with triangular mesh containing 183223 nodes with 451683 elements is considered to be fine enough to resolve the flow features in all simulations.

3.4 Code validation

To ensure the numerical results are unswerving, calculations were first prepared to scrutinize the recital of wavy 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. As mentioned earlier that three different turbulent models were tested to settle on the most accurate method for the current analysis. Among these, k- ω model showed the best matching with the experimental data published by others. The maximum differences in f and j between the numerical and the experimental data using the k- ω model were found to be 10.22% and 11.25% respectively as presented in fig. 6.

4. RESULTS AND DISCUSSION

For the flow feature, the effects of heat transfer and pressure drop between the wavy fin staggered and in lined configurations are studied for the turbulent flow regime. The special effects of different geometrical parameters such as Ll , Lt , Fp and Wa on the heat transfer and the pressure drop characteristics are investigated. The effects of inlet flow angle are also studied as it has significant effects on the heat transfer and pressure drop characteristics. Flow path for wavy fin is complex as



there is wavy corrugation as the flow passes through the domain. The flow interruption is caused by the wavy corrugation and the tubes. So the flow is re-oriented at each of the time when wavy corrugation starts. For this reason, less flow recirculation is observed in the wake of the tubes. This is distinctive flow pattern compared to other fin arrangements. The difference between staggered and in-lined configuration can be observed from the velocity vectors and the streamlines pattern as shown in figs. 7 and 8 for the staggered and in-lined arrangements. For the 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, a smaller recirculation zone is observed in the trailing edge of the domain. On the other hand for the 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 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 in-lined configuration as compared with the staggered configurations.

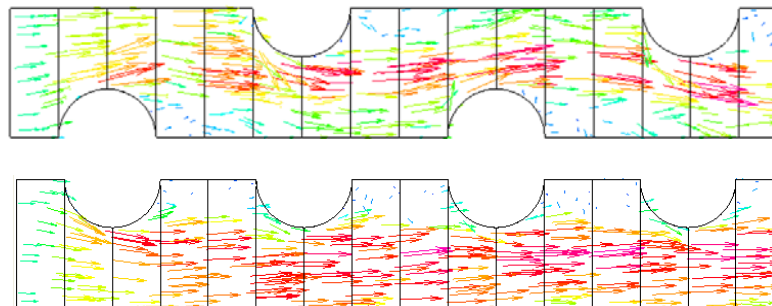


Figure 7: The velocity vectors for the staggered and in-lined arrangements

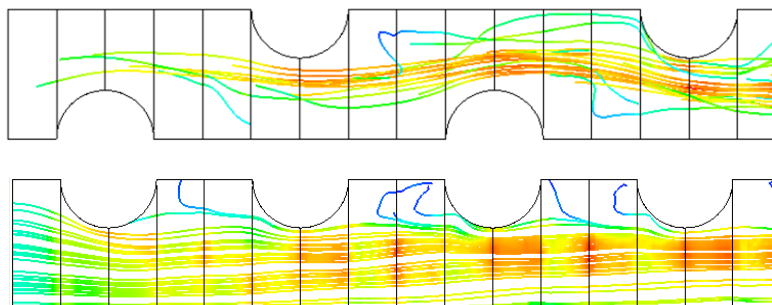


Figure8: Streamlines pattern for the staggered and in-lined arrangements

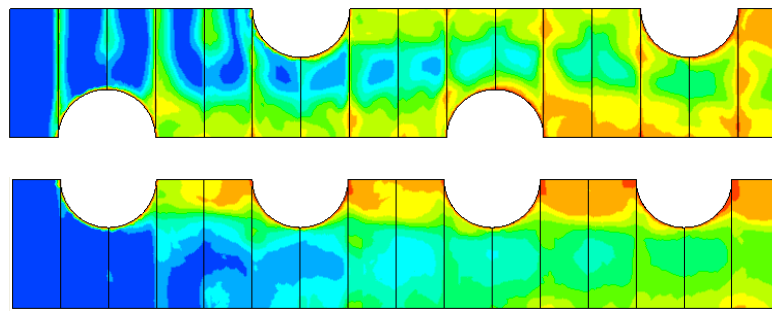


Figure 9: Temperature contour for the staggered and in-lined arrangements

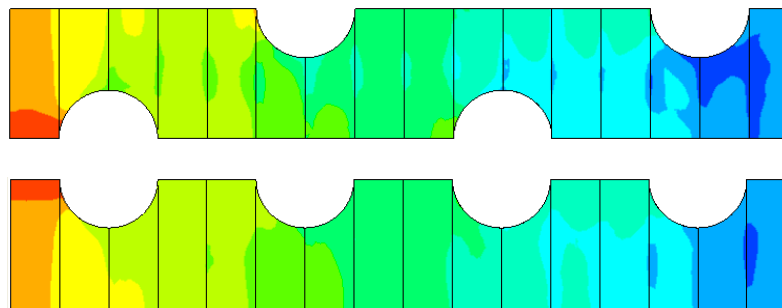


Figure 10: Pressure distribution for the staggered and in-lined arrangements

Figs. 9 and 10 show the temperature contour and the pressure distribution for the staggered and in-lined arrangements taken on the X - Y planes at $Z=1.765$ mm for $Re_H=1200$. The temperature profile study provides the same behavior as the streamline pattern and the velocity vector. Much larger recirculation zones are observed in the trailing edge of the tubes for in-lined arrangements as compared to staggered configurations. From the temperature profile of 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 variation of friction factor (f), Colburn factor (j) and efficiency index (j/f) against the Reynolds number (Re_H) for different tube arrangements in the turbulent flow range are shown in fig. 11.

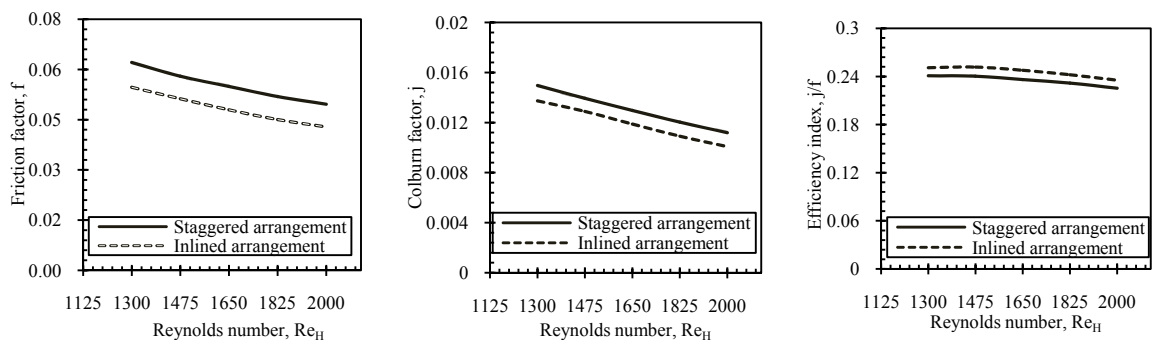


Figure 11: Comparison of performance characteristics for different tube arrangements



The variation in f and j in case of 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 behavior 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 wavy staggered configuration will be considered to investigate the effect of different geometrical parameter in the turbulent range.

Variation of Ll has some effects in f , j and j/f with the increase in Re_H for turbulent flow range as illustrated in fig. 12. The graphical presentation indicates that f decreases with the increase in Ll . For, high velocity at $Re_H=2000$, 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. This is because increase in pitch decreases flow restriction and as such lower pressure drop. The same behavior as f is found for j . For example, at $Re_H=2000$, 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 $Re_H=2000$ in the turbulent 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 .

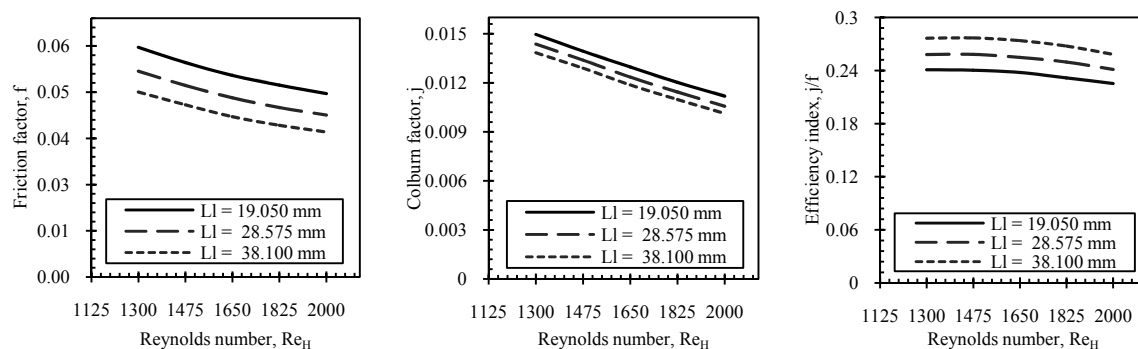


Figure 12: Effects of longitudinal tube pitch (Ll) for turbulent flow range

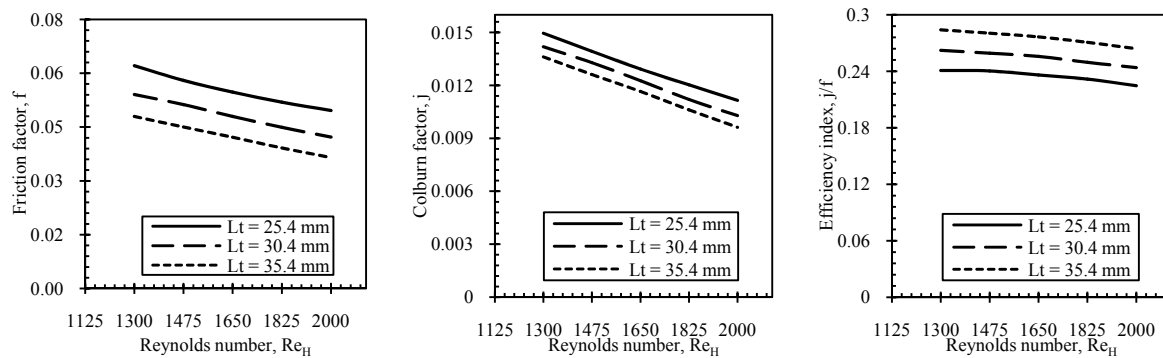


Figure 13: Effects of transverse tube pitch (L_t) for turbulent flow range

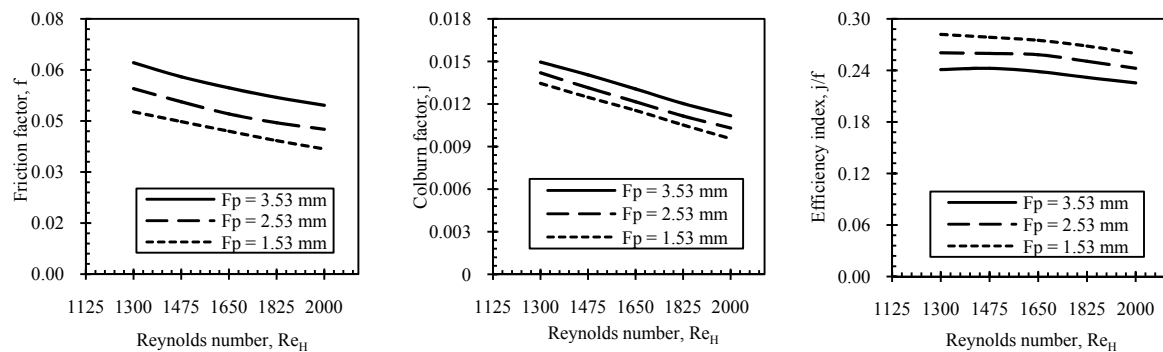


Figure 14: Effects of fin pitch (F_p) for turbulent flow range

The effects of transverse pitch (L_t) on the pressure drop and heat transfer and efficiency for the staggered arrangements are shown in fig. 13. These figures indicate that f and j decreases with the increase in L_t . For a particular Reynolds number of $Re_H=2000$, for the increase of L_t 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 figure that efficiency goes up with the increase of L_t as L_t . It increases about 8.55% and 17.50 % for the same change in L_t 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 L_t . At the same time, f decreases with the increases of L_t . 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 L_t the change of pressure is more significant as compared to heat transfer.

Fig. 14 depicts the effects of fin pitch (F_p) on the f , j and j/f . The effect of this is totally different compared to L_t and L_t . It shows that f decreases with the decrease in F_p . For the decrease in fin pitch (F_p) from 3.53 to 2.53 mm and 3.53 to 1.53 mm, f decreases by 14.23% and 25.89%. This is drastic change compared to other pitch. This is because when F_p 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.

Wavy angle (Wa) is one of the important parameter in the design of heat exchanger of wavy type. Little attention has given on the performance of different wavy angle in the literature. In the present approach, mesmerizing performance was observed in the performance of various Wa as shown in fig. 14. The pressure drop and heat transfer increases with wavy angle for all the fin pitches. The increasing trend is especially significant as Fp is reduced from 3.53 mm to 1.53 mm. Figure also be evidence for that smaller Fp would result in better heat transfer performance. This is attributable to periodic renewal of the boundary layer. This can be explained that when the wavy angle increased, the length of the wavy corrugation decreased. As a result there are more wavy corrugation for a given flow length. This wavy corrugation provides the flow complexity and better air flow mixing. That's why f and j increases with the increase in Wa . But j/f increase with the decrease in Wa . With the increase in Wa , twice and four times from 8.75° the decrease in j/f is about 40.45 % and 74.47 % respectively. This is totally a momentous change. Due to increase in the wavy angle the heats transfer performance increase. At the same time f increases. But j/f decrease, because of the drastic increase in pressure drop compared to heat transfer.

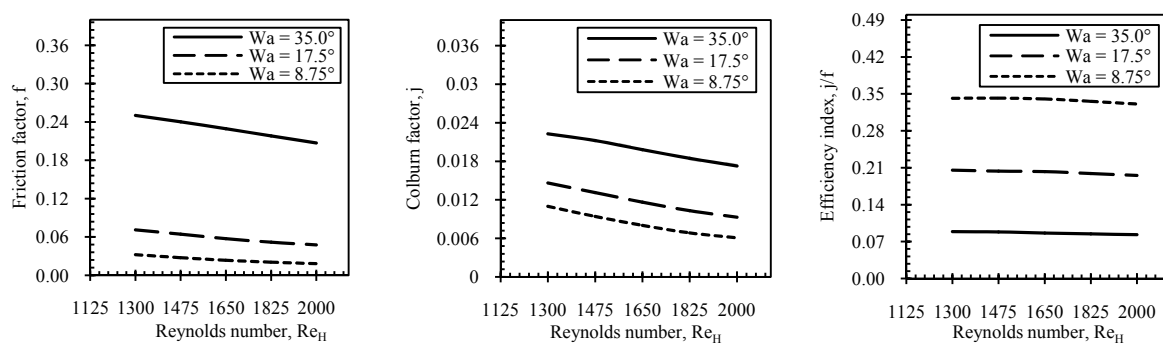


Figure 14: Effects of wavy angle (Wa) for turbulent flow range

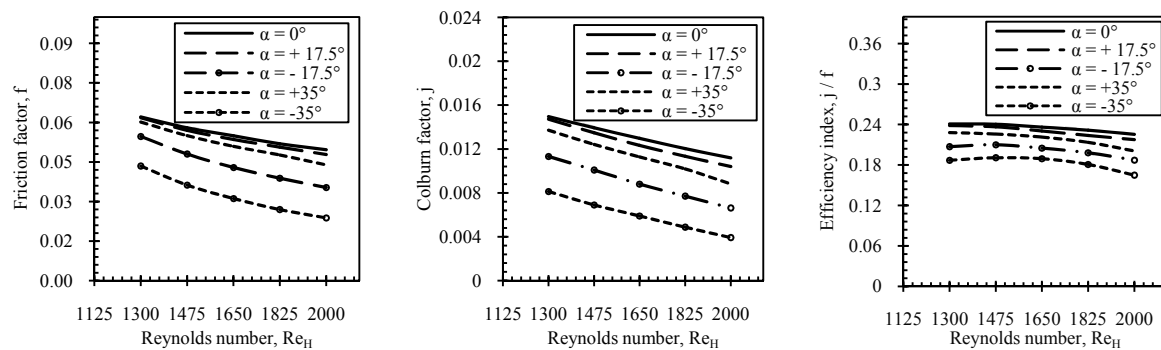


Figure 15: Effects of inlet flow angle (α) for turbulent flow range

The special effects of inlet flow angle on the heat transfer and pressure drop distinctiveness are investigated in this study. The effects of flow angle on the flow efficiency, f and j show similar trend in the turbulent flow as the laminar. With the increase in inlet flow angle which is in the same direction as the wavy angle called positive flow angle, the effects in the heat transfer and pressure drop is decreased. Again, with the decrease in inlet flow angle which is in the negative direction as the wavy angle called negative flow angle, the effects in those criteria is same. But the decrease for the negative flow angle is higher compared to the positive flow angle. The variation of f , j and j/f against Re_H for different inlet flow angle are presented in fig. 15. Five different inlet flow angle (α) such as 0° , $+17.5^\circ$, -17.5° , $+35^\circ$ and -35° are considered in the present study. The decrease in f and j with $(+\alpha)$ from 0° to $+17.5^\circ$ and 0° to $+35^\circ$ are 3.56%, 11.51% and 6.92% and 20.21% while with $(-\alpha)$ these are 28.78%, 52.08% and 40.81% and 64.96% respectively at point, $Re_H=2000$. Figure also shows the dissimilarity of j/f against Re_H for different inlet flow angle. For a particular Reynolds number of $Re_H=2000$, the decrease in the efficiency index j/f with the negative flow angle are more significant compared to positive flow angles.

5. ACKNOWLEDGEMENT

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6. CONCLUSION

In this paper, the effects of tube arrangements, different geometrical parameters and inlet flow angles are investigated in terms of heat transfer and pressure drop and efficiency for the wavy fin-and-tube heat exchanger for turbulent flow regime. The $k-\omega$ turbulence model is employed with 5% turbulence intensity. Based on the present numerical study, the following results are obtained:



1. Comparatively higher heat transfer and pressure drop is found in staggered arrangement than in lined for both laminar and turbulent case.
2. By increasing Ll and Lt, f and j both decreases as the flow becomes free and less compact. But efficiency goes high.
3. Deduction in Fp shows opposite performance as Ll and Lt because of flow streamlining.
4. Higher f and j are observed for more inclination in a given flow length, j/f decrease with the increase in the wavy angle.
5. The positive flow angles provide the higher performance compared to negative angle.

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