CHAPTER 4

STUDIES ON WAVY FINS

The wavy fin is one of the most popular fin types in plate fin heat exchangers, particularly where superior heat transfer performance is demanded under tight pressure drop allowance. They are uninterrupted surface with cross section shape similar to that of plain fins except for the undulations in the flow direction. In aircraft applications, it is the preferred fin type on the ram air side, where available pressure drop allowance is rather small. Parametric study is carried out on the wavy fin parameters using CFD software. In addition to that generalized correlations in the form of \( f \) and \( j \) have been developed for both laminar and turbulent regimes.

4.1 PARAMETRIC STUDY ON WAVY FIN PARAMETERS

The wavy fin performance depends on the fin parameters like the fin height \((h)\), fin spacing \((s)\), fin amplitude \((A)\), fin wavelength \((\lambda)\) and wavy tip radius\((R)\). To study the effect of fin parameters individually, the fin parameter for which the effect is to be studied is varied from minimum to maximum values, keeping the remaining all other parameters as constants. The minimum and maximum values are selected based on manufacturing feasibility. In order to carry out the parametric studies, a set of 25 fins are selected. The influence of individual parameter plays a significant role on the performance of heat exchangers. These effects are studied using CFD software. The computational model and CFD results are explained in following sections.
4.1.1 Geometrical Details and Computational Domain

The geometrical features and computational domains of the three dimensional wavy fin - 1 among the 25 fins considered for flow analysis are shown in Figure 4.1(a) to 4.1(c). Figure 4.1(d) shows the computational domain taken for modeling fluid flow and heat transfer over a smooth wavy surface. The advantages of geometrical symmetry along the fin height have been taken into account while considering the computational domain. Half the fin height (h/2), fin spacing(s) and projected length (ie., peak to peak distance or valley to valley distance) called wavelength constitute the computational domain. The height, spacing, amplitude, wavelength and wavy tip radius of the wavy fin computational domain are 2 mm, 2 mm, 1.95 mm, 9.525 mm and 2 mm respectively. Similarly, the geometrical parameters of 24 different wavy fins geometries considered for the parametric studies are shown in Table 4.1.

![Figure 4.1](image)

(a) Wavy fin (b) Photographic view (c) and (d) dimensional notations (d) Computational domain

Figure 4.1 Wavy fin (a) Photographic view (b) and (c) dimensional notations (d) Computational domain
Table 4.1 Wavy fins used for parametric study

<table>
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<tr>
<th>Fin No</th>
<th>Fin Height h (mm)</th>
<th>Fin Spacing s (mm)</th>
<th>Fin Amplitude 2A (mm)</th>
<th>Fin Wavelength L (mm)</th>
<th>Curvature Radius R (mm)</th>
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4.1.2 CFD Approach

The CFD approach for both pressure drop and heat transfer prediction across the fins is explained in this section.

Initially the mesh finalization is done across the computational domain. Different mesh configurations starting with very coarse to very fine are taken at a particular Reynolds number and analyzed using Fluent. The grid independence test is carried out and a graph is drawn between the number of elements and the performance parameter (pressure drop) for the fin -1 as shown in Figure 4.2. It is seen from the figure that the minimum number of elements required for consistent results is 2, 20,000. Hence, further analysis is conducted with 2, 20,000 grid elements.

![Figure 4.2 Grid independency graph](image)

In the flow analysis, in order to overcome the entrance effect, the concept of periodic fully developed flow (Patankar et al 1977) is implemented. The friction factor is calculated from the pressure drop obtained
under quasi isothermal condition and using the basic equations given in the next section. The corresponding velocity profile is also obtained, which is used as an input for the thermal analysis. The same procedure is repeated for the range of Reynolds numbers from 100 to 15,000. This process requires 24 hrs to get the converged solutions for all range of Reynolds numbers for a single fin. This analysis is repeated for all the 25 wavy fin geometries.

In the thermal analysis, the “velocity inlet” (taken from flow analysis) and “outflow” boundary conditions are used at the inlet and outlet of the fin geometry, respectively. Constant wall temperature boundary condition is employed for walls. After the thermal analysis, post processing is done for temperatures and pressures at the inlet and outlet over the computational domain using mass weighted average. The pressure, temperature and velocity profiles are taken at the various sections of the fin for corresponding Reynolds number. This temperature difference between inlet and outlet of the core, in turn, is used for calculating \( j \) factor using the basic equation given below. The same procedure is repeated for the range of Reynolds numbers from 100 to 15,000. The same approach is used to characterize 18 different fin geometries. For all the cases, the value of the dimensionless distance \( Y^+ \) is always maintained less than 1.

**Equations:** The equations employed in evaluation of the Fanning friction factor \( (f) \) and Colburn factor \( (j) \) are given below.

i) The hydraulic diameter \( (D_h) \) of wavy fin is defined as

\[
D_h = \frac{4A_p}{P} = \frac{4(s-t)h}{2[(s-t)+h]} = \frac{2(s-t)h}{[(s-t)+h]} \tag{4.1}
\]
ii) Fanning friction factor \((f)\) is defined on the basis of an equivalent shear force in the flow per unit heat transfer area, and it is represented by the equation,

\[
f = \frac{1}{2\ell} \frac{\Delta P D_h}{\rho_{air} v^2}
\]  

(4.2)

iii) The dimensionless Colburn factor \((j)\) is represented by the equation,

\[
j = St \times Pr^{2/3} = \frac{D_h}{4\ell} \ln \left( \frac{T_i-T_w}{T_o-T_w} \right) Pr^{2/3}
\]  

(4.3)

4.1.3 Reliability of CFD Software Used

In order to ensure the reliability of CFD software (FLUENT), which is used in the present analysis, a rectangular channel has been analyzed for which exact solutions are well established. This configuration is selected, since the configuration used in the present analysis will converge to rectangular fin under some limiting conditions. For example, if the amplitude \((A) = 0\) or wavelength \((\lambda) = \infty\) in the wavy fin configuration, it becomes rectangular fin. Similarly in the offset fin considered in the next chapter, if the offset is made to zero, it becomes a rectangular fin.

**Rectangular Fin Analysis:** A rectangular fin (the same as a rectangular duct considering fin efficiency as 100%) has been analyzed using CFD by considering uniform wall temperature boundary condition. Most of the authors like Jeihai Zhang et al (2004), Metwally and Manglik (2004) and Manglik et al (2005) who did extensive numerical work on wavy fins followed the same boundary condition.
The dimensions of the fin considered in the analysis are $h = 10.2 \text{ mm}$; Fin density = 28 fins/inch and $t = 0.152 \text{ mm}$. The CFD results of rectangular fin are compared with the results of Shah and London (1978) and shown in Figure 4.3. CFD results are in well accordance with the analytical results given by Shah and London (1978) for the low Reynolds number region and the variations are found to be about 2% in $j$ and 9% in $f$ values. Hence, it is concluded that the FLUENT software may provide reliable results for the present analysis.
4.2 DEVELOPMENT OF CORRELATIONS FOR WAVY FIN

The dimensionless group considered for the correlation development and its effects on $f$ and $j$ are results are briefly discussed in the following sub sections.

4.2.1 Physical model

The geometrical features and computational domains of the three dimensional wavy fin are shown in Figure 4.1. To develop a correlation for wavy fins, 18 fins are considered for the analysis. The height, spacing, amplitude, wavelength and wavy tip radius of the wavy fin-1 computational domain are 25 mm, 1.16 mm, 1.56 mm, 7.8 mm and 2 mm respectively. Similarly, the geometrical parameters of 17 different wavy fins geometries considered for the generation of the correlations are shown in Table 4.2.

4.2.2 Numerical simulation

The mathematical model used for this CFD simulation is explained in the previous section 3.2. Similarly, the CFD approach adopted for this work and equations used for the $f$ and $j$ computations are also explained previously in the section 4.1.2.
Table 4.2 Dimensionless wavy fin parameters

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* The first number indicates fin height, W indicates wavy fin surface and the second number indicates FPI.

4.3 RESULTS AND DISCUSSIONS

The effect of wavy fin parameters on $f$ and $j$ and the development of correlations for $f$ and $j$ are discussed in this section.
4.3.1 Parametric studies

4.3.1.1 Influence of fin height on $f$ and $j$

Figure 4.4, it shows that as the fin height increases the $j$ and $f$ factors also increase significantly. This is due to the increase in cross-sectional area of fin channel, which provides space for development of more recirculation zones. The velocity, pressure and temperature contours are shown in Figures 4.5 and 4.6 for the minimum and maximum values of the fin height. From the figures, it is seen that the recirculation zones are more at the middle of the bottom valley and end of the peak regions. These recirculation zones are responsible for higher value of $f$ and $j$ values. However, the rate of increase is higher for $Re < 5000$ and lower for $Re > 5000$. 
Figure 4.4  Effect of fin height on $f$ and $j$ (a) $f$ vs Fin height (b) $j$ vs Fin height
Figure 4.5 Contours for wavy fin surface: h=2, s=2, 2A=1.95, L=9.525 and Re = 300. (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
Figure 4.6  Contours for wavy fin surface: h=25, s=2, 2A=1.95, L=9.525 and Re = 300. (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
4.3.1.2 Influence of fin spacing on $f$ and $j$

To study the effect of fin spacing, it is varied from 2 to 4.2 mm ($2 \leq s \leq 4.2$), keeping all other parameters as constants. Five different spacing of wavy fins are analyzed and the results are plotted. Figure 4.7 shows that as the fin spacing increases, the friction factor increases up to $s = 3.2$ mm and then decreases. This trend is observed at $Re < 1000$. For $Re > 1000$, the friction factor always increases with fin spacing.

In the case of $j$ factor, it decreases with increase in fin spacing. However, the rate of decrease is higher at low Reynolds number. The spacing effect plays a predominant role at the low Reynolds number due to viscous force domination, which suppress the swirl motion and hence, the heat transfer is decreased. The velocity, pressure and temperature contours for maximum and minimum spacing cases are shown in Figure 4.8 and Figure 4.9 respectively.
Figure 4.7  Effect of fin Spacing on $f$ and $j$ (a) $f$ vs s (b) $j$ vs s
Figure 4.8  Contours for wavy fin surface: h=2, s=2, 2A=1.95, L=9.525 and Re = 300 (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
Figure 4.9  Contours for wavy fin surface: h=2, s=4.2, 2A=1.95, L=9.525 and Re = 300 (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
4.3.1.3 Influence of fin amplitude on $f$ and $j$

In order to study the effect of fin amplitude, it is varied from 1.4 to 2.2 mm ($1.4 \leq 2A \leq 2.2$), keeping all other parameters as constants. As the fin amplitude increases the $f$ and $j$ factors increases significantly as shown in Figure 4.10.

As the fin amplitude increases, the peak to valley distance increases. This creates more resistance to flow, which in turn leads to higher pressure drop. In the case of heat transfer, the increase in amplitude creates more turbulence and promotes better mixing, which in turn leads to higher heat transfer. Figure 4.11 and Figure 4.12 show the velocity, pressure and temperature contours for minimum and maximum amplitude cases respectively. From the velocity contours, it is observed that more recirculation zones are found in maximum amplitude case.
Figure 4.10 Effect of fin Amplitude on $f$ and $j$ (a) $f$ vs 2A (b) $j$ vs 2A
Figure 4.11  Contours for wavy fin surface: $h=2$, $s=2$, $2A=1.4$, $L=9.525$ and $Re = 300$ (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
Figure 4.12  Contours for wavy fin surface: $h=2$, $s=2$, $2A=2.2$, $L=9.525$ and $Re = 300$ (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
4.3.1.4 Influence of wavelength on $f$ and $j$

The fin wavelength is varied from 7.5 to 12 mm keeping all other parameters as constants to study the wavelength effects. Five different wavelengths of wavy fins are analyzed in the similar manner and the results are plotted in Figure 4.13. As the wavelength increases, the $f$ and $j$ factors decrease significantly for all Reynolds numbers.

If the wavelength increases, wavy channel becomes more or less straight and looks like plain fin. Therefore the flow becomes smooth and resistance to flow in the wavy channel is reduced. Because of less resistance to flow the pressure drop as well as friction factor is reduced. In the case of $j$ factor, for the higher value of wavelength, the formation of recirculation zones at peaks and valleys are reduced drastically, which in turn leads to the poor mixing. Hence, the heat transfer rate is reduced. This can be seen clearly from Figure 4.14 and Figure 4.15.

The variation of $f$ and $j$ is higher at lower Reynolds number cases and vice versa.
Figure 4.13 Effect of fin Wavelength on f and j (a) f vs L (b) j vs L
Figure 4.14  Contours for wavy fin surface: h=2, s=2, 2A=1.95, L=7.5, and Re = 300 (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
Figure 4.15 Contours for wavy fin surface: $h=2$, $s=2$, $2A=1.95$, $L=12$ and $Re = 300$ (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
4.3.1.5 Influence of wavy tip radius on $f$ and $j$

To study the effect of wavy fin tip radius, it is varied 1 to 3 mm ($1 \leq R \leq 3$), keeping all other parameters as constants. Five different wavy tip radii of wavy fins are modeled and analyzed using CFD in the similar manner. The results are plotted in Figure 4.16. From the figure, it is seen clearly that in the case of $f$ factor, it is almost constant with increase in wavy tip radius up to $R = 2$mm. Beyond $R= 2$mm, the friction factor increases with wavy tip radius.

In the case of $j$ factor, for all the values of wavy tip radius, the $j$ factor becomes more or less constants. The velocity, pressure and temperature contours for maximum and minimum wavy tip radius are shown in Figure 4.17 and Figure 4.18. Further, it is concluded that wavy tip radius as noticeable influence only on $f$ factor alone, not on $j$ factor.
Figure 4.16 Effect of wavy tip radius on $f$ and $j$ (a) $f$ vs $R$ (b) $j$ vs $R$
Figure 4.17  Contours for wavy fin surface: $h=2$, $s=2$, $2A=1.95$, $L=9.525$
$R=1$ and $Re = 300$ (a) Velocity (m/s) (b) Pressure (Pascal) and (c) Temperature (K)
Figure 4.18  Contours for wavy fin surface: $h=2$, $s=2$, $2A=1.95$, $L=9.525$  
$R=3$ and $Re = 300$  (a) Velocity (m/s) (b) Pressure (Pascal)  
and (c) Temperature (K)
4.3.2 Correlations for wavy fin

4.3.2.1 Effect of dimensionless geometrical parameters on $f$ and $j$

Numerical results for isothermal Fanning friction factor, Colburn factor and their variation with flow Reynolds number in different wavy plate channel for the range of parameters $(1.176 \leq \alpha \leq 20.8, 0.636 \leq \beta \leq 2.6$ and $3.85 \leq \gamma \leq 7.69)$ are presented.

Figures 4.19(a) and 4.19(b) show the variation of performance factors $f$ and $j$ with respect to flow aspect ratio ($\alpha$) for various Reynolds number. It is observed that the fin, which is having highest aspect ratio, leads to highest values of friction and Colburn factors throughout the Reynolds number range. With higher aspect ratio (either by increasing height or decreasing fin spacing), the total secondary heat transfer surface area increases and this in turn leads to higher heat transfer. The increase of the friction factor is justified by the increase in the hydraulic diameter. Figure 4.20 shows the variation in the velocity contours for $h/s = 20.8$ and $h/s = 1.176$. It is observed that more recirculation zones appeared near peak and valley regions for $h/s = 20.8$. These recirculation zones decrease the flow velocity. However the flow velocity is more for $h/s = 1.176$, and it diminishes the recirculation zones. From these observations, it is found that more recirculation zones enhance the heat transfer rate and pressure drop.
Figure 4.19 Effect of fin height to fin spacing (h/s) ratio on wavy fin surface performance: (a) Friction factor $f$ (b) Colburn factor $j$
Figure 4.20  Velocity contour for wavy fin surface: $2A/s = 1.3$, L/2A= 5 and Re = 5000.  (a) h/s = 20.8  (b) h/s = 1.176
Figures 4.21(a) and 4.21(b) show the dependence of $f$ and $j$ on the wave-amplitude-to-fin-spacing ratio ($\beta$). From these figures, it is found that as wave-amplitude-to-fin-spacing ratio decreases, $f$ and $j$ factors increase. The increase in plate spacing promotes vortex to grow in the trough region and envelop much of the core region, thereby promoting flow mixing and momentum transport. Further, as plate separation decreases, viscous effects suppress swirl and undisturbed streamlines flows prevail. Hence the heat and momentum transfer decreases. This phenomenon is clearly seen from Figure 4.22.

Figures 4.23(a) and 4.23(b) shows the variation of $f$ and $j$ with change of wavelength-to-wave-amplitude ($\gamma$). Both $f$ and $j$ increases with decrease of $\gamma$ for all range of Reynolds numbers. The extent of lateral swirl flow coverage area increases with decrease in $\gamma$ value (ie decreasing wavelength or increasing amplitude) and hence the friction factor increases. With the development of trough region recirculation, there is a thinning of the boundary layer and the flow field exhibits regions of thermal mixing. This in turn leads to higher heat transfer. The stream traces for highest and lowest values of $\gamma$ are shown in Figure 4.24.
Figure 4.21 Effect of wave amplitude to fin spacing (2A/s) ratio on wavy fin surface performance: (a) Friction factor \( f \) (b) Colburn factor \( j \)
Figure 4.22  Stream traces for wavy fin surface: $h/s = 11$, $L/2A = 5$ and $Re = 5000$. (a) $2A/s = 2.6$  (b) $2A/s = 0.63$
Figure 4.23 Effect of wavelength to wave amplitude (L/2A) ratio on wavy fin surface performance: (a) Friction factor $f$ (b) Colburn factor $j$
Figure 4.24 Stream traces for wavy fin surface: h/s = 11, 2A/s = 1.3, Re = 5000 (a) L/2A = 7.69 (b) L/2A = 3.84
Figure 4.25 Stream traces for wavy fin surface: h/s = 11, 2A/s = 0.636 and L/2A= 5 (a) Re = 500 (b) Re = 5000
In order to show the effect of Reynolds number, the stream traces are drawn in Figure 4.25 for one specific configuration (Fin 1) for two different Reynolds numbers 500 and 5000. From these figures, it is observed that more number of recirculation zones and destabilization of boundary layer exist at low Re = 500 when compared to high Re = 5000. This is the reason for increasing $f$ and $j$ values with decreasing Reynolds numbers as expected.

**Correlation for $f$ and $j$ factors:** From the foregoing study, it is evident that $f$ and $j$ are functionally related to Re, $\alpha$, $\beta$, $\gamma$ and they are correlated in the present investigation as below.

$$f = C \text{Re}^{a_1} (\alpha)^{a_2} (\beta)^{a_3} (\gamma)^{a_4} \quad \text{and}$$

$$j = D \text{Re}^{b_1} (\alpha)^{b_2} (\beta)^{b_3} (\gamma)^{b_4}$$

(4.4)

The use of these power law expression is justified because variations in $f$ and $j$ with Re, $\alpha$, $\beta$ and $\delta$ follow constant slope log-linear lines in both deep laminar and turbulent regions. A multivariable regression analysis conducted for laminar and turbulent regions yielded the following results.

**For laminar Range** ($100 \leq \text{Re} \leq 800$)

$$f = 9.827 \text{Re}^{-0.705} (h/s)^{0.322} (2A/s)^{-0.394} (L/2A)^{-0.603}$$

(4.6)

$$j = 2.348 \text{Re}^{-0.786} (h/s)^{0.312} (2A/s)^{-0.192} (L/2A)^{-0.432}$$

(4.7)

**For turbulent Range** ($1000 \leq \text{Re} \leq 15000$)

$$f = 10.628 \text{Re}^{-0.359} (h/s)^{0.264} (2A/s)^{-0.848} (L/2A)^{-1.931}$$

(4.8)

$$j = 0.242 \text{Re}^{-0.375} (h/s)^{0.235} (2A/s)^{-0.288} (L/2A)^{-0.553}$$

(4.9)
Figure 4.26 Parity plots showing $f$ and $j$ factors obtained from the correlations and numerical models for laminar and turbulent regimes: (a) $f_{\text{correlated}}$ vs $f_{\text{numerical}}$, and (b) $j_{\text{correlated}}$ vs $j_{\text{numerical}}$
While generating the correlations the curvature radius of 2 mm has been taken for CFD analysis in both laminar and turbulent regimes. Error analysis is carried out to evaluate the efficacy of these correlations. Figures 4.26(a) and 4.26(b) show the $f_{\text{correlated}}$ vs $f_{\text{numerical}}$ and $j_{\text{correlated}}$ vs $j_{\text{numerical}}$ respectively. It is observed that most of the data points in the Figures lie near the 45° - line. The maximum absolute deviation is found to be less than 20% for $f$ factor and 10% for $j$ factor. Therefore these correlations can be conveniently adopted to predict the Fanning friction and Colburn factors for the wavy fins in both laminar and turbulent regimes.

### 4.3.2.2 Comparison of the present correlation with literature results

In order to explore the validity of Wavy fin correlations, comparison has been made with the experimental results available in the literature. These comparisons are shown in Figures 4.27(a) and 4.27(b). The line with the solid square symbol represents the values of $f$ and $j$ generated using the present correlations for the parameter prescribed by Kays and London (11.44-3/8W) fin. The $f$ and $j$ values of present correlation under predict the Kays and London (1984) experimental data by 15% and 20% in the laminar region, 8% and 25% in the turbulent region respectively. This difference may be due to manufacturing irregularities such as burred edges, bonding imperfection and other CFD assumptions.
Figure 4.27  Comparison of (a) $f$ and (b) $j$ factors experimental data with correlations results for Fin 11.5-3/8W Kays and London (1984)