Development of heat transfer coefficient and friction factor correlations for offset fins using CFD

Article *in* International Journal of Numerical Methods for Heat &amp Fluid Flow · November 2011

DOI: 10.1108/09615531111177732

2 authors:

Ranganayakulu Chennu
Birla Institute of Technology and Science Pilani

Pallavi Paturu
Indian Institute of Information Technology, Design and Manufacturin…

The current issue and full text archive of this journal is available at
www.emeraldinsight.com/0961-5539.htm

Development of heat transfer coefficient and friction factor correlations for offset fins using CFD

Ranganayakulu Chennu
Aeronautical Development Agency, Bangalore, India, and
Pallavi Paturu
National Institute of Technology, Warangal, India

Abstract

Purpose – In aerospace applications, due to the severe limitations on the weight and space envelope, it is mandatory to use high performance compact heat exchangers (CHEs) for enhancing the heat transfer rate. The most popularly used ones in CHEs are the plain fins, offset strip fins (OSFs), louvered fins and wavy fins. Amongst these fin types, wavy and offset fins assume a lot of importance due to their enhanced thermo-hydraulic performance. The purpose of this paper is to investigate the influence of geometrical fin parameters, in addition to Reynolds number, on the thermo-hydraulic performance of OSFs.

Design/methodology/approach – A computational fluid dynamics approach is used to conduct a number of numerical experiments for determination of thermo-hydraulic performance of OSFs considering the various geometrical parameters, which are generally used in the aerospace industry. These investigations include the study of flow pattern for laminar, transition and turbulent regions. Studies are conducted with different fin geometries and comparisons are made with available data in open literature. Finally, the generalized correlations are developed for OSFs taking all geometrical parameters into account for the entire range of operations of the aerospace industry covering laminar, transition and turbulent regions. In addition, the effects of various geometrical parameters are presented as parametric studies.

Findings – Thermo-hydraulic design of CHEs is strongly dependent upon the predicted/measured dimensionless performance (Colburn factor "j" and Fanning friction "f" vs Reynolds number Re) of heat transfer surfaces. Several types of OSFs used in the compact plate-fin heat exchangers are analyzed numerically.

Research limitations/implications – The present numerical analysis is carried out for "air" media and hence these results may not be accurate for other fluids with large variations of Prandtl numbers.

Practical implications – In open literature, these fins are generally evaluated as a function of Reynolds number experimentally, which are expensive. However, their performance will also depend to some extent on geometrical parameters such as fin thickness, fin spacing, offset fin length and fin height.

Originality/value – This numerical estimation can reduce the number of tests/experiments to a minimum for similar applications.

Keywords Aerospace industry, Heat transfer, Compact heat exchangers, Offset strip fins, Numerical analysis, Thermal-hydraulic performance, Friction factor "f", Colburn factor "j"

Paper type Research paper

The authors wish to acknowledge the Aeronautical Development Agency for allowing publication of the paper.

International Journal of Numerical Methods for Heat & Fluid Flow
Vol. 21 No. 8, 2011
pp. 935-951

0961-5539
DOI 10.1108/09615531111177732

Nomenclature

A = exchanger total heat transfer on one side (m^2)
A_c = exchanger minimum free-flow area (m^2)
A_f = exchanger total fin area on one side (m^2)
A_{fr} = exchanger total frontal area (m^2)
C = flow-stream capacity rate (kJ/K)
C_p = specific heat at constant pressure (kJ/kg K)
D_h = hydraulic diameter ($D_h = [2(s - t)h]/[(s + h) + ht/l]$) (m)
f = fanning friction factor dimensionless
FPI = fins per inch
G = mass flow rate per unit area (kg/s m^2)
h = fin height (mm)
j = Colburn factor ($StPr^{2/3}$) (dimensionless)
k = thermal conductivity (W/m K)
l = offset strip fin length (mm)
l^* = ratio of ($l/4r_h$) (dimensionless)
Nu = Nusselt number (hD_h/k) (dimensionless)
Pr = Prandtl number ($\mu Cp/k$) (dimensionless)
ΔP = pressure drop (Pa)
Q = heat transfer rate (W)
Re = Reynolds number (dimensionless)
Re^* = Re at point of transition
Re_D = Re at transitional range
s = fin spacing (mm)
S_t = Stanton number (dimensionless)
T_i = inlet temperature (°C)
T_o = outlet temperature (°C)
T_w = wall temperature (°C)
t = fin thickness (mm)
U = overall heat transfer coefficient
u = velocity component in x-direction
v = velocity component in y-direction
W = mass flow rate (kg/s)
w = velocity component in z-direction
x/D = length/hydraulic diameter
$y^* y^+$ = wall functions (dimensionless)

Greek symbols

α^* = ratio of $(h - t)/(s - t)$ (dimensionless)
α = ratio of (s/h) (dimensionless)
δ = ratio of (t/l) (dimensionless)
δ^* = ratio of ($t/4r_h$) (dimensionless)
μ = dynamic viscosity, (Ns/m^2)
γ = ratio of (t/s) (dimensionless)
ρ = density of the air (kg/m^3)
τ_w = wall shear stress (N/m^2)
φ = generalized transport variable
k = turbulent kinetic energy (m^2/s^2)
ε = turbulence dissipation rate (m^2/s^3)
υ = specific volume (m^3/kg)
? = effective diffusivity (m^2/s^2)

1. Introduction

Compact heat exchangers (CHEs) are widely used in aerospace, automobile and cryogenic industries due to its compactness (i.e. high heat transfer surface area to volume ratio) for desired thermal performance, resulting in reduced space, weight, support structure, footprint, energy requirement and cost. Depending on the application, various types of augmented heat transfer surfaces such as wavy fins, offset strip fins (OSFs), louvered fins and perforated fins are used in CHEs.

Among the heat transfer surfaces, OSFs, louvered fins and wavy fins are widely used. They have a high degree of surface compactness and substantial heat transfer enhancement obtained as a result of the periodic starting and development of laminar boundary layers over interrupted channels formed by the fins and their dissipation in the fin wakes. There is, of course, an associated increase in the pressure drop due to increased friction and form drag contribution from the finite thickness of the interrupted fins. Thermo-hydraulic design of a CHE is strongly dependent upon the performance of heat transfer surfaces (in terms of Colburn factor j and Fanning friction factor f vs Reynolds number Re characteristics). This paper focuses on generation of Colburn factor j and Fanning friction factor f for OSFs. The isometric view of an offset fin is shown in Figure 1(a).

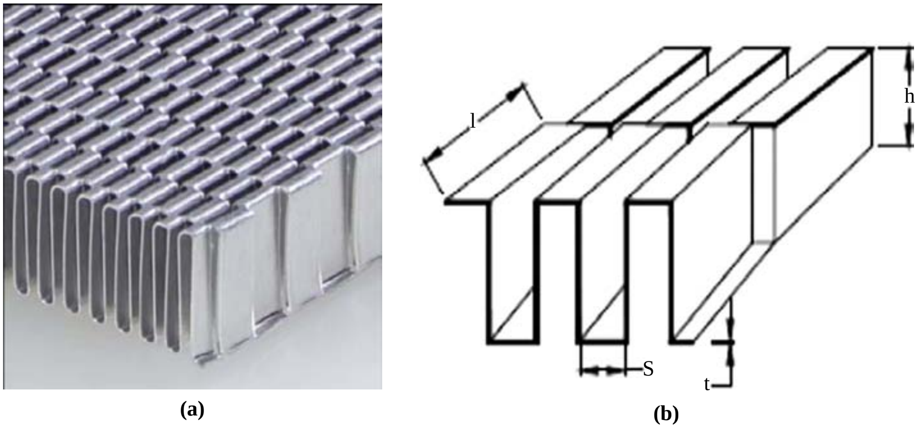

Figure 1. Schematics of fin geometry

Notes: (a) Offset fin – isometric view and (b) offset fin dimensional notations

The geometrical features of the three-dimensional offset fin flow channel are described by the fin height(h), fin spacing(s), fin thickness(t), offset strip length(l) as shown in Figure 1(b). The dimensionless representations of these variables are given by the flow cross-section aspect ratio ($\alpha = s/h$), $\gamma = t/s$ and $\delta = t/l$. The performance factors are evaluated for this offset fin using fluent. The performance of the offset fin is studied by evaluating "*j*" and "*f*" factors. In the fin designation, the first number indicates the fin height, the second number indicates the fin density (fins/in.) and the third number indicates the fin thickness as 1.5S-20-0.076.

London and Shah (1968) discussed performance of strip-fin core due to the following four non-dimensional geometrical parameters: dimensionless fin thickness δ^*, aspect ratio of flow passage in lone fin pitch α^*, fin surface area to total surface area on the fin side A_f/A, and dimensionless strip length l^* of OSF geometry. Higher α^*, δ^*, A_f/A ratios tend to make higher *j* and *f* factors, and when l^* is higher, both the *j* and *f* factors will tend to be lower. Because of smaller hydraulic radius, the non-dimensional roughness characterization influences much in *f* vs Re characteristics. Kays and London (1984) made one of the attempts at analytical modeling of the heat transfer and friction loss in OSFs and proposed a laminar boundary layer solution that includes the finite drag contribution of the blunt fin edges. However, his correlations do not take into account the fin parameters indicated in this paper.

Joshi and Webb (1987) developed elaborate analytical models to predict *f* and *j*. They incorporated the heat transfer from the fin ends, the form drag due to the finite fin thickness, and heat transfer and friction from the parting plates. An attempt was also made to model the effects of fin burrs and roughness with the semi-empirical approach. However, these models are quite cumbersome and in cognizance of the need for an engineer to use correlation, they reevaluated the empirical equations of Weiting (1975). Weiting (1975) proposed empirical correlations for *j* and *f* performing experiments on 22 rectangular OSF configurations over two Reynolds number ranges: $Re_D \leq 1{,}000$ which is primarily laminar and $Re_D \geq 2{,}000$ which is primarily turbulent. The correlation is determined using weighted least squares slope. The experimental data used have been obtained from experiments using air as working fluid and hence the applicability of the correlations to fluids outside the gas Prandtl number range may be open to question. In addition, approximately 80 percent of the data are for fins with $1.5 \leq x/D \leq 2.5$; consequently, the exponents of x/D are heavily weighted to the data of these surfaces. Similarly the j factor was correlated over the range Re_D 370-32,000 and hence the correlations can be extrapolated for limited range and serious error may occur if the correlations are employed out of the parameter range. The entire experimental data are predicted within ± 85 percent and few data with discrepancy of 40 percent. The correlations also allow the effect of fin length, height, thickness, spacing and hydraulic diameter on the performance to be assessed.

Among the very few numerical analysis for generation of heat transfer and pressure drop characteristics of an OSF heat exchanger is a study by Bhowmik and Lee (2009). They recalculated the Nusselt number using the correlated j and the effect of Prandtl numbers were studied. They also indicated that an air model should not be used to predict the fluid flow and heat transfer of OSF heat exchangers operated with liquid media. Mochizuki *et al.* (1987) developed their own correlations based on seven aluminum test cores of plate fin type heat exchangers with strip fins and selecting the fin spacing and strip length systematically. Experiments were carried out for each core using air as the test fluid and condensing steam as the other medium.

For generation of j and f vs Re data numerically, the entry effects into the fin play a predominant role. In order to overcome this difficulty, Patankar *et al.* (1977) introduced the concept of periodic fully developed flow and heat transfer. The underlying concept was that for a constant property flow in a duct of constant cross-section, the velocity and temperature distributions become independent of the stream-wise coordinate at sufficiently large distances from the inlet. Manglik and Bergles (1995) investigated most of the correlations available in open literature, studied the concepts of each correlation and generated their own correlations. They took 18 different cores given in standard literature. Various fin parameters are taken into account while generating the correlations. Equations that describe the asymptotic behavior of the data in the deep laminar and fully turbulent regions have been developed.

Sheik Ismail *et al.* (2010) focuses on the research and development of compact offset and wavy plate-fin heat exchangers. The review is summarized under three major sections. They are offset fin characteristics, wavy fin characteristics and non-uniformity of the inlet fluid flow. The various research aspects relating to internal single-phase flow studied in offset and wavy fins by the researchers are compared and summarized. Further, the works done on the non-uniformity of this fluid flow at the inlet of the CHEs are addressed and the methods available to minimize these effects are compared. Sheik Ismail *et al.* (2009) analyzed numerically three OSF and 16 wavy fin geometries used in the compact plate-fin heat exchangers. They also analyzed for quantification of flow maldistribution effects with real and ideal cases by providing suitable baffle plates for improvement in flow distribution using computational fluid dynamics (CFD). Similar numerical analysis was carried out by Ranganayakulu *et al.* (2008) for optimization of wavy fin parameters and provided j and f correlations for laminar and turbulent regions.

2. Mathematical model

Following are some of the assumptions made in the CFD simulation:

- the flow is stable in the computational domain;

- the fluid flows meet the Boussinesq assumption; and
- the fluid in the domain is incompressible.

In this work, the CFD package FLUENT is used for simulation. In FLUENT, the conservation equations of mass, momentum and energy are solved using the finite volume method. There are several turbulence models available in the code. The turbulent flow is calculated by the semi-implicit SIMPLER as mentioned in Versteeg and Malalasekera (1995). Algorithm method in the velocity and pressure conjugated problem, and a first-order upwind differential scheme is applied for the approximation of the convection terms.

A standard *k*-ε model as given in Versteeg and Malalasekera (1995) with enhanced wall treatment is used to predict turbulent flow in the fin geometry. The Reynolds transport equations can be written in a generalized form as given by Anderson (1995) and Patankar (1980):

$$\mathrm{div}(\rho u \varphi) = \mathrm{div}(\Gamma' \, grad\ \varphi) + S_\varphi \tag{1}$$

where φ stands for a generalized transport variable, which is used for all conserved variables in a fluid flow problem, including, mass, momentum and the turbulence variables k and ε. Γ represents the effective diffusivity (sum of the eddy diffusivity and the molecular diffusivity). S_φ is the source term for the respective dependent variable. The solution of the above set of equations is applied to the prediction of velocity and turbulence levels throughout the domain. The convergent criterion is specified to absolute residuals ($\leq 1.0 \times 10^{-4}$).

3. Problem description

The geometrical features of the three-dimensional offset fin flow channel are described by the fin height (h), fin spacing (s), fin thickness (t) and offset strip length (l). The dimensionless representations of these variables are given by the flow cross-section aspect ratio ($\alpha = s/h$), $\gamma = t/s$ and $\delta = t/l$. These dimensional notations are shown in Figure 1(b).

The performance of a plate fin surface is not uniquely determined by the hydraulic diameter. Other geometric parameters such as fin height (h), fin spacing (s), fin thickness (t), and OSF length (l) plays significant role as indicated by London and Shah (1968). It will be prohibitively expensive and time consuming to fabricate heat exchanger cores and conduct experiments over reasonable ranges of all the geometric variables. In contrast, it is relatively easy and cost-effective to carry out parametric study through numerical simulation and derive acceptable correlations for use in industry.

Several authors have contributed significantly in some or the other way to develop a correlation that can take into account all the possible effects in order to determine j and f, but none of them had the same conclusions. In fact plenty of research work is carried out, yet there was no unique correlation developed by any one that can be relied upon in order to determine j and f especially for the OSFs. So, this work mainly focused to develop good correlations for offset fins taking all the necessary parametric effects.

The optimum OSF surfaces are taken for parametric study. A total of 15 different fin surfaces are modeled and CFD analysis is carried out for developing correlations. Numerical procedure has been used to find out j and f factors for the offset fin surfaces with in the ranges of dimensionless parameters of offset fin surfaces. The range of dimensionless geometric parameters and Reynolds number are taken as follows:

$$0.254 \leq \frac{s}{h} \leq 1.693 \qquad (2)$$

$$0.1 \leq \frac{t}{s} \leq 0.2 \qquad (3)$$

$$0.023 \leq \frac{t}{l} \leq 0.0714 \qquad (4)$$

$$300 \leq \mathrm{Re} \leq 500 \quad \text{for Laminar region} \qquad (5)$$

$$1,000 \leq \mathrm{Re} \leq 15,000 \quad \text{for Turbulent region} \qquad (6)$$

The performance factors are evaluated for different types offset fins using FLUENT. The performance of offset fins are presented for both laminar and turbulent regions of Re vs "*j*" and "*f*" factors.

4. CFD approach

A CFD approach is used to conduct a number of numerical experiments to determine the thermo-hydraulic parameters with a view to arrive at the optimum parameters for a offset fin. The computational domain of offset fin model shown Figure 2 is analyzed using CFD simulation tool FLUENT. The CFD analysis is carried out for different geometric parameters and Reynolds numbers using FLUENT 6.2 in Red Hat Linux OS/HP xw8000 workstation with 2 GB RAM. The analysis is carried out in two phases. In first phase, an offset fin is taken and characterized for *f*-values over a range of Reynolds number. In second phase, the *j*-value is determined for the same range by switching on the energy equation. The mass flow rates are determined for a range of Reynolds number from 300 to 15,000. In order to overcome the entrance effect, the concept of periodic fully developed

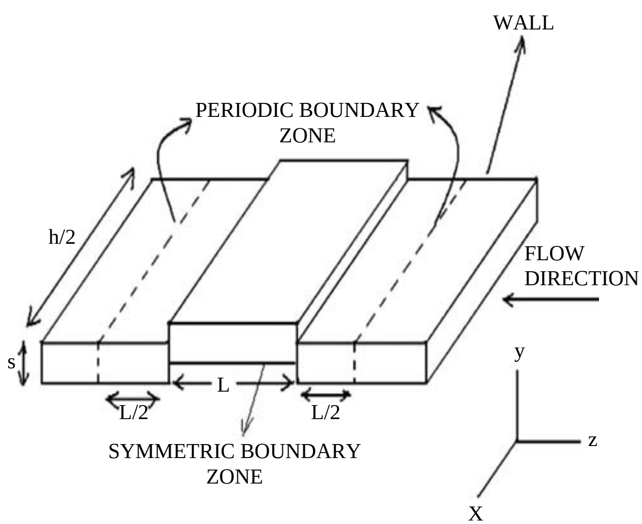

Figure 2.
Computational domain for an offset fin

flow as suggested by Patankar *et al.* (1977) is implemented for flow analysis. After the analysis, the pressure drop for unit length is one of the outputs, and that is multiplied by the actual length to get the total pressure drop for corresponding fins. From the pressure drop, friction factor is calculated as per Appendix A of Kays and London (1984). Finally, the corresponding two-dimensional fully developed velocity profile is listed out. Similarly, the same procedure is repeated for the range of Reynolds numbers from 300 to 15,000 in order to draw the *f* vs *Re* curves.

The mesh finalization is done before analyzing for the "*j*" and "*f*" factors. Different mesh configurations starting with very coarse to very fine are taken at a particular Reynolds number and analyzed using FLUENT. The grid independency curve is drawn between number of elements and the pressure drop parameters. Then the mesh size is determined based upon the curve where the slope is almost zero as shown in Figure 3.

In second phase, the "velocity inlet" and "outflow" boundary conditions are used at the inlet and outlet of the fin geometry, respectively. The two-dimensional fully developed velocity profile, which is taken from first-phase analysis (pressure drop analysis), is used in the "velocity inlet" boundary condition.

The assumption of 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 using mass weighted average, and pressure, temperature and velocity profiles are taken at the various sections of the fins for corresponding Reynolds numbers. This temperature difference between inlet and outlet of the fin, in turn, is used for calculating j factor using Kays and London (1984) methodology. Similarly, the same procedure is repeated for the range of Reynolds numbers 1,000-15,000 for turbulent flow and from 300 to 500 for laminar flow in order to draw the *j* vs *Re* characteristic curves. The actual mass flow rate is used as the boundary condition. No-slip boundary condition is used for walls. The actual computation time taken for solving first phase using three-dimensional segregated *k*-ε turbulent model with

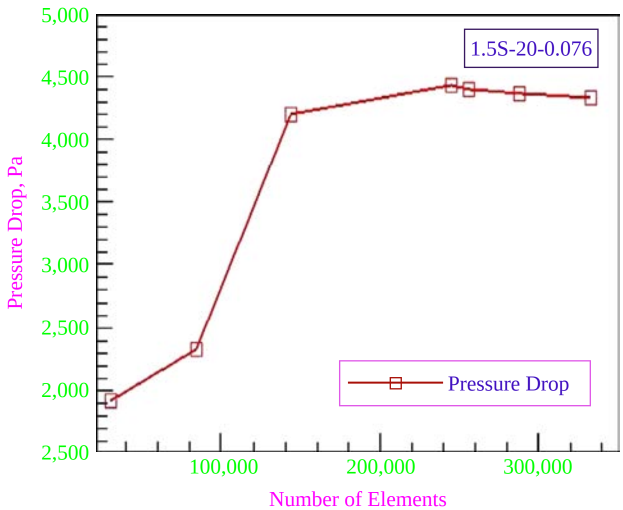

Figure 3.
Grid independency graph

enhanced wall treatment is 30 min. The time taken for solving second phase using the same model is 20 min. The "*j*" and "*f*" data are generated as stated above.

5. Validation

Offset fin channel has been analyzed with CFD and compared with experimental data. The results are further compared with open literature. Offset fin channel has been analyzed using CFD by considering uniform wall temperature boundary condition. For the validation of the numerical analysis conducted in the present study, two of the offset fins have been analyzed for the following cases:

- 2.49 mm fin height, 19.86 fins/in. (0.781 fins/mm) fin density and 0.102 fin thickness (fin: 2.49S-19.86-0.102); and
- 5.21 mm fin height, 19.82 fins/in. (0.78 fins/mm) fin density and 0.102 mm fin thickness (fin: 5.21S-19.82-0.102).

In addition, a grid independence test is carried out for the same fin and graph is plotted as the number of elements verses pressure drop as shown in Figure 3. This figure shows that after 256,000 cells there is not much variation in the pressure drop.

The CFD results of offset fins (2.49S-19.86-0.102 and 5.21S-19.82-0.102) are compared with Kays and London (1984) as shown in Figures 4 and 5, respectively. The CFD results are in good agreement with the analytical results given by Kays and London (1984) for the low Reynolds number region. The variations are found to be 4-16 percent in *f* and 8-14 percent in *j*-values. Figure 6 shows the comparison of FLUENT results with the literature correlations (Kays and London, 1984; Joshi and Webb, 1987; Weiting, 1975; Mochizuki *et al.*, 1987; Manglik and Bergles, 1995). It is evident for *f* and *j* factors that Wieting (1975) correlations are close to FLUENT results for high and low Reynolds numbers region. The *f* factor is also close to FLUENT with Mochizuki *et al.* (1987) for low Reynolds number and *j* factor is also close to FLUENT with Mochizuki *et al.* (1987) for high Reynolds number. However, some variations are observed in some region of Reynolds numbers when compared with FLUENT data. Giving exact reasons for

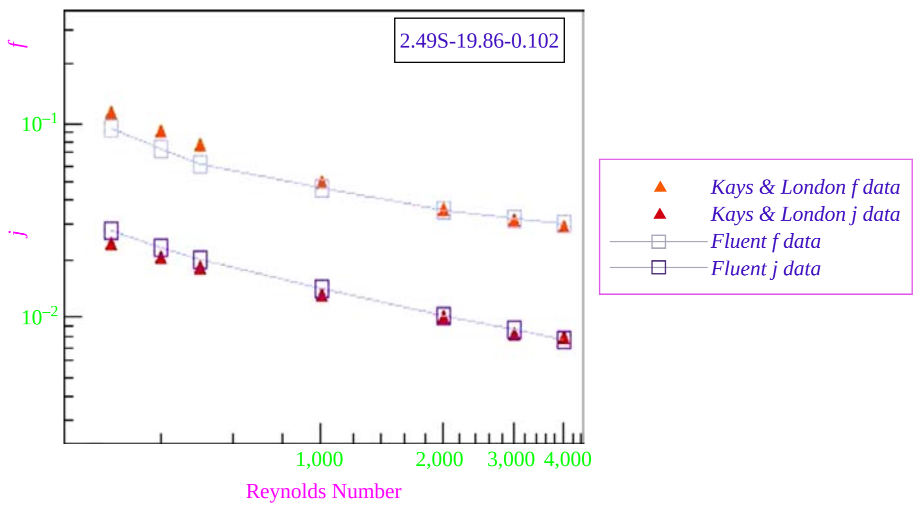

Figure 4.
Validation with the experimental results for the fin 2.49S-19.86-0.102

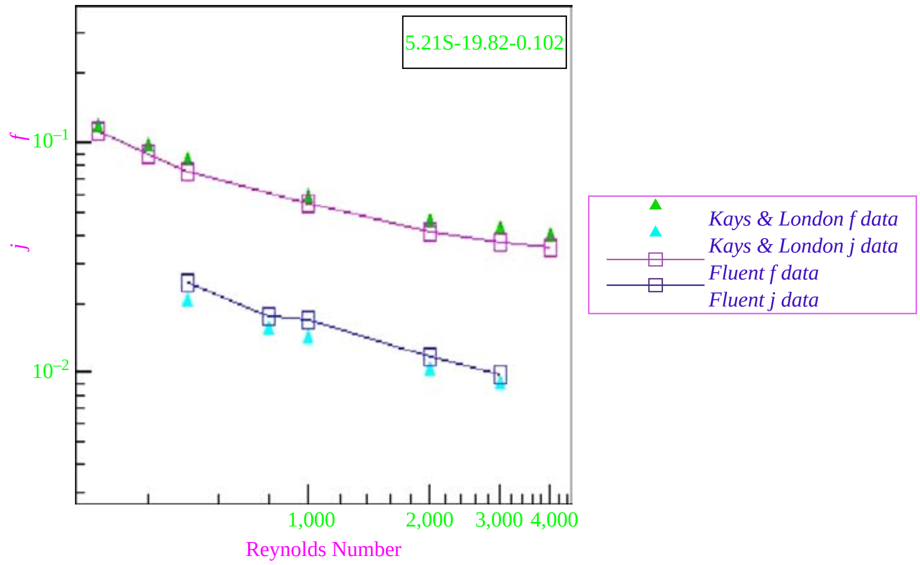

Figure 5.
Validation with the experimental results for the fin 5.21S-19.82-0.102

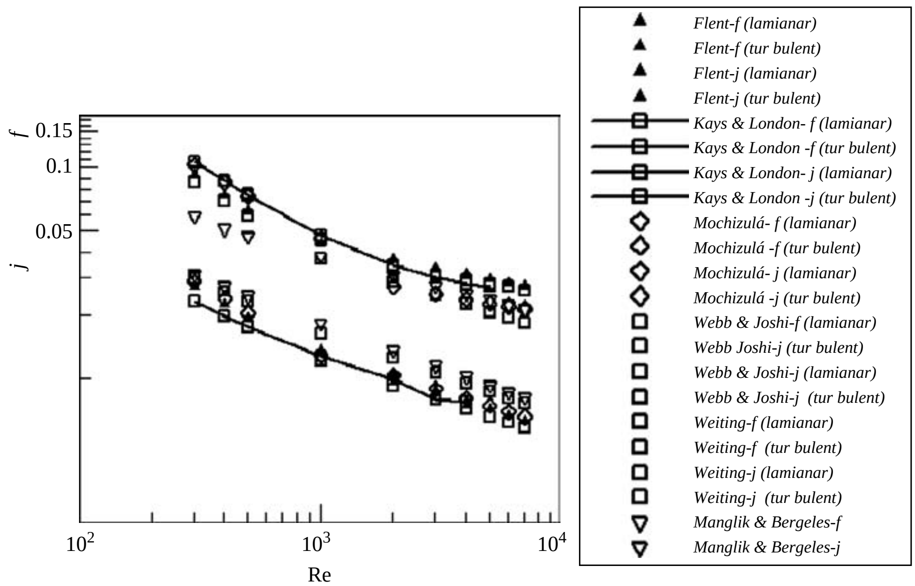

Figure 6.
Validation with the open literature for the fin 2.49S-19.86-0.102

variations of these factors may not be possible due to involvement of so many parameters such as manufacturing aspects testing conditions, etc.

6. Results and discussion

The CFD analysis for offset fin surfaces is carried out using FLUENT with boundary conditions and respective mass flow rates for various Reynolds numbers. From these graphs (Figures 4-6), it is observed that the *j* and *f* vs *Re* curves of OSFs follow the same trends as Kays and London (1984) experimental results. The effects of variation of dimensionless geometrical parameters on offset fin performance are presented below. The response of velocity pressure and temperature fields to changes in geometric parameters and Reynolds number is clearly manifested. Figures 7-10 show the role of geometric parameters *s/h, t/s, t/l* vs *f* and parameters *s/h, t/s, t/l* vs *j* in determining the heat transfer and flow friction performance of offset fin surfaces. The individual effects of geometrical parameters are explained in the following paragraphs.

6.1 Effect of s/h ratio on j

The Colburn *j* factors are plotted against the fin spacing (s) to fin height (h) ratio as shown in Figure 7. The *j* factor decreases as *Re* number increases as expected. The rate

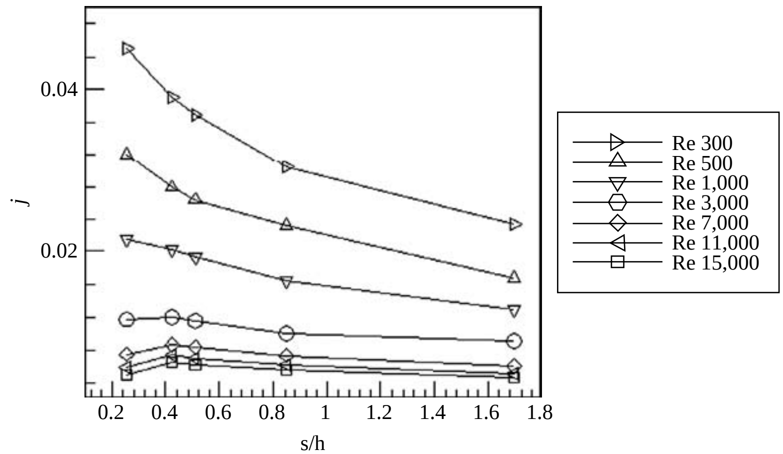

Figure 7.
Dimensionless parameter *s/h* ratio vs *j*

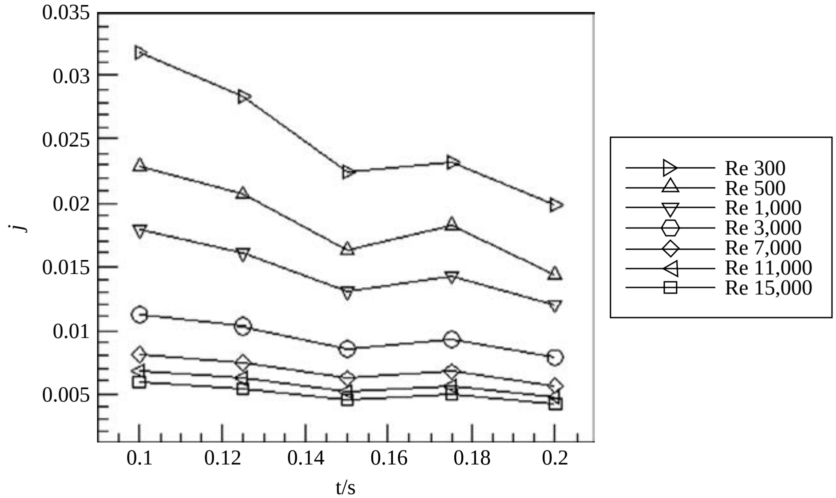

Figure 8.
Dimensionless parameter *t/s* ratio vs *j*

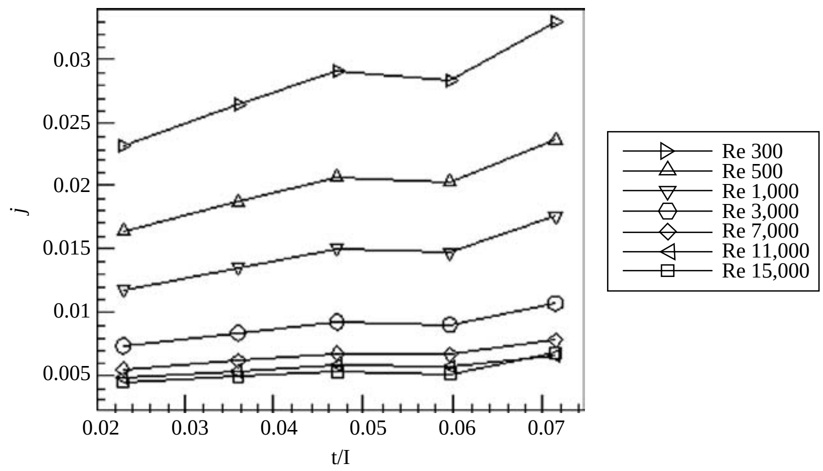

Figure 9.
Dimensionless parameter
t/l ratio vs *j*

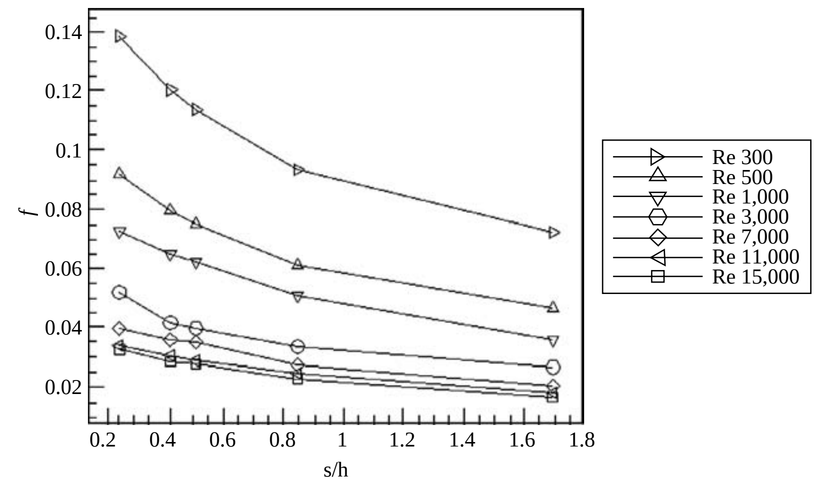

Figure 10.
Dimensionless parameter
s/h ratio vs *f*

of decrease is higher for low Reynolds numbers and lower for high Reynolds number. However, the ratio of increase is predominant for low *Re* (below 1,000) and it is not significant for high *Re* (above 1,000). The increase of this factor is mainly due to higher value of hydraulic diameter. As aspect ratio increases, the hydraulic diameter is also increased.

6.2 Effect of t/s ratio on j

The Colburn *j* factors are plotted against the fin thickness (t) to fin spacing (s) ratio (0.1-0.2) for various Reynolds numbers as shown in Figure 8. It is observed that as Reynolds number increases, *j* factors reduces due to same reasons as mentioned above. Also, it is noted that the *j* factor decrease with increase of *t/s* ratio (up to 0.15) for all

Reynolds numbers. As *t/s* ratio increases, the recirculation zone tends to diminish due to higher flow velocity in the Offset passages. This leads to decrease in heat transfer rate.

6.3 Effect of t/l ratio on j

The Colburn *j* factors are plotted against the fin-thickness (t) to fin length (l) ratio (0.023-0.0714) for various Reynolds numbers as shown in Figure 9. The *j* decreases with increase of *t/l* for all Reynolds numbers. Also, it is noted that the *j* factor increase with increase of *t/l* ratio (up to 0.05) for all Reynolds numbers. It is found that *t/l* ratio increases the recirculation zone as the interrupted layer increases.

6.4 Effect of s/h ratio on f

The friction factor *f* is plotted against the fin spacing (s) to fin height (h) ratio (0.2-1.8) as shown in Figure 10. The *f* factor decreases as *Re* number increases as expected. As aspect ratio increases, the hydraulic diameter is also increased.

6.5 Velocity, temperature and pressure contours

The velocity vector contours for the Reynolds numbers 500 and 11,000 are shown in Figures 11 and 12 for comparison of velocity magnitudes. From the velocity vectors it is quite clear that the flow is more laminar at Reynolds number 500 and is turbulent for the Reynolds number 11,000. This is because the velocity profile is more parabolic in shape for the Reynolds number 500 and is more flat for the Reynolds number 11,000. One more interesting feature is the shape of the vectors, which are pulled towards the interrupted part. This is basically because of the formation of the recirculation zone

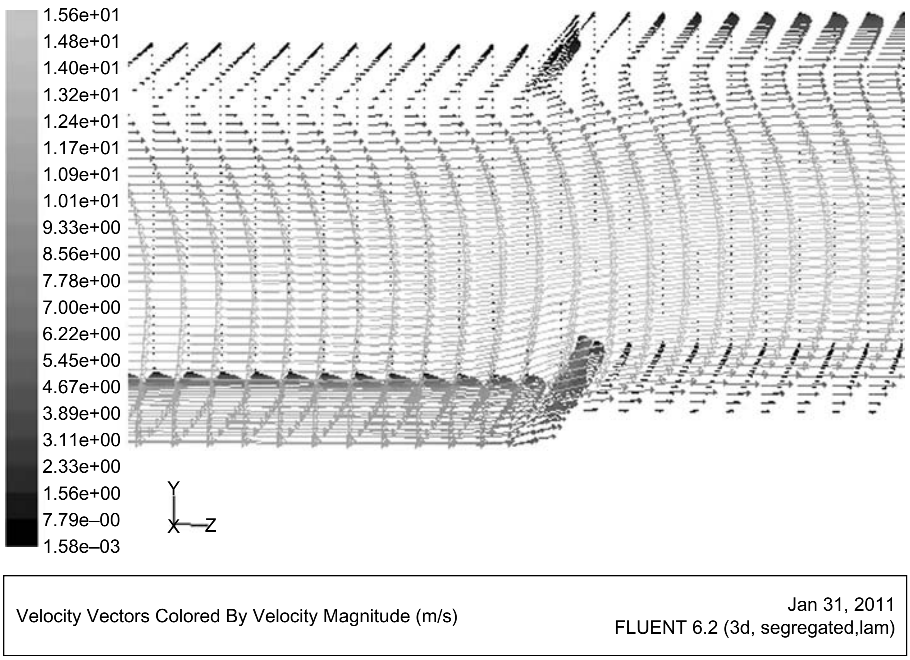

Figure 11.
Velocity vector for Re 500 for fin surface 1.5S-30-0.076

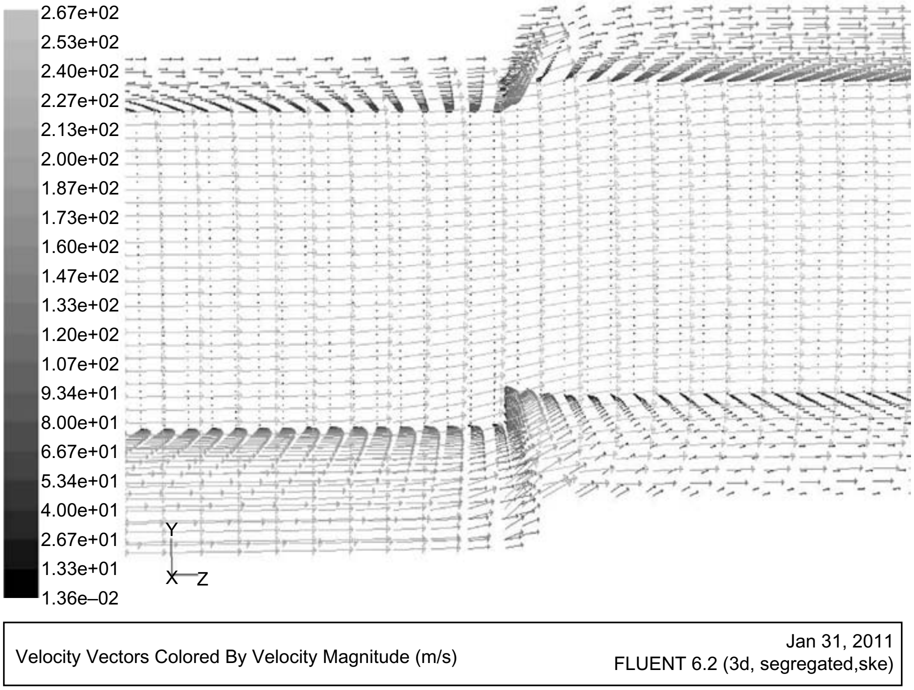

Figure 12. Velocity vector for Re 11,000 for fin surface 1.5S-30-0.076

that is pulling the velocity contours towards it in order to compensate for the excess pressure drop due to the recirculation pocket.

The velocity, temperature and pressure contours are shown in Figures 13-15, respectively, for the same fin surface. It is clear that the boundary layer interrupts and fresh boundary layer starts from the interruption. This is more predominantly seen for the low Reynolds number case ($Re = 500$). Moreover, high-pressure drop occurs at the interruptions and the velocity reaches its maximum value at the same place. It is also quite interesting to note that the temperature enhancement starts building from the interruptions as there is no significant change in temperature at the entrance but significant change is visualized at the first interruption.

7. Correlations for *f* and *j* factors

An exhaustive numerical study has been carried out on the heat transfer phenomena in plate fin surfaces with offset fins. The f vs Re and j vs Re curves show significant non-linearity as shown in the above figures. The correlations have been expressed in terms of two separate equations over the low and high Re regions along with dimensionless geometric parameters. The power law expressions have been used for determining the Colburn factor j and friction factor f as a function of the Reynolds number and dimensionless fin parameters.

The fanning friction factor f are functionally related to Re, s/h, t/l and t/s and it can be represented as:

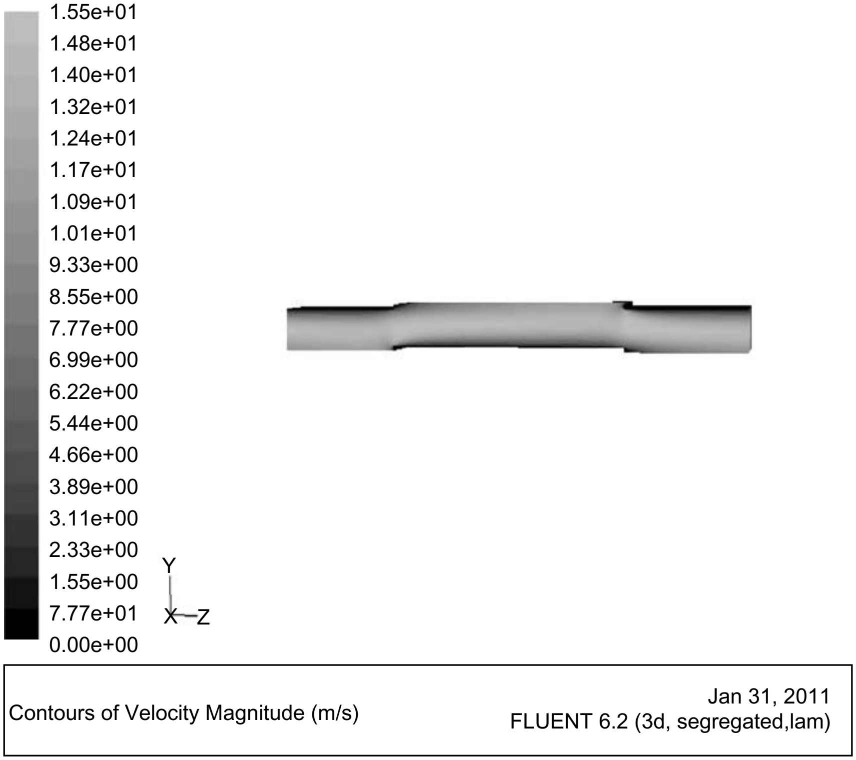

Figure 13.
Velocity contour for Re 500 for fin surface 1.5S-30-0.076

$$f = \mathrm{B}\,\mathrm{Re}^{a1}\left(\frac{s}{h}\right)^{a2}\left(\frac{t}{l}\right)^{a3}\left(\frac{t}{s}\right)^{a4} \tag{7}$$

where B, a1, a2, a3 and a4 are constants.

The use of these power law expressions is justified because variations in f with Re, s/h, t/l and t/s follow constant slope log-linear lines in both laminar and fully turbulent flow regions. The f vs Re data for offset fin surfaces show significant non-linearity over the Reynolds number range $300 \le \mathrm{Re} \le 15{,}000$. Therefore, two separate equations have been proposed for the low and the high Re regions as follows:

$$\mathrm{f} = 10.882(\mathrm{Re})^{-0.79}\left(\frac{s}{h}\right)^{-0.359}\left(\frac{t}{s}\right)^{-0.187}\left(\frac{t}{l}\right)^{0.284} \quad \text{for Laminar range } (300 \le \mathrm{Re} \le 800) \tag{8}$$

$$f = 2.237(\mathrm{Re})^{-0.236}\left(\frac{s}{h}\right)^{-0.347}\left(\frac{t}{s}\right)^{0.151}\left(\frac{t}{l}\right)^{0.639} \quad \text{for Turbulent Range } (1{,}000 \le \mathrm{Re} \le 15{,}000) \tag{9}$$

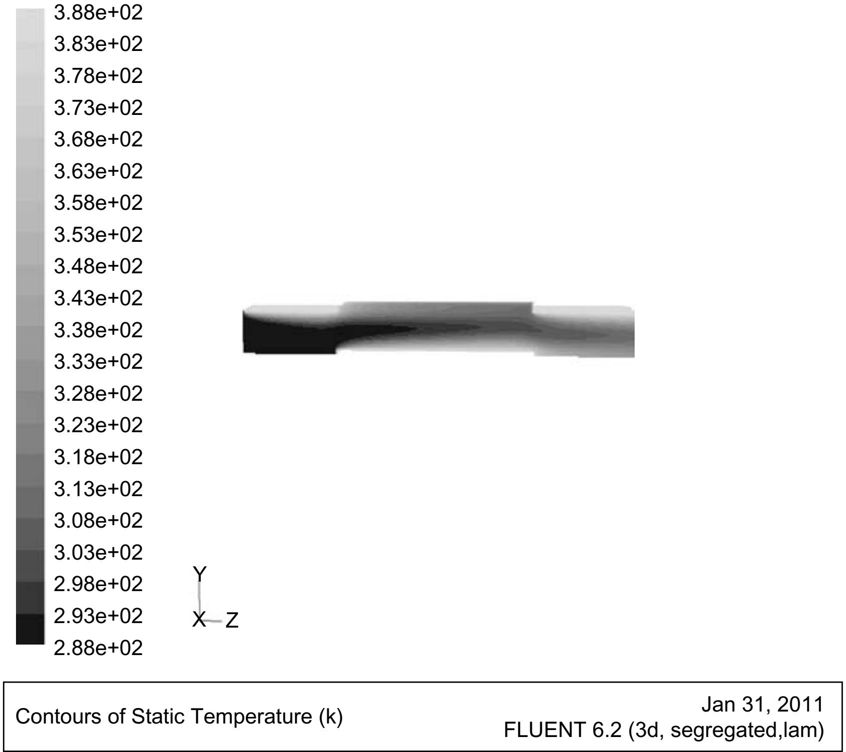

Figure 14.
Temperature contour for Re 500 for fin surface 1.5S-30-0.076

The above correlations correctly predict 99 percent of the f data for laminar regions and 96 percent of the f data for the turbulent regions.

The Colburn factor j are functionally related to Re, s/h, $t/$ and, t/s and it can be represented as:

$$j = \mathrm{C}\,\mathrm{Re}^{\mathrm{b1}}\left(\frac{s}{h}\right)^{\mathrm{b2}}\left(\frac{t}{l}\right)^{\mathrm{b3}}\left(\frac{t}{s}\right)^{\mathrm{b4}} \tag{10}$$

where C, b1, b2, b3 and b4 are constants.

The use of these power law expressions is justified because variations in j with Re, s/h, t/l and t/s follow constant slope log-linear lines in both laminar and fully turbulent flow regions. The j vs Re data for offset fin surfaces show significant non-linearity over the Reynolds number range $300 \leq \mathrm{Re} \leq 15{,}000$. Therefore, two separate equations have been proposed for the low and the high Re regions as follows:

$$j = 0.661(\mathrm{Re})^{-0.651}\left(\frac{s}{h}\right)^{-0.343}\left(\frac{t}{s}\right)^{-0.538}\left(\frac{t}{l}\right)^{0.305} \tag{11}$$

$$\text{for Laminar range } (300 \leq \mathrm{Re} \leq 800)$$

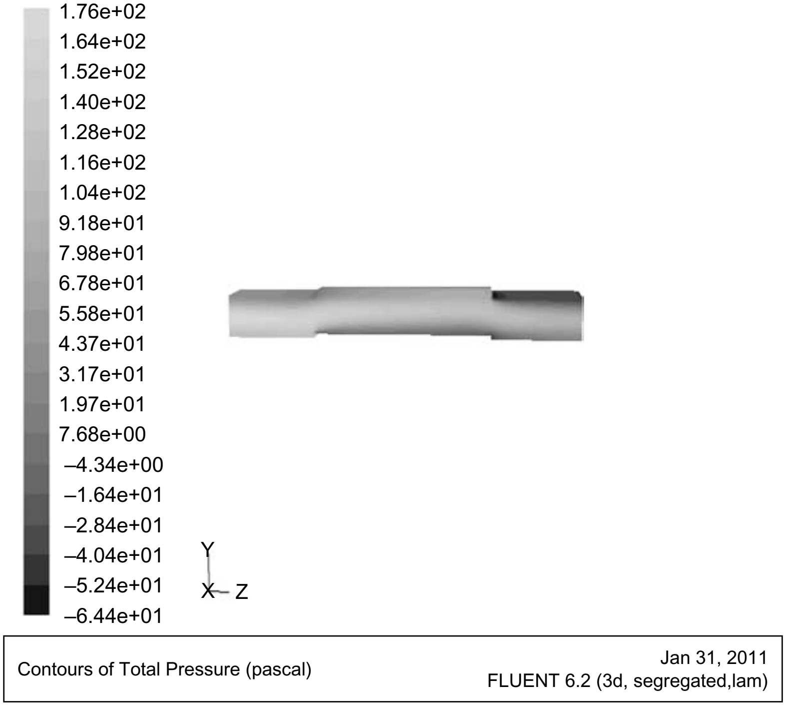

Figure 15.
Pressure contour for Re 500 for fin surface 1.5S-30-0.07

$$j = 0.185(\mathrm{Re})^{-0.396}\left(\frac{s}{h}\right)^{-0.178}\left(\frac{t}{s}\right)^{-0.403}\left(\frac{t}{l}\right)^{0.29} \tag{12}$$

$$\text{for Turbulent range } (1,000 \leq \mathrm{Re} \leq 15,000)$$

The above correlations correctly predict 96 percent of the j data for laminar regime, and 96 percent of the j data for the turbulent regions. Comparisons are, therefore, made with the data from Figures 10.56 to 10.71 of Kays and London (1984). It has been found to be quite similar as shown in Figures 4 and 5.

8. Conclusion

This paper presents the pressure drop and the heat transfer correlations for offset fins which are widely used in Aerospace industry. The expressions provided for the heat transfer coefficient in terms of Colburn j factor and frication factor f allows the computation for all values of Reynolds number, including the laminar and turbulent regions. The generalized correlations are developed for OSFs taking all geometrical parameters into account for entire range of operations of aerospace industry. In addition, the effects of various geometrical parameters are presented as parametric studies. These expressions are well-formed in the laminar and fully turbulent regions, since they can be considered as the standard expressions modified by correction factors. The values obtained from these expressions are in agreement with the literature data. The correlations for the friction factor f and Colburn j factor have been found to be good by comparing with other references. The application of heat transfer and friction factor expressions to a compact plate type heat exchanger gives very good agreement with experimental data. The above correlations can be used by heat exchanger designers and can reduce the number of tests and modification of the prototype to a minimum for similar applications.

References

Anderson, J.D. (1995), *Computational Fluid Dynamics – The Basics with Applications*, McGraw-Hill, New York, NY.

Bhowmik, H. and Lee, K.S. (2009), "Analysis of heat transfer and pressure drop characteristics in an offset strip fin heat exchanger", *International Communications in Heat and Mass Transfer*, Vol. 36, pp. 259-63.

Joshi, H.M. and Webb, R.L. (1987), "Heat transfer and friction in the offset strip – fin heat exchanger", *International Journal of Heat and Mass Transfer*, Vol. 30, pp. 60-84.

Kays, W.M. and London, A.L. (1984), *Compact Heat Exchangers*, McGraw-Hill, New York, NY.

London, A.L. and Shah, R.K. (1968), "Offset rectangular plate fin surfaces – heat transfer and flow friction characteristics", *ASME. Eng. Power 90* (*Series A*), pp. 218-28.

Manglik, R.M. and Bergles, E. (1995), "Heat transfer and pressure drop correlations for the rectangular offset strip fin compact heat exchanger", *Experimental Thermal Fluid Science*, Vol. 10, pp. 171-80.

Mochizuki, S., Yagi, Y. and Yang, W.J. (1987), "Transport phenomena in stacks of interrupted parallel-plate surfaces", *Experimental Heat Transfer*, Vol. 1, pp. 127-40.

Patankar, S.V. (1980), *Numerical Heat Transfer and Fluid Flow*, Hemisphere, Washington, DC.

Patankar, S.V., Liu, C.H. and Sparrow, E. (1977), "Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross-sectional area", *Journal of Heat Transfer*, Vol. 99, pp. 180-6.

Ranganayakulu, C., Ismail, L.S., Vasudeva Rao, V. and Rajeshwar, S. (2008), "Optimization of wavy fin parameters in compact heat exchangers", *Proceedings of XIX National Heat and Mass Transfer & VIII ISHMT-ASME Heat and Mass Transfer Conference, JNTU, Hyderabad, India, January*.

Sheik Ismail, L., Ranganayakulu, C. and Shah, R.K. (2009), "Numerical study of flow patterns of compact plate-fin heat exchangers and generation of design data for offset and wavy fins", *International Journal of Heat and Mass Transfer*, Vol. 52, pp. 3972-83.

Sheik Ismail, L., Velraj, R. and Ranganayakulu, C. (2010), "Studies on pumping power in terms of pressure drop and heat transfer characteristics of compact plate-fin exchangers – a review", *Renewable and Sustainable Energy Reviews*, Vol. 14, pp. 478-85.

Versteeg, H.K. and Malalasekera, W. (1995), *An Introduction to Computational Fluid Dynamics – The Finite Volume Method*, Prentice-Hall, London.

Wieting, A.R. (1975), "Empirical correlations for heat transfer and flow friction characteristics of rectangular offset-fin plate-fin heat exchangers", *Journal of Heat Transfer*, Vol. 97, pp. 488-90.

Corresponding author
Ranganayakulu Chennu can be contacted at: r_chennu@hotmail.com