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Numerical analysis of compact plate-fin heat exchangers for aerospace applications

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NUMERICAL ANALYSIS OF COMPACT PLATE-FIN HEAT EXCHANGERS FOR AEROSPACE APPLICATIONS

1. INTRODUCTION

Depending on the application, various types of augmented heat transfer surfaces such as wavy fins, offset strip fins, louvered fins, plain and perforated fins are used in aerospace applications. The surface geometries of wavy and OSF fins are described by the fin height (*h*), transverse spacing (*s*) and thickness (*t*). Interrupted flow length of the offset strip fin is described by offset strip/fin length (*l*), and that of the wavy fin, by the pitch of the wave (L). Thermo-hydraulic design of a compact heat exchanger 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). We focus here on offset strip fins, wavy fins, plain and perforated fins. The orientation of inlet and outlet headers plays a major role in performance especially in aerospace applications, where the orientation of headers and nozzles are not straight and uniform due to space limitations. The accurate prediction of the thermal performance of a compact heat exchanger in the design stage is highly desirable for aerospace applications.

The longitudinal heat conduction (LHC) through the heat exchanger wall structure in the direction of fluid flows has the effect of decreasing the exchanger performance for a specified NTU, and this reduction may be quite serious in exchangers with short flow length designed for high effectiveness (> 80%) [1]. These effects have been well recognized and the numerical data are available in [2,3] for periodic-flow heat exchangers and in [4-6] for the direct transfer type heat exchangers. The flow mal-distribution effects have been well recognized for heat exchangers. The flow non uniformity through the exchanger is generally associated with improper exchanger entrance configuration, due to poor header design and imperfect passage-to-passage flow distribution in a highly compact heat exchanger caused by various manufacturing tolerances. The flow non uniformity (FN) effects have been well recognized and presented for heat exchangers[7-14]. Ranganayakulu et al.[7] carried out the Finite Element analysis for effects of FN on cross flow Compact Heat Exchanger (CHE). Chiou [8-11] carried out the FN effects using Finite Difference Method for various types of heat exchangers, such as cross flow heat exchanger [8], automobile airconditioning condenser [9] and evaporator [10] and FN effects on both cold and hot fluids of cross flow heat exchanger [11]. Kranc [12] studied the effect of non uniform water distribution on cooling tower performance. Similarly, the fluid inlet temperature non uniformity (TN) effects have also been investigated for cross flow heat exchangers [15,16].

In actual practice, heat exchangers may be subjected to wall LHC, inlet FN and TN together. Literature on the investigation of combined effects of LHC, TN and FN for a cross flow plate-fin heat exchanger is limited [15,16]. Moreover, all the previous works [8-12] were limited to specific types of non-uniform flow models and can not be interpolated or extrapolated for other types of flow mal-distributions. Also, Chiou [13] analysed the effects of LHC and FN on compact heat exchangers. Zhang et al. [17] have investigated the flow non uniformity in a plate-fin heat exchanger by a CFD software. Based on the investigation, two modified headers with a two-stage-distributing structure are proposed to reduce the flow non uniformity. Ranganayakulu et al. [18] studied the effects of the fluid flow non uniformity due to the improper header/nozzle configuration with the CFD tool for a typical stainless steel compact plate-fin heat exchanger. Wen et al. [19] investigated flow characteristics of flow field in the entrance of a plate-fin exchanger by means of Particle Image Velocimetry (PIV). Based on experiments, they suggested that punched baffle could effectively improve fluid flow distribution in the header.

In addition to the design optimization studies using Finite Element Analysis discussed above, the generation of heat transfer *j* factors and friction *f* factors for various types of CHE fins are also presented here. Webb and Joshi [20] presented analytical models to predict the heat transfer coefficients and friction factors of an offset strip-fin heat exchanger by idealizing an unit cell model. Wieting [21] gave empirical correlations based on the work done by Kays and London [1]. While generating those correlations, he had taken two Reynolds number regimes, such as, primarily laminar (Re ≤ 1000) and primarily turbulent (Re ≥ 2000). Mochizuki and Yagi [22] attempted to find the effect of the strip length on the j and f factors using their experimental study. They concluded that the optimum strip length has to be selected to get maximum goodness factor (j/f) value. Manson [23] developed correlations to predict thermo-hydraulic performances of offset fins. Manglik and Bergles [24] provided a single correlation that was applicable for laminar and turbulent regions for offset fins. Maiti [25] attempted multiple regression analysis using data of Kays and London [17] and in-house experimental test results to establish general correlations.

Guannan and Shah [26] carried out the 2D and 3D numerical computations for the idealized OSF in the laminar and transition flow regions to investigate differences between numerical results and experimental data and showed excellent correlation with the experimental data except at the highest Reynolds number. For generation of j and f vs. Re data numerically, the entry effects into the fin plays a predominant role. In order to overcome this difficulty, Patankar et al. [27] introduced the concept of periodic fully developed flow and heat transfer. The underlying concept is 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. The other important boundary condition that played a predominant role is the wall boundary condition for thermal analysis. Ciofalo et al. [28] mentioned that the constant temperature boundary condition yielded lower j values compared to those for the constant heat flux boundary condition but agreed well with the experimental values.

2. FINITE ELEMENT ANALYSIS

A discretized model of a cross flow plate-fin heat exchanger is shown in Fig.1-(a). It is divided into a number of equal strips. The strip 1 is isolated and shown in Fig.1-(b). The exchanger may be visualized as a wall separating the two fluid streams flowing at right angles with plate-fins on both sides as shown in Fig.1-(c). Each strip consists of a number of pairs of stacks which carry hot and cold fluids. A pair of stacks is separated and shown in Fig.1-(d). It is the basic element for which the element matrices are derived. In the cross flow plate-fin exchanger wall, a 4-noded element has been considered for studying the two-dimensional longitudinal wall heat conduction effects. Two-noded linear elements have been considered for both hot and cold fluids in the present analysis. Thus a 16-noded box element has been obtained as shown in Fig.1-(d). Similarly, the discretized exchangers for counter flow plate-fin and parallel-flow plate-fin type heat exchangers are shown in Figs.1-(e)-(f) respectively. The wall temperature distribution in counter flow plate-fin and parallel-flow plate-fin heat exchangers is one-dimensional and hence 2-noded elements are considered in the exchanger plate for longitudinal wall heat conduction effects. Thus a 10-noded element has been obtained for counter flow and parallel-flow plate-fin heat exchanger as shown in Figs.1-(e)-(f) respectively.

Here, the possible inlet fluid flow non uniformity models are generated by distorting the velocity profile and keeping the average fluid velocity as unity [6-7]. One of this model named as Model A1 is tabulated in Table 1. Also, a typical fluid flow mal-distribution model is shown in Fig. 2. The velocity at the wall of inlet duct is zero. The non-zero velocity values in the proposed models are at the points away from the wall of transition duct. In each model, there are 10 x 10 local flow non uniformity dimensionless parameters (α's), which corresponds to the 10 x 10 subdivisions on the x-z plane perpendicular to the direction of non-uniform fluid flow. The same model can also be used for TN cases. In view of the symmetry with respect to o-x and o-y, only one-fourth of flow non uniformity parameters (α's) are presented in Table 1.

The local inlet flow non uniformity parameter (α) is defined as [8],

$$\alpha = \frac{actual\ inlet\ \ flow}{average\ inlet\ \ flow\ if\ \ flow\ distribution\ is\ uniform} \tag{1}$$

A discretized model of a cross flow plate-fin heat exchanger is shown in Fig. 1(c). It is divided into a number of equal strips. Each strip consists of number of pairs of 16-noded stacks which carries hot and cold fluids as shown in Fig. 1(d). In the cross flow heat exchanger, a 4-noded element is considered for studying the two-dimensional LHC effects on exchanger wall. These are the basic elemental exchangers for which the finite element equations are formulated as coupled conduction-convection problems [14].

The following dimensionless parameters are introduced to study the influence of LHC and FN on the exchanger performance:

a) LHC parameter (λ_h) = (kA_w)/(LC_h) (2)

b) LHC parameter (λ_c) = (kA_w)/(lC_c) (3)

$$c)\quad Correction\ \ factor\,(\tau) = \frac{\varepsilon_0 - \varepsilon_{LHC,FN}}{\varepsilon_0} \tag{4}$$

The correction factor (τ) directly shows the degree of deterioration of the exchanger effectiveness.

2.1 GOVERNING EQUATIONS

The following assumptions are made for FEM analysis:

1) Steady state conditions are assumed.
2) No phase change and no heat generation within the exchanger.
3) The exchanger where both the fluids are unmixed is considered. Cross or transverse mixing of fluids is not considered. Change of flow distribution inside the exchanger is neglected.
4) In the elements, the temperatures of the fluids are assumed to vary only along their flow lengths.
5) The entry length effects are not considered.
6) No heat transfer between the exchanger and the surrounding is assumed.

Based on the above assumptions, the governing energy balance equations (considering two-dimensional longitudinal heat conduction in the exchanger plate for a cross flow plate-fin exchanger) are formulated as shown below:

$$\frac{(kA_w)_h \partial^2 T_{w,t}}{l\partial x^2} + \frac{(kA_w)_c \partial^2 T_{w,t}{}^t}{L\partial y^2} - (\vartheta h a')_h (T_h - T_{w,t}) = q_t \tag{5}$$

$$\frac{-(\alpha MC_P)_h}{l}\frac{\partial T_h}{\partial x}+(\vartheta\, h\, a')_h(T_h-T_{w,t})+(\vartheta\, h\, a')_h(T_h-T_{w,m})=0 \tag{6}$$

$$\frac{(kA_w)_h\,\partial^2 T_{w,m}}{l\,\partial x^2}+\frac{(kA_w)_c\,\partial^2 T_{w,m}}{L\,\partial y^2}-(\vartheta h a')_h(T_h-T_{w,m})+(\vartheta\, h\, a')_c(T_{w,m}-T_c)=0 \tag{7}$$

$$\frac{(\alpha MC_P)_c}{L}\frac{\partial T_c}{\partial y}+(\vartheta\, h\, a')_c(T_{w,m}-T_c)+(\vartheta\, h\, a')_c(T_{w,b}-T_c)=0 \tag{8}$$

$$\frac{(kA_w)_h\,\partial^2 T_{w,b}}{l\,\partial x^2}+\frac{(kA_w)_c\,\partial^2 T_{w,b}}{L\,\partial y^2}+(\vartheta\, h\, a')_c(T_{w,b}-T_c)=q_b \tag{9}$$

The boundary conditions are,

$$T_h(0,y)=\beta T_{h,in} \quad ; \quad T_c(x,0)=\beta T_{c,in} \tag{10}$$

$$\frac{\partial T_w(0,y)}{\partial x}=\frac{\partial T_w(L,y)}{\partial x}=0 \quad ; \quad \frac{\partial T_w(x,0)}{\partial y}=\frac{\partial T_w(x,l)}{\partial y}=0 \tag{11}$$

The temperature variation of the hot fluid (T_h), and cold fluid (T_c) in the element are approximated by a linear variation as,

$$T_h = N_i\, T_i + N_j\, T_j \tag{12}$$

$$T_c = N_k\, T_k + N_l\, T_l \tag{13}$$

The temperature variation of exchanger plate (T_w) is approximated as

$$T_w = N_m\, T_m + N_n\, T_n + N_o\, T_o + N_p\, T_p \tag{14}$$

where N_i, N_j, N_k, N_l, N_m, N_n N_o and N_p are shape functions.

Substituting the approximations in the above equations and using Galerkin's method, the final set of element matrices are obtained. The element matrices for other pairs of the stacks in the strip are evaluated and assembled into a global matrix. The final sets of simultaneous equations are solved after incorporating the known boundary conditions (inlet temperatures). Thus by marching in a proper sequence, the temperature distribution in the exchanger is obtained. Analytical solutions without considering the effects of LHC and FN are obtained using the solution procedure given by Kays and London [1]. Here, the exchanger thermal performance deteriorations due to LHC and FN are plotted as a function of NTU (NTU overall) for three magnitudes of C_{min}/C_{max} (1.0, 0.6 and 0.2) and for three magnitudes of λ (0.05, 0.1 and 0.2) for the following cases:

a) The combined effects of LHC and FN on C_{min} fluid side

b) The combined effects of LHC and FN on C_{max} fluid side

c) The combined effects of LHC and FN on both C_{min} and C_{max} fluid sides.

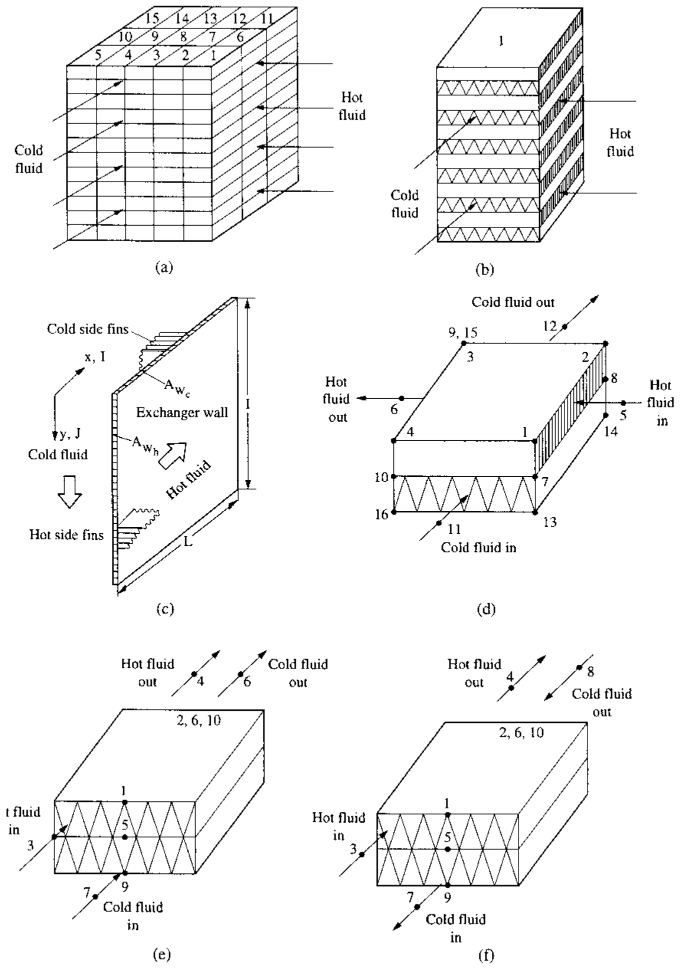


Fig. 1 Single pass Pate-fin Heat Exchanger

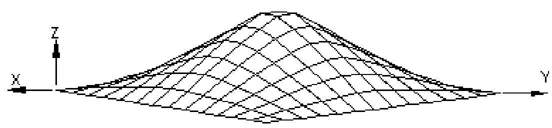
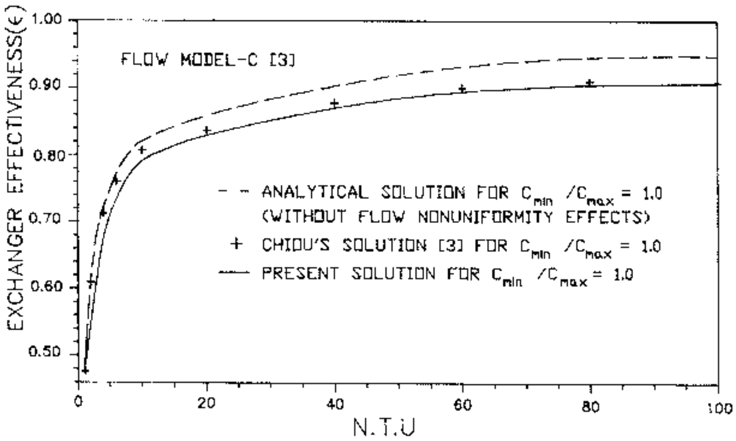
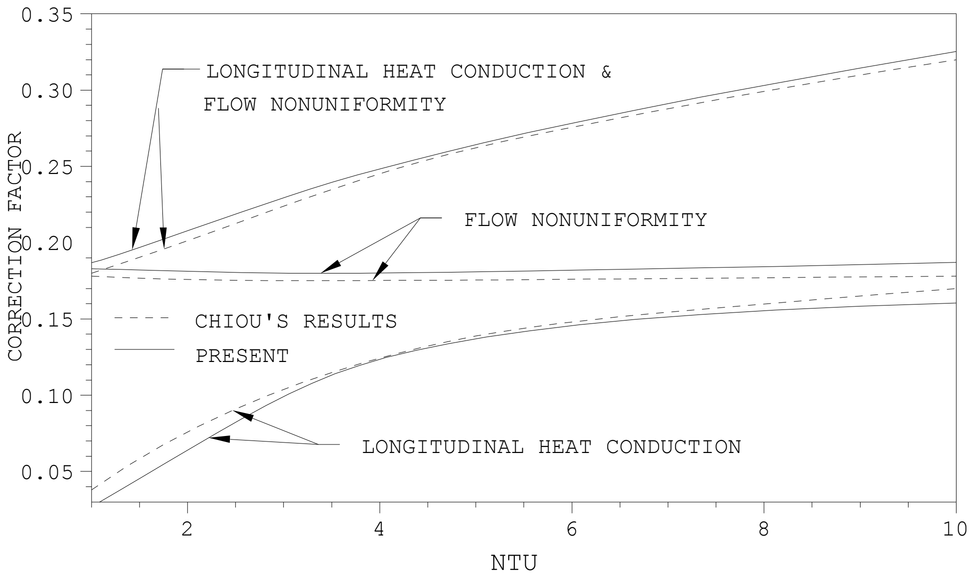


Fig. 2 Flow non-uniformity model



a) Comparison with exchanger effectiveness



b) Longitudinal Heat Conduction and inlet flow non-uniformity effects

Fig. 3 Comparison of results

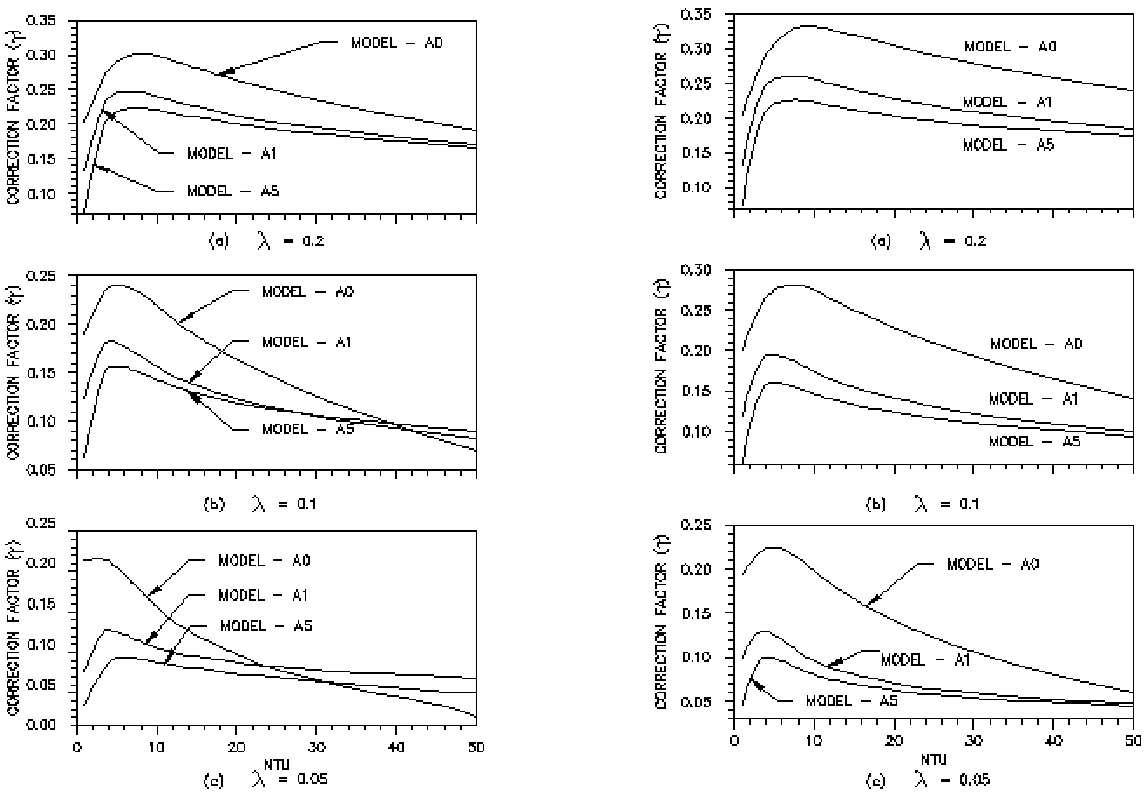


Fig. 4 Combined effects of LHC and Flow non-uniformity–Plate-fin Heat Exchanger,

(i) C_{max} fluid side (ii) both fluid side: C_{min}/C_{max}=1.0

2.2 VALIDATION OF FEM DATA

The accuracy of the solution is depending on the number of the elements used. Actually, the number of elements used is determined by a compromise between the accuracy desired and the time required by the computer. The present Finite Element Analysis is compared with analytical results and the individual effects of LHC [6] and FN [7] of cross flow heat exchanger. However, Chiou [13] has attempted the combined effects of LHC and FN on cross flow heat exchanger for specific flow non uniformity models. The relative comparison of these cases with the present finite element results is shown in Fig. 3. The finite element results are compared with analytical results as shown in Fig. 3(a) and numerical results as shown in Fig. 3(b). This comparison is found to be good. Also, Fig. 3(b) shows the relative comparison of results of LHC, FN and the combined effects of LHC and FN for cross flow plate-fin heat exchanger.

2.3 FEM RESULTS

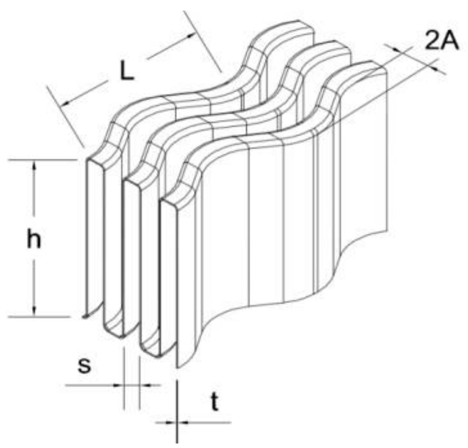
The performance evaluation with the combined effects of wall LHC and inlet fluid FN on cross flow heat exchanger is presented for balanced flow, $C_{min}/C_{max} = 1$, as well as for unbalanced flow, C_{min}/C_{max} not equal to one. Detailed results for all cases can be found in earlier papers [15-16, 30 & 31]. However, a sample case is shown here. The relation between the ratio of C_{min}/C_{max} and λ with correction factor (τ) is shown in Fig. 4. This figure shows that the performance deteriorations are higher for balanced flows as compared to that of unbalanced flows. For example, the performance deteriorations (at NTU = 10 and λ = 0.1) are around 15.5% for C_{min}/C_{max} = 1.0, 13% for C_{min}/C_{max} = 0.6 and 8.5% for C_{min}/C_{max} = 0.2 for flow model A1 and when the flow non uniformity is considered on C_{max} fluid side. Similarly, the relation between λ and correction factor (τ) is shown in Fig. 4(a). This figure shows that the correction factor (τ) increases with the increase of longitudinal heat conduction parameter, λ. For example, the correction factors at NTU = 10, are 23.5% for $\lambda = 0.2$, 15% for $\lambda = 0.1$ and 8% for $\lambda = 0.05$ for flow model A1. It has been observed that the effects of longitudinal heat conduction and flow non-uniformity on the deterioration of exchanger performance tend to augment each other in the regions of lower NTU, but tend to eliminate each other in the regions of higher NTU. The complete results covering various ranges of design parameters are available in references [5-11, 15-18 & 29-32].

3. CFD ANALYSIS

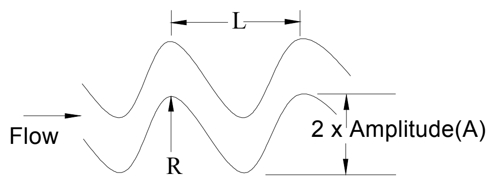
The following are some of the assumptions made in the CFD simulation: (a) the flow is stable in the computational domain; (b) the fluid flow meets the Boussinesq assumption and (c) the fluid in the domain is incompressible. In this work, CFD software FLUENT is employed for simulation.

3.1 FIN GEOMETRIES

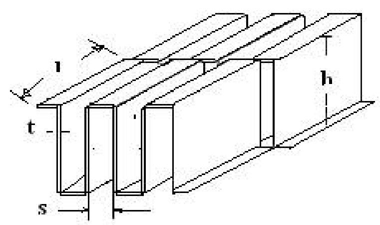
In this paper the following geometries of various types of fins are considered for CFD analysis for estimation of *j* and *f* data: Fig. 5(a) shows the isometric view of a Wavy fin, Fig 5(b) shows Wavy fin dimensional notations, Fig. 5(c) shows the schematic of Offset fins, Fig. 5(d) shows the geometry of Rectangular plain fin, Fig. 5(e) shows the Model of Rectangular perforated fin, Fig. 5(f) shows the Fin geometry of Triangular plain fin and Fig 5(g) shows the model of Triangular perforated fin. The dimensionless representations of these variables are given by ratios of s/h, 2A/s and L/2A. In the fin designation as denoted by Kays and London [1], the first number indicates the fin density (fins/inch), the second number indicates the fin wavy length (L) in inches and the third number indicates the fin thickness in inches as 11.44-3/8W-0.006.



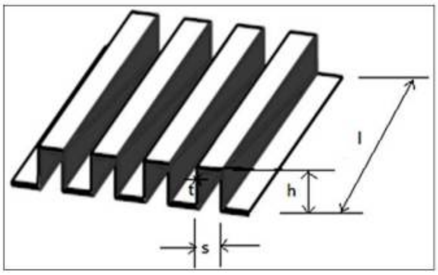
a) Schematic of Wavy fin



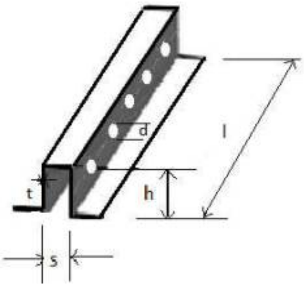
(b) Wavy fin dimensional notations



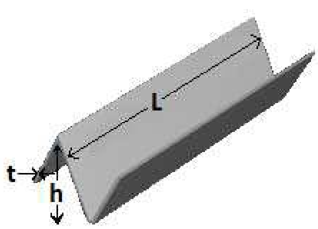
(c) Offset fin dimensional notations



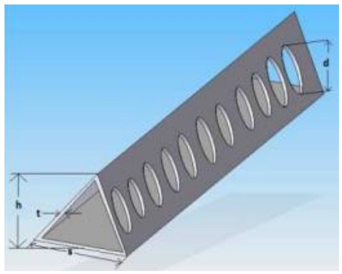
(d) Rectangular Plain fin geometry



(e) Model of Rectangular perforated fin



(f) Fin geometry of Triangular plain fin

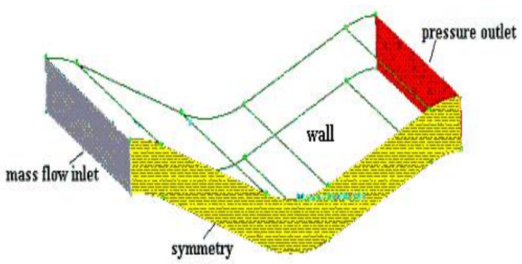


(g) Model of triangular perforated fin

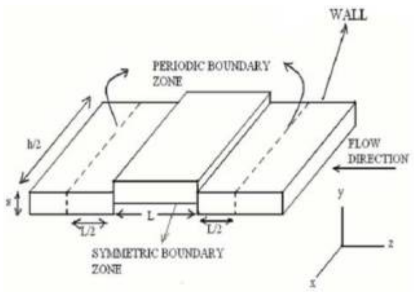
Fig. 5 Geometries of different types of Fins

3.2 NUMERICAL SIMULATION

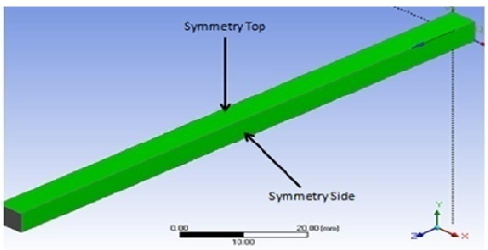
The computational domains of various types of fin models are shown in Fig. 6. The analysis is carried out in two phases. In first phase, the fin is taken and characterized for f values over a range of Reynolds number. In second phase, the j value is determined for the same range by switching on the energy equation. The mass flow rates are determined for a range of Reynolds number from 100 to 15000. In order to overcome the entrance effect, the concept of periodically fully developed flow as suggested by Patankar [33] is implemented for flow analysis. After the analysis, the pressure drop for unit length 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 Kays and London [1]. 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 100 to 15,000 in order to draw the f vs. Re curve.



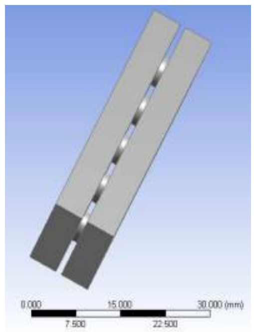
(a) Computational domain of a Wavy Fin



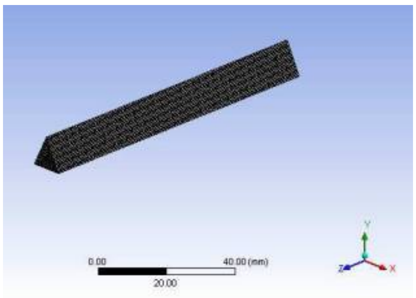
(b) Computational domain for an Offset fin



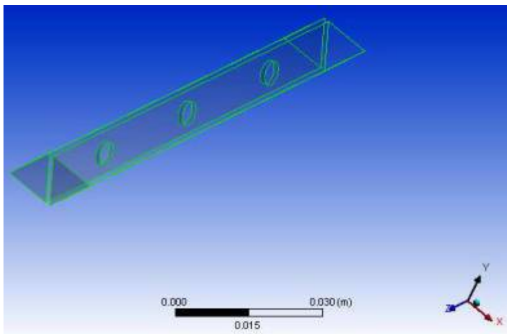
(c) Computational domain of a plain fin



(d) Computational domain of a perforated fin



(e) Isometric view of meshed model



(f) Element Model of Triangular perforated fin

Fig. 6 Computational domains of various types of fin models

3.3 VALIDATION OF CFD DATA

The results obtained from FLUENT for Wavy fins in the form of Colburn *j* and Fanning friction *f* factors are compared in Fig. 7 for Wavy fin, which are taken from Kays and London [1]. It is evident that all data of both *j* and *f* factors are matched well as these are close each other except Jungi et al. [34] for a Wavy fin.

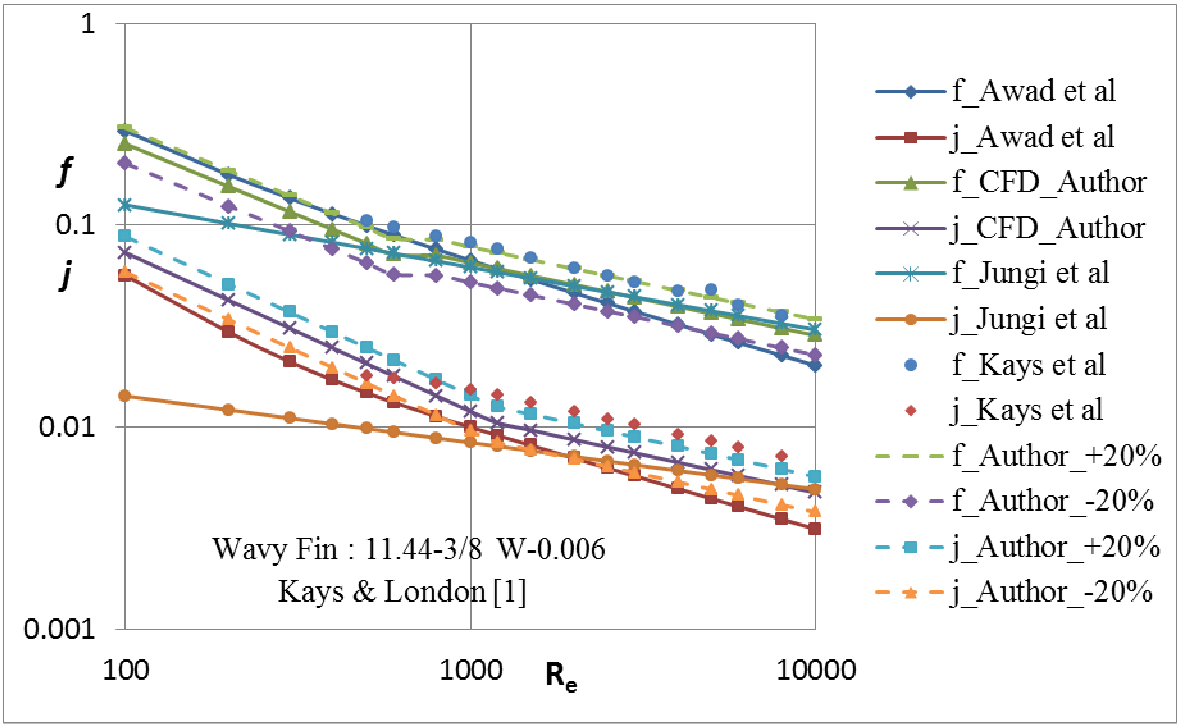


Fig. 7 Comparison of Kays and London [1] and CFD data of *j* and *f* vs Re along with Awad et al [35] and Jungi et al [34] data for the Wavy fin: 11.44-3/8W-0.006.

3.4 RESULTS OF CFD ANALYSIS

Finally, the *f* and *j* values are generated using CFD technique for different types of offset strip fins and wavy fins. The complete results can be seen in earlier papers [36-44]. However, the correlations generated using CFD are listed below for various types of fins in Table 2 for ready reference:

4. CONCLUSION

This paper presents both Finite Element Method and Computational Fluid Dynamic Analysis of Compact Heat Exchangers for Aerospace applications. Using FEM the Flow Non-uniformity, Temperature Non-uniformity and Longitudinal Heat Conduction effects are analysed. The combined effects of FN, TN and LHC are also estimated and compared with individual effects. The thermal performance deterioration of cross flow compact heat exchanger due to the combined effects LHC and FN is not always negligible, especially when the fluid capacity rate ratio of both fluids is equal to 1.0 and when the longitudinal heat conduction parameter(λ) is greater than 0.005.

Using CFD, the various types of fins such as Offset Strip fins, Wavy fins, Rectangular fins, Triangular fins, Triangular and Rectangular perforated fins, which are widely used in aerospace industry, are analysed. The expressions provided for the heat transfer coefficient in terms of Colburn *j* factor and friction factor *f* allows the computation for all values of Reynolds number, including the laminar and turbulent regions for CHE design ranges. In addition, the data of these correlations are compared with other numerical data by analyzing open literature thoroughly. These correlations are well formed in the laminar and fully turbulent regions, since they can be considered as the standard correlations.

The correlations for the friction factor *f* and Colburn factor *j* have found to be good by comparing with other references. The above FEM results and CFD 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.

NOMENCLATURE

A	heat transfer area , m^2
A_f	free flow area, m^2
a'	elemental heat transfer area per unit core area, dimensionless
A_W	total solid elemental area available for longitudinal heat conduction, m^2
a	elemental length of the exchanger in the x-direction, m
b	elemental length of the exchanger in the y-direction, m
c_p	specific heat of the fluid at constant pressure, J/kg K
$C = mC_p$	fluid heat capacity rate, J/s K
CHE	Compact Heat Exchanger
d	width of the exchanger inlet duct, m
FN	Flow Non uniformity case
f	Fanning friction factor, dimensionless
FPI	fins per inch
h	convection heat transfer coefficient, W/m^2 K
h	fin height, mm
I	divisions in the x-direction (1,2,3 ....n)
j	Colburn factor ($S_tPr^{2/3}$), dimensionless
J	divisions in the y-direction (1,2,3 ....n)
k	thermal conductivity of the exchanger wall, W/m K
l	offset strip/fin length, mm
L	pitch of fin waviness, mm
LHC	Longitudinal Heat Conduction case
m	mass flow rate, kg/s
Pr	Prandtl number, dimensionless
q	heat flux, W/m^2
Re	Reynolds number, dimensionless
R	Wavy fin curvature radius, mm
S_t	Stanton number, dimensionless
s	fin spacing, mm
T	Temperature, ^oC
TN	Temperature Non uniformity case
T_{wall}	wall temperature, ^oC
t	fin thickness, m

GREEK SYMBOLS

α	flow non uniformity parameter as defined in equation (1)
ε_o	exchanger effectiveness without longitudinal heat conduction and flow non uniformity, dimensionless
$\varepsilon_{LHC,FN}$	exchanger effectiveness with longitudinal heat conduction and flow non uniformity, dimensionless
λ	longitudinal heat conduction parameter,, dimensionless
τ	conduction effect factor or correction factor, dimensionless

η fin effectiveness

ϑ Overall surface effectiveness

SUBSCRIPTS

b - bottom plate, c - cold side, h - hot side, i - inlet, m - middle plate, min – minimum, max – maximum, o - outlet, t - top plate.

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Model – A1					
J = 1 ; 10	0.500	0.500	0.500	0.500	0.500
2 ; 9	0.500	0.639	0.776	0.899	0.998
3 ; 8	0.500	0.776	1.045	1.291	1.489
4 ; 7	0.500	0.899	1.291	1.655	1.956
5 ; 6	0.500	0.998	1.489	1.956	2.356

Table 1- Flow Non-uniformity Parameters (α's)

S. No	Type of fin surface	Design data	Correlations	Range of Applicability
1	Wavy fins	*f*	$9.827R_e^{-0.705}\left(\frac{h}{s}\right)^{0.322}\left(\frac{2A}{s}\right)^{-0.394}\left(\frac{L}{2A}\right)^{-0.603}$	$100 \le R_e \le 800$
		f	$10.628\,R_e^{-0.359}\left(\frac{h}{s}\right)^{0.264}\left(\frac{2A}{s}\right)^{-0.848}\left(\frac{L}{2A}\right)^{-1.931}$	$1000 \le R_e \le 15000$
		j	$2.348\,R_e^{-0.786}\left(\frac{h}{s}\right)^{0.312}\left(\frac{2A}{s}\right)^{-0.192}\left(\frac{L}{2A}\right)^{-0.432}$	$100 \le R_e \le 800$
		j	$0.242\,R_e^{-0.375}\left(\frac{h}{s}\right)^{0.235}\left(\frac{2A}{s}\right)^{-0.288}\left(\frac{L}{2A}\right)^{-0.553}$	$1000 \le R_e \le 5000$
2	*Wavy fin R134a*	*j*	$2.989Re^{-0.54241}\frac{h^{-0.72276}}{s}\frac{2A^{-0.83914}}{s}\frac{L^{-0.7588}}{2A}$	$100 \le Re \le 1000$
3	*Wavy fin Water*	*j*	$1.154Re^{-0.65938}\frac{h^{-0.96698}}{s}\frac{2A^{0.176702}}{s}\frac{L^{0.288785}}{2A}$	$100 \le Re \le 1000$
4	*Wavy fin Water & R134a*	*f*	$18.607Re^{-0.59381}\frac{h^{-0.088954}}{s}\frac{2A^{-0.46976}}{s}\frac{L^{-0.92621}}{2A}$	$100 \le Re \le 1000$
5	Offset fins	*f*	$10.882(R_e)^{-0.79}(s/h)^{-0.359}(t/s)^{-0.187}(t/l)^{0.284}$	$300 \le R_e \le 800$
		f	$2.237(R_e)^{-0.236}(s/h)^{-0.347}(t/s)^{0.151}(t/l)^{0.639}$	$1000 \le R_e 15000$
		j	$0.661(R_e)^{-0.651}(s/h)^{-0.343}(t/s)^{-0.538}(t/l)^{0.305}$	$300 \le R_e \le 800$
		j	$0.185(Re)^{-0.396}(s/h)^{-0.178}(t/s)^{-0.403}(t/l)^{0.29}$	$1000 \le R_e \le 15000$
6	Rectangular plain	*f*	$12.892(Re)^{-1.229}(h/s)^{0.452}(t/s)^{-0.198}$	$100 \le Re \le 1000$
		f	$3.133\,(Re)^{-1.285}(h/s)^{0.247}(t/s)^{-0.181}$	$1000 < Re \le 7500$
		j	$0.454\,(Re)^{-0.977}(h/s)^{0.435}(t/s)^{-0.227}$	$100 \le Re \le 1000$
		j	$0.166\,(Re)^{-1.011}(h/s)^{0.228}(t/s)^{-0..366}$	$1000 < Re \le 7500$
7	Rectangular perforated fin	*f*	$0.7127(Re)^{-1.8858}(h/s)^{0.4196}(t/s)^{-1.4826}$	$100 \le Re \le 1000$
		f	$0.4345(Re)^{-1.3029}(h/s)^{0.3725}(t/s)^{-1.3178}$	$1000 < Re \le 7500$
		j	$0.121\,(Re)^{-2.2920}(h/s)^{2.75}(t/s)^{-1.830}$	$100 \le Re \le 1000$
		j	$11.71\,(Re)^{-2.3111}(h/s)^{2.144}(t/s)^{-1.9237}$	$1000 < Re \le 7500$

8	Triangular Plain fin	f	$3.12(\mathrm{Re})^{-0.852}(h/s)^{0.156}(t/s)^{-0.184}$	100 ≤Re≤ 1000
		f	$2.69(\mathrm{Re})^{-0.918}(h/s)^{0.355}(t/s)^{-0.175}$	1000<Re≤10000
		j	$0.718(\mathrm{Re})^{-0.625}(h/s)^{0.765}(t/s)^{0.765}$	100 ≤Re≤1000
		j	$0.789(\mathrm{Re})^{-1.1218}(h/s)^{1.235}(t/s)^{-0.764}$	1000<Re≤10000
9	Triangular Perforated fin	f	$10.127(\mathrm{Re})^{-1.588}(h/s)^{0.778}(t/s)^{-0.868}$	300≤Re≤1000
		f	$1.685(\mathrm{Re})^{-0.798}(h/s)^{0.447}(t/s)^{-0.276}$	1000< Re≤7500
		j	$0.544(\mathrm{Re})^{-1.673}(h/s)^{2.278}(t/s)^{-1.589}$	300≤ Re≤1000
		j	$7.579(\mathrm{Re})^{-1.626}(h/s)^{1.185}(t/s)^{-1.689}$	1000<Re ≤7500

Table 2: f and j correlations for various types of CHE fins