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Development of Pressure Drop and Heat Transfer Characteristics for A Plate Type Offset-Fin Compact Heat Exchanger Using Numerical Investigation

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Abstract—*A steady-state, three-dimensional numerical model is used to study the heat transfer and pressure drop characteristics of an offset strip fin compact heat exchanger application of which is in oil cooler using ANSYS Fluent 16.0. Variations in the Fanning friction factor f and the Colburn factor j relative to Re is observed. Finally, taking oil as working fluid, 7 samples of offset strip fin cores are simulated with different fin height, fin spacing and fin length to study the effect of geometrical parameters on f and j. Also the effect of Pr on f and j is studied. Correlations for the f and j factors are derived, and these are used to analyze fluid flow and heat transfer characteristics of offset strip fins for* $19 \leq Re \leq 145$ *and* $50 \leq Pr \leq 395$. *The correlations for f and j factors can predict 100% and 86% of the CFD data for cores listed within ± 15% and ± 20% respectively.*

Introduction

The plate fin compact heat exchanger is widely used as the automobile radiators, charge air coolers, air conditioning evaporators and condensers, aircraft applications, cryogenics exchanger and electronic cooling devices for desired thermal performances, resulting to reduce space and weight, save energy and resources, ultimately the cost. The offset fins are widely used as it is characterized by high degree of surface compactness and significant heat transfer enhancement due to its periodically interrupted flow, and leading to creation of fresh boundary layers which enhances heat transfer. The present investigation is on the compact plate-fin cross flow heat exchanger which is used as oil cooler consisting of alternate stacks of offset fin and louvered fins. In this oil cooler the oil is passed over the offset fins and the forced induced atmospheric air is passed through louvered fins which act as a coolant. The offset strip fin is one of the most widely used finned surfaces, particularly in high effectiveness heat exchangers used in cryogenic, aircraft applications and automotive heat exchangers [1].

Mangik et al. [2] carried out experimental research on OSFs and developed correlations for all 3 regions and these correlations were validated with Kays and London [3]. Hu et al. [4] performed experimentation on OSFs to show effect of Prandtl number on heat transfer and pressure drop, concluded that there is significant effect on heat transfer but no effect on pressure drop. Patankar et al. [5] presented a two dimensional analysis for flow and heat transfer in an interrupted plate passage, which is idealization of the OSFs heat exchanger. Significant increase in pressure drop is observed in the thick plate situation while heat transfer doesn't sufficiently improve. Joshi et al. [6] developed an analytical model to predict heat transfer coefficient and friction factor of offset fin geometry. Flow visualization experiments were conducted to modified correlations of Weiting [7]. Michna et al. [8] investigated the effect of increasing Reynolds number on the performance of OSFs. He conducted the experiment at Reynolds number between 5000 to 120000 and found that both heat transfer and pressure drop increased with increasing Reynolds number. Dong et al. [9] made experiments and analysis to get better thermal and hydraulic performance from the OSFs. Sixteen types of OSFs and flat tube heat exchangers were used to make the experimental studies on heat transfer and pressure drop characteristics. Hong et al. [10] carried a 3-D numerical simulation. The simulation was done under laminar forced convection with an objective to study thermal performance of offset strip fin for microelectronic cooling application. Bhowmik et al. 11] studied the heat transfer and pressure drop characteristics of an offset strip fin heat exchanger and developed general correlations to compute f and j factors. These factors are used to analyse fluid flow and heat transfer characteristics of offset strip fins in the laminar, transition, and turbulent regions of the flow.

Literature review shows that most of the studies are concerned to only correlations limited to particular Prandlt number. This study considers a range of Prandtl number as $50 \leq Pr \leq 395$ for the correlations.

Description of problem and geometry

Offset fin geometry

As shown in Figure 1 offset fin consists of number of convolutions. For simplicity only one convolution is taken into consideration.

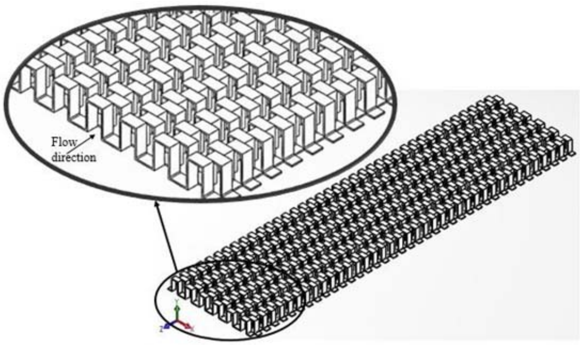

Figure 1. Three dimensional geometry of Offset Fin

Figure 2 shows the front view of the fin with notations used. The cores used to study effect of geometry on *f* and *j* are listed in Table 1. Correlations are developed based on core 1. The correlations are later checked for the use on other cores listed in Table 1. Total length of fin L is 306mm.

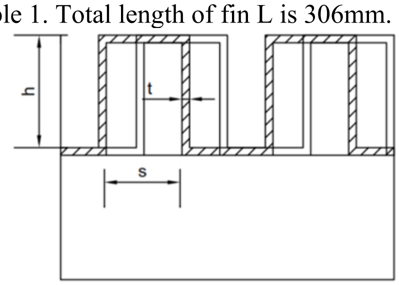

Figure 2. Schematic Diagram of Offset Fin

Table 1. Specimen geometries

Sr. No.	Geometrical Parameters (mm)			
	h	s	ls	t
1	2.26	1.52	6.12	0.152
2	1	1.52	6.12	0.152
3	1.5	1.52	6.12	0.152
4	2.26	1.52	1.2	0.152
5	2.26	1.52	3	0.152
6	2.26	1.79	6.12	0.152
7	2.26	2.35	6.12	0.152

Modelling Approach

Due to the repetition of the flow geometry, the computation domain is selected in such a way that all flow features are captured without loss of any significant information. Also this strategy helps to save the computational time. Commercial software ANSYS Fluent 16 has been used for investigation. Governing equations are discretized using finite volume method.

Following assumptions are considered for offset fin.

- Flow is steady, laminar and incompressible.
- Pressure gradient and viscous forces drive the flow.
- Gravity effect is negligible.
- Viscous dissipation is neglected.
- No-slip wall condition is applicable to all walls.
- Ideal surfaces, no burrs on offset fin.

The continuity equation,

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (1)$$

The Momentum equations,

x- momentum equation,

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{\partial p}{\partial x}+\nu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right] \quad (2)$$

y- momentum equation,

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{\partial p}{\partial y}+\nu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right] \quad (3)$$

z- momentum equation,

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{\partial p}{\partial x}+\nu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right] \quad (4)$$

The Energy equation,

$$u\frac{\partial \mathrm{T}}{\partial x}+v\frac{\partial \mathrm{T}}{\partial y}+w\frac{\partial \mathrm{T}}{\partial z}=\frac{\mathrm{k}}{\rho C_p}\left[\frac{\partial^2 \mathrm{T}}{\partial x^2}+\frac{\partial^2 \mathrm{T}}{\partial y^2}+\frac{\partial^2 \mathrm{T}}{\partial z^2}\right] \quad (5)$$

A 3D-formulation with pressure based scheme using laminar flow model for steady state condition is used. The pressure velocity coupling is done using SIMPLE algorithm. The convective effect and energy is captured by using QUICK scheme. Relaxation factors used are the default values. Convergence is ensured by setting residue in mass and velocity (10^{-4}) and energy (10^{-7}). CD 15 W/40 lubricant [12] was used as fluid, at 110°C inlet temperature the density is 851 kg/m^3, the specific heat is 2330 J/kg K, the heat conductivity is 0.1424 W/m K and the dynamic viscosity is 0.0095 kg/m s.

Boundary conditions

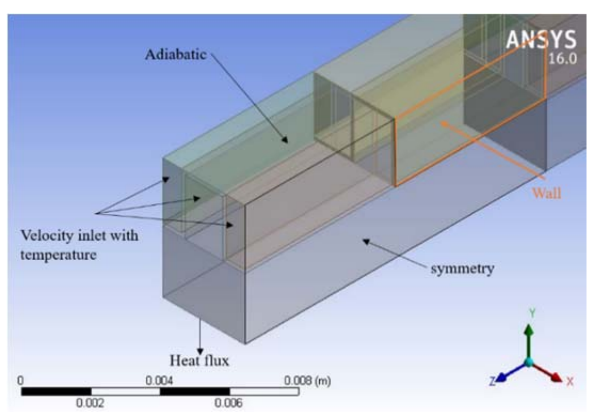

Figure 3. Boundary conditions at inlet

The inlet condition is defined by velocity inlet, temperature and outlet condition as pressure outlet. No-slip condition is considered for each wall. Use of symmetry boundary condition is done as per the geometrical similarities. The boundary conditions to the respective domain are represented in Figure 3.

Grid independence test

To ensure grid independent solution a grid sensitivity test has been carried out. Uniform grid size of mesh sizes: 0.13mm, 0.12mm, 0.11mm, 0.1mm 0.09mm and 0.08mm is selected for sensitivity test. The effect of mesh size on pressure drop is studied. A minor variation in pressure drop is observed which goes on increasing with the fineness of the grid and finally almost remains same. Thus grid size of 0.09mm is chosen for further investigation.

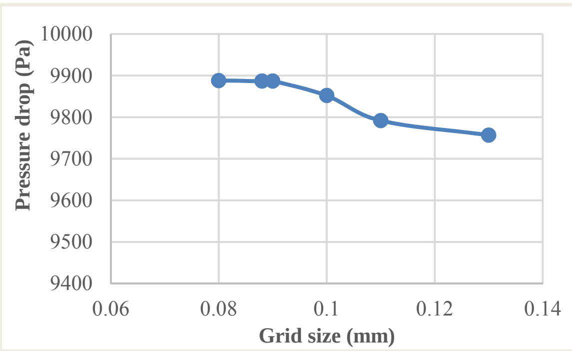

Figure 4. Grid independence test

Model validation

Experimental data [4] of the cold plate 3, are chosen for model validation. The deviation of the fluid outlet temperature between the CFD results and the experimental data is only 2.6 %. The experimental fin surface temperature coincides fairly well with the simulation results along the fin array except two ends due to conduction effect by thermal spreading to unheated end sections during experimental test [4].

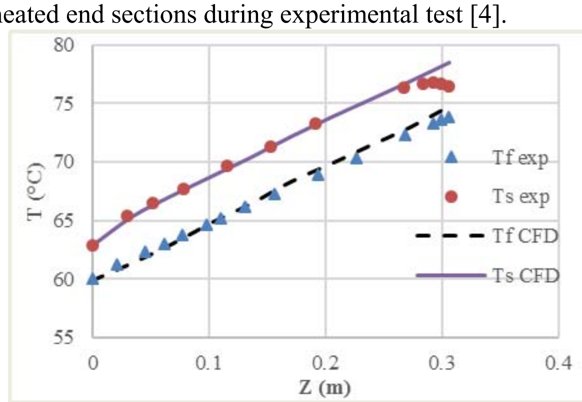

Figure 5. Comparison of fluid and fin surface temperature between CFD and experimental data at Re = 285

Data Reduction

To study the pressure drop and heat transfer in an offset fin Colburn factor and fanning friction factor are calculated using following procedure [13].

Hydraulic diameter D_h is defined in as

$$D_h = \frac{2shl_s}{sl_s + hl_s + th}$$

Reynolds number Re is evaluated as follows

$$Re = \frac{\rho u D_h}{\mu}$$

Friction factor f for fin array of length L is

$$f = \frac{\Delta P_L D_h}{2\rho L u^2}$$

The resistance of the bottom aluminum cover plate is also considered to calculate heat transfer coefficient h_o of oil.

$$\frac{1}{UA} = \frac{t_b}{A_b K_f} + \frac{1}{A h_o \eta_o}$$

U is calculated by

$$U = \frac{Q}{\Delta T_m A}$$

$$\text{Where, } \Delta T_m = \frac{(T_{f,out} - T_{o,out}) + (T_{f,in} - T_{o,in})}{2}$$

$$Q = m_o C_o (T_{o,out} - T_{o,in})$$

Overall surface efficiency η_o is

$$\eta_o = 1 - \frac{A_f}{A}(1 - \eta_f)$$

For data reduction purposes, the fin efficiency, η_f is calculated based on a one-dimensional fin model with an insulated tip.

$$\eta_f = \frac{\tanh(ml)}{ml}$$

$$\text{Where, } m = \sqrt{\frac{2h_o}{K_f t}}$$

$$l = \frac{h}{2} - t$$

h_o is used to calculate Nu

$$Nu = \frac{h_o D_h}{K_o}$$

Colburn factor j used to evaluate heat transfer is

$$j = \frac{Nu}{Re Pr^{1/3}}$$

Results and discussion

Empirical Correlations

It is observed from simulated results that variation of pressure along length is linear in nature. Using the pressure drop data from simulated results, f is calculated using data reduction method. Figure 6 shows variation of f with respect to Reynolds number.

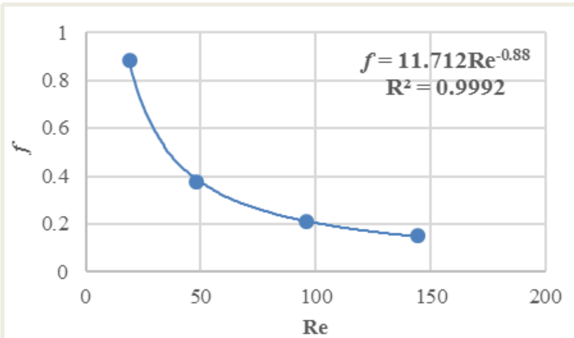

Figure 6. Correlation for Fanning's friction factor (*f*)

It is observed from simulated results that variation of temperature along length is linear in nature. It is observed that higher the velocity lesser is the temperature reduction because the fluid has less time to transfer heat. Using the temperature drop data from simulated results, j is calculated using data reduction method. Figure 7. shows variation of j with respect to Reynolds number.

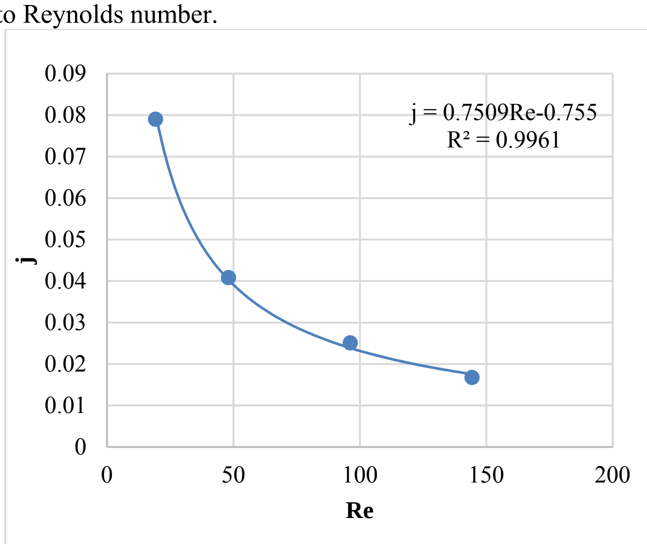

Figure 7. Correlation for Colburn's factor (*j*)

Also the nature of curves for both f and j vs Re in Figure 6 and 7 is similar to [2-4,6-7,11-13].

Effect of fin geometry on f *and* j

CFD simulations were performed on geometrical specimens listed in Table 1 to study the effect of geometrical parameters on f and j.

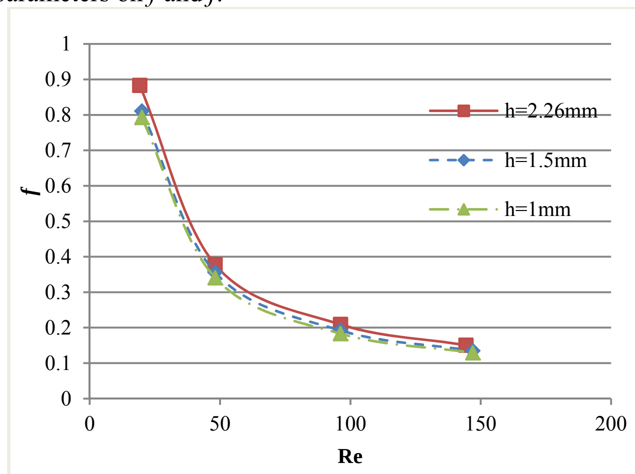

Figure 8. Effect of fin height h on the *f* factor

The effects of fin height (h) on hydraulic and thermal performance of offset strip fins are displayed in Figure 8 and 9. With the increase of fin height both the Colburn factor j and the Fanning friction factor f increase at the same Reynolds number. The reason being that, the larger heat transfer area due to higher fin height not only enhances the heat transfer performance but also leads to the more pressure loss.

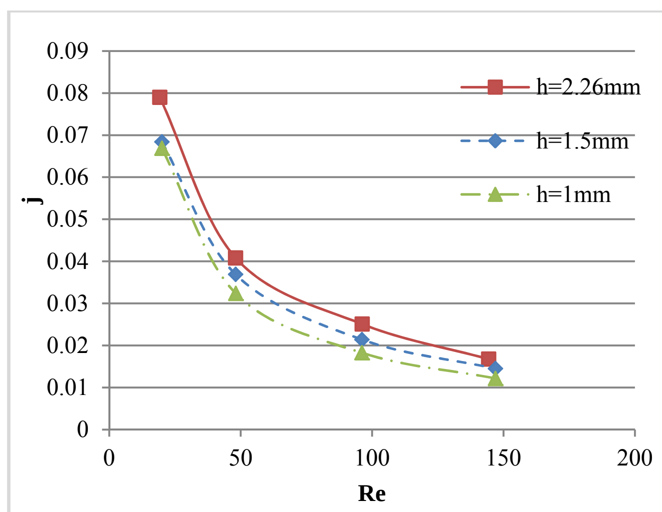

Figure 9. Effect of fin height h on the *j* factor

Figure 10 and 11 show the effects of fin spacing (s) on hydraulic and thermal performance of offset strip fins against Reynolds number. The effect of fin spacing s on the Colburn factor j is much more distinct than that on the Fanning friction factor f. With the increase of fin spacing, both the Colburn factor *j* and the Fanning friction factor *f* decrease at the same Reynolds number. The reason being that for the fixed fin length, the smaller fin spacing is, the larger are the fin number and heat transfer area. So the heat transfer is enhanced as well as the surface friction loss becomes larger.

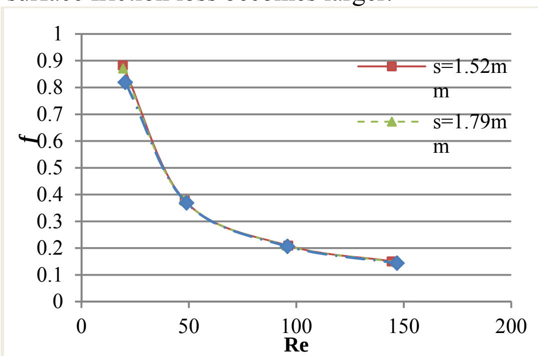

Figure 10. Effect of fin spacing s on the *f* factor

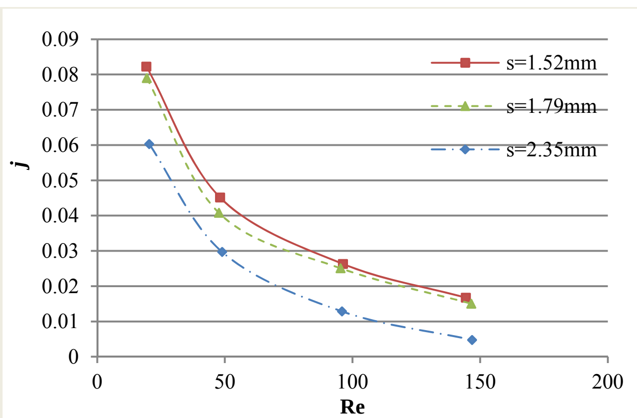

Figure 11. Effect of fin spacing s on the *j* factor

Figure 12 and 13 show the effects of fin length (ls) on hydraulic and thermal performance of offset strip fins against Reynolds number. With the increase of fin length, both the Colburn factor *j* and the Fanning friction factor *f* decrease at the same Reynolds number. The reason being that for the fixed fin length, the smaller fin length is, the larger are the fin number and heat transfer area. So the heat transfer is enhanced as well as the surface friction loss becomes larger.

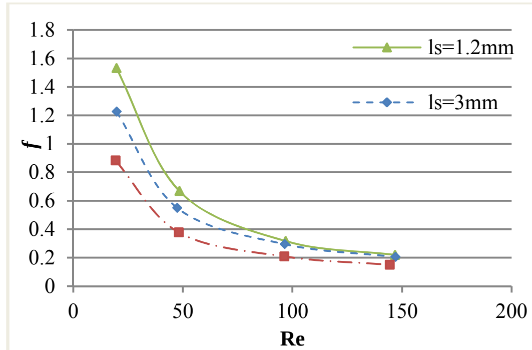

Figure 12. Effect of fin length ls on the *f* factor

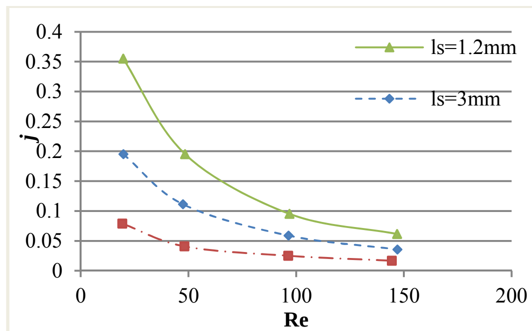

Figure 13. Effect of fin length ls on the *j* factor

Effect of Prandtl number and Reynolds number on Nusselt number

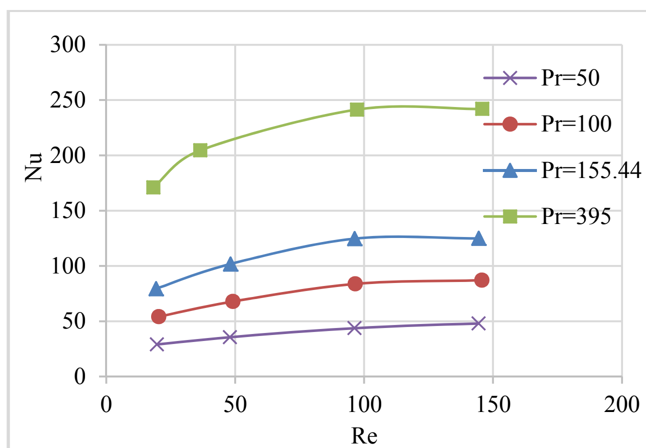

Figure 14. Comparison of predictions for *f* and *j* given by correlations with CFD data

The variations in Nu relative to Re for different values of Pr, as predicted from CFD are plotted in Figure 14. Pr is varied from 50 to 395, with a range of Re 19 to 145 covering the laminar flow region. Nu increases with increasing Re and Pr. This might have occurred because flows with higher

Prandtl numbers have longer thermal developing regions on each fin, which contributed to the attainment of a higher heat transfer rate. Therefore, the thermal field of an offset strip fin is strongly influenced by Reynolds and Prandtl numbers.

Comparison of predictions for f *and* j *given by correlations with CFD data*

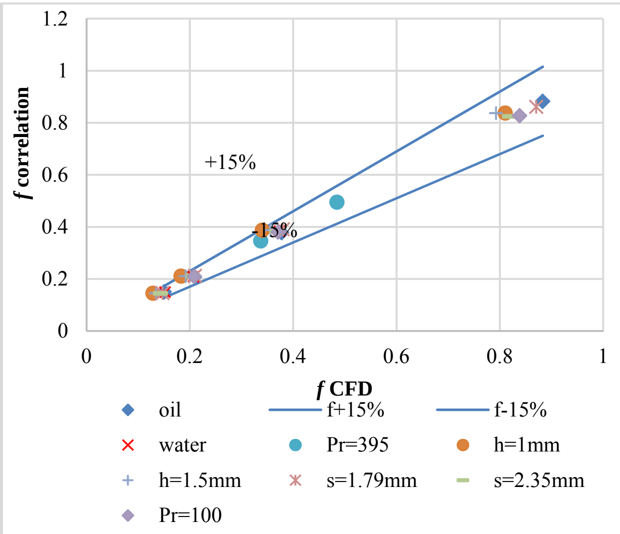

Figure 15. Comparison of predictions for *f* given by correlation with CFD data for offset strip fin cores listed in Table 1

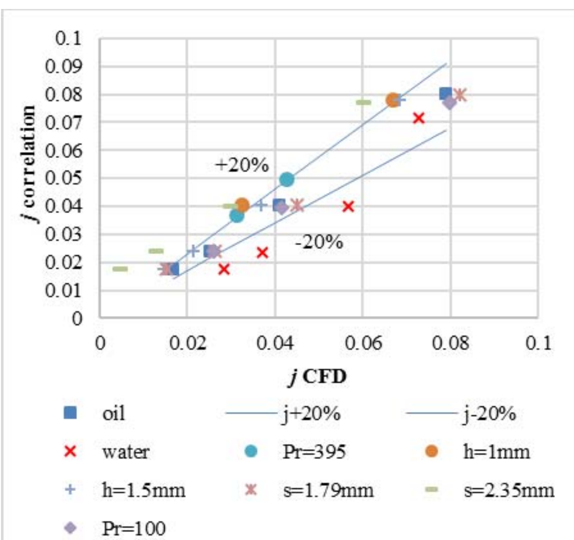

Figure 16. Comparison of predictions for *j* given by correlation with CFD data for offset strip fin cores listed in Table 1

The correlations for *f* and j factors can predict 100% and 86% of the CFD data for cores listed in Table 1 within ± 15% and ± 20% respectively, as it is seen in the scatter plots in Figure 15 and 16. More important these figures describe the similar trend to [2] in the heat transfer and friction loss behaviour of offset strip fins in laminar regime. The equations are applicable for 100≤Pr≤395. The Prandtl number was found to have a significant effect on the Colburn factor *j* of offset fin and little effect on the Fanning friction factor *f*. Also for Pr<50 in case of water from Figure 16 it is seen that deviation is higher than ± 20%, in that case correlation of *j* is not applicable but in case of *f* as seen in Figure 15 for water it is within ± 15% so *f* correlation is applicable.

Conclusion

Following conclusions can be drawn from the above analysis:

1. CFD simulation is carried out for an offset fin for 19≤Re≤145 and 50≤Pr≤395 for oil cooler application. The validity of the simulation model is verified by comparing the computed results of the offset fin with the corresponding experimental data from literature. Furthermore, data reduction method for calculation *j* factor and *f* factor is presented, based on which, the heat transfer and pressure drop characteristics in the offset fin is obtained and analyzed in detail. The correlations for *f* and *j* factors can predict 100% and 86% of the CFD data for cores listed within ± 15% and ± 20% respectively.
2. As height h increases both the Colburn factor *j* and the Fanning friction factor *f* increase at the same Reynolds number.
3. As fin length ls increases, both the Colburn factor *j* and the Fanning friction factor *f* decrease at the same Reynolds number.
4. As fin spacing s increases, both the Colburn factor *j* and the Fanning friction factor *f* decrease at the same Reynolds number.
5. Nu increases with increasing Re and Pr..

NOMENCLATURE

OSF	Offset fin	
D_h	Hydraulic diameter	m
C	Specific heat	J/kg K
K	Thermal conductivity	W/m K
h_o	Oil side heat transfer coefficient	W/m^2 K
ΔP	Pressure drop	Pa
ΔT_m	Arithmetic mean temperature difference	K
U	Overall heat transfer coefficient	W/m^2 K
s	Fin spacing	m
h	Fin height	m
ls	Offset fin length	m
L	Fin array length	m
t	Thickness of fin	m
t_b	Thickness of base plate	m
u, v, w	Velocity in x, y, z direction	m/s
T	Temperature	K
m	Mass flow rate	kg/s
η_o	Overall efficiency	
η_f	Fin efficiency	

ρ	Density	kg/m^3
μ	Dynamic viscosity	N s/m^2

Subscripts

f	Fin
o	oil
in	inlet
out	outlet
w	wall

Dimensionless

f	Fanning's Friction Factor
j	Colburn's Factor
Re	Reynolds Number
Pr	Prandtl Number
Nu	Nusselt Number

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