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Analysis of Plate Type Offset Fin Compact Heat Exchanger – A Review

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Abstract – This article is a condensed overview of research work on offset strip fin (OSF) compact heat exchangers. It features a broad discussion on the application of enhanced heat transfer surfaces to offset strip fin heat exchangers. In this paper, experimental investigation on OSF, analytical model to predict the heat transfer coefficient and the friction factor of the OSFs geometry have been discussed. Also, heat transfer and pressure drop characteristics of an OSF, 3D CFD analysis of OSF, effect of flow angle at inlet on f and j, effect of geometrical parameters on f and j, effect of burrs and roughness of fins on f and j have been discussed. The data reduction technique has been given to develop Fanning friction factor f and Colburn factor j. appropriate evaluations have been given which will be very helpful for designers to reasonably select OSF geometries and correctly conduct the thermo-hydraulic design of plate-fin heat exchangers.

Keywords— Offset Fins, Compact Heat Exchanger, Plate-Fin Heat Exchanger, Pressure Drop, Heat Transfer.

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INTRODUCTION

The high efficiency, space saving and lightweight associated with plate fin heat exchangers have led to their extensive use in aerospace, automobile industries. Cryogenic separation, liquefaction of air, natural gas processing and liquefaction, production of petrochemicals, automotive radiators are few applications. Shah [1] had discussed the classification of heat exchangers which defined the “compact heat exchangers” as one having a surface area density of more than 700 m^2/m^3. Such compactness is achieved by providing the extended surfaces. These extended surfaces on the flow passages work as the secondary heat transfer area. In a plate-fin heat exchanger, plate-fin structure is the most basic and important consideration in passage selection. Various fin surfaces are available such as plain fin, OSF, louvered, perforated fin, and wavy fin. OSF is one of the high-performance heat transfer surfaces, which becomes the main kind of fin selection. Heat transfer enhancement is obtained by periodic growth of laminar boundary layers on the fin length (l) and their dissipation in the fin wakes. The substantial heat transfer enhancement of OSF fin, however, is usually accompanied by considerable pressure drop owing to the fin offset that also results in form-drag. Therefore, evaluation of the heat transfer enhancement of OSF fins becomes a critical problem. The basis of the evaluation is a trade-off between the benefits and the corresponding penalties.

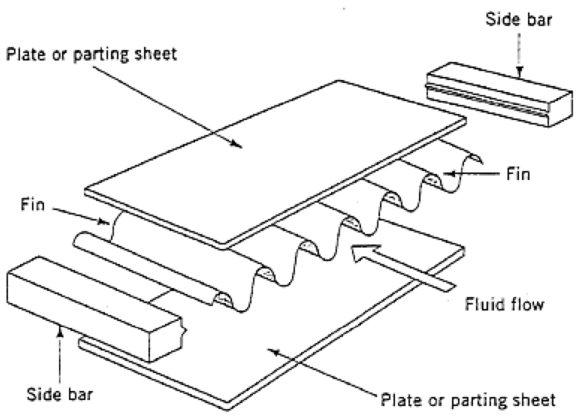

Fig. 1 Basic components of plate fin heat exchanger [2]

Plate fin heat exchangers have corrugated fins sandwiched between parallel plates or parting sheets as shown in Fig. 1.

Nomenclature					
A	Heat transfer surface area		*Greek letters*		
D_h	Hydraulic diameter	m	ρ	Density	kg/m^3
V	Velocity	m/s	μ	Dynamic viscosity	$N\text{-}s/m^2$
C	Specific heat	J/kg-K	*Subscripts*		
K	Thermal conductivity	W/m-K	f	Fin	
h_o	Oil side heat transfer coefficient	$W/m^2\text{-}K$	F	fluid	
ΔP	Pressure drop	Pa	in	inlet	
ΔT_m	Log mean temperature difference	K	out	outlet	
U	Overall heat transfer coefficient	$W/m^2\text{-}K$	w	wall	
s	Fin spacing	m	*Dimensionless*		
h	Fin height	m	f	Fanning's Friction Factor	
l	Offset fin length	m	j	Colburn's Factor	
L	Fin array length	m	Re	Reynolds Number	
t	Thickness of fin	m	Pr	Prandtl Number	
t_b	Thickness of base plate	m	Nu	Nusselt Number	
u	Velocity in x direction	m/s	η_o	Overall efficiency	
T	Temperature	K	η_f	Fin efficiency	
m	Mass flow rate	kg/s	OSF	Offset Strip fin	

The corrugations serve as both secondary heat transfer surfaces and mechanical supports for the internal pressures between layers. Fig. 3 defines the geometry of an offset fin. Fig.4 shows the different flow configurations as defined by ALPEMA and Fig.2 shows cross flow arrangement which is most commonly used.

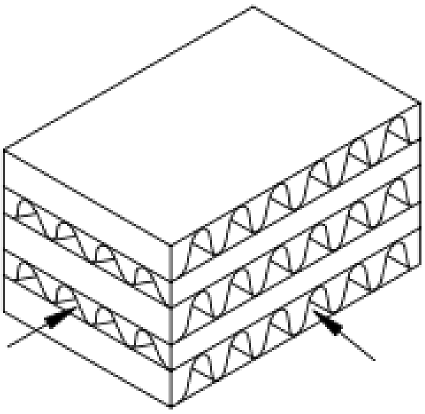

Fig. 2 Cross flow arrangement [3]

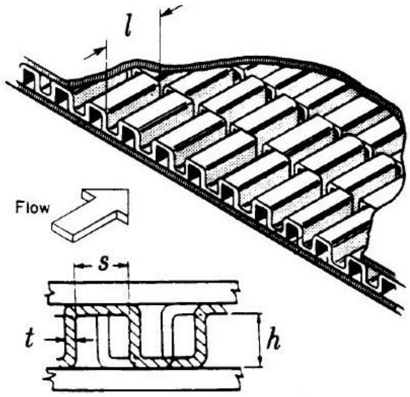

Fig. 3 Schematic Diagram of Offset Fin [4]

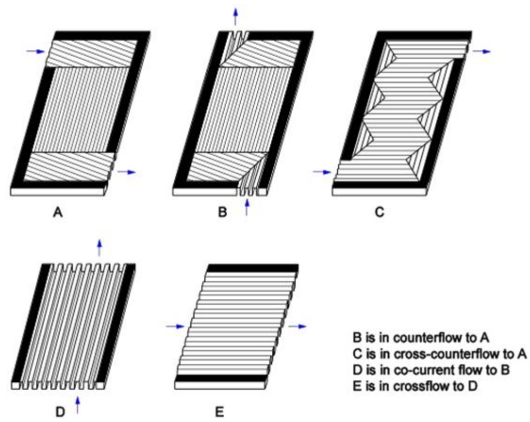

Fig. 4 Flow arrangements [5]

Definition of hydraulic diameter taken by various investigators is presented in Table 1.

Table 2 shows correlations obtained by different investigators.

TABLE V Definitions of hydraulic diameter for Offset fin

Investigator	Definition of D_h
Weiting [6]	$\frac{2sh}{s+h}$
Joshi and Webb [8]	$\frac{2(s-t)hl}{sl+hl+th}$
Manglikand Bergles [4]	$\frac{4shl}{2(sl+hl+th)+ts}$
Hu [9]	$\frac{2shl}{sl+hl+th}$

Wieting [6] had used data of 22 samples of offset strip fins from four researchers and put forward empirical correlations for their thermal-hydraulic performance. In table 2 since the correlations of Re ≤ 1000 do not contain the parameter t/D_h, it was concluded that this parameter has little or no effect on f or j within the parameter range of this correlation.

With similar reasoning, the same is concluded about s/h for Re ≥ 2000. A decrease in the value of l/D_h results in an increase in both f and j. An increase in s/h results in a decrease in both f and j. No correlation was given to predict f and j in transition region.

Patankar et al. [7] had presented a two dimensional analysis for the flow and heat transfer in an interrupted plate passage which is an idealization of the OSFs heat exchanger. The main aim of the study was investigating the effect of plate thickness in a non-dimensional form t/H (H is transverse spacing between the plates) on heat transfer and pressure drop in OSF channels because the impingement region resulting from thick plate on the leading edge and recirculating region behind the trailing edge are absent if the plate thickness is neglected. Calculation method was based on the periodically fully developed flow through one periodic module. Since the flow in OSF channels attains a periodic fully developed behavior after a short entrance region, which may extend to about 5 (at the most 10) ranks of plates. Steady and laminar flow was assumed between Reynolds numbers 100 to 2000. Flow was found to be mainly laminar in this range, although in some cases just before the Reynolds number 2000 there was a transition from laminar to turbulence. Especially for the higher values of t/H. Constant heat flow boundary condition was used. Analysis was done for different fin thickness ratios t/H= 0, 0.1, 0.2, 0.3 for the same fin length l/H = 1 and the Prandtl number of fluid was 0.7.

TABLE 2 Listing of Heat Transfer and Friction Factor Correlations for Offset Fin

Investigator	Correlations	Working Fluid
Weiting [6]	Laminar region: $Re_{D_h} \le 1000$ $f = 7.661\,(l/D_h)^{-0.384}\alpha^{-0.092}Re_{D_h}^{-0.712}$ $j = 0.483\,(l/D_h)^{-0.162}\alpha^{-0.184}Re_{D_h}^{-0.536}$ Turbulent region: $Re_{D_h} \ge 2000$ $f = 8.12\,Re_{D_h}^{-0.74}(l/D_h)^{-0.41}(\alpha)^{-0.02}$ $j = 0.53\,Re_{D_h}^{-0.5}(l/D_h)^{-0.15}(\alpha)^{-0.14}$ where α=s/h	air
Joshi and Webb [8]	Laminar region: $Re_{D_h} \le 1000$ $f = 7.661\,(l/D_h)^{-0.384}\alpha^{-0.092}Re_{D_h}^{-0.712}$ $j = 0.483\,(l/D_h)^{-0.162}\alpha^{-0.184}Re_{D_h}^{-0.536}$ Turbulent region: $Re_{D_h} \ge 2000$ $f = 1.12\,Re_{D_h}^{-0.36}(l/D_h)^{-0.65}(t/D_h)^{0.17}$ $j = 0.21\,Re_{D_h}^{-0.4}(l/D_h)^{-0.24}(t/D_h)^{0.02}$	Aqueous ethylene glycol
Manglik and Bergles [4]	Laminar region: $f = 9.6243\,Re_{D_h}^{-0.7422}\alpha^{-0.1856}\delta^{0.3053}\gamma^{-0.2659}$ $j = 0.6522\,Re_{D_h}^{-0.5403}\alpha^{-0.1541}\delta^{0.1499}\gamma^{-0.0678}$ Turbulent region: $f = 1.8699\,Re_{D_h}^{-0.7422}\alpha^{-0.0936}\delta^{0.6820}\gamma^{-0.2423}$ $j = 0.2435\,Re_{D_h}^{-0.4063}\alpha^{-0.1037}\delta^{0.1955}\gamma^{-0.1733}$ where α=s/h, δ=t/l, γ=t/s	air
Guo [10]	$j = 0.163\,Re_{D_h}^{-0.353}\left(\frac{s}{D_h}\right)^{-0.249}\left(\frac{h}{D_h}\right)^{-0.0625}\left(\frac{t}{D_h}\right)^{-0.52}\left(\frac{l}{D_h}\right)^{0.388}$ $f = 555\,Re_{D_h}^{-0.196}\left(\frac{s}{D_h}\right)^{-1.58}\left(\frac{h}{D_h}\right)^{0.868}\left(\frac{t}{D_h}\right)^{1.14}\left(\frac{l}{D_h}\right)^{0.905}$	HPD, CD 15 W/40 lubricant oil
Bhowmik [12]	$f = 10\,Re_{D_h}^{-0.68}$ $j = 0.489\,Re_{D_h}^{-0.445}$	water

For proper validation numerical results were compared with the experimental results of London and Shah for offset strip fin heat exchangers. The result indicates reasonable agreement for the f factors, but the predicted j factor is twice as large as the experimental data. It was concluded that the thick plate situation leads to significantly higher pressure drop while the heat transfer does not sufficiently improve despite the increased surface area and increased mean velocity.

Joshi et al. [8] had developed an analytical model to predict the f and j of an offset strip fin surface geometry using momentum and energy equation respectively. Flow visualization experimentation was performed to determine different flow regimes. The effect of burrs and roughness of fin in turbulent region which affects f by 4-14% and h_o by 2-5% had been obtained in this investigation.

Hu et al. [9] had presented two papers to show the effect of Prandtl number on heat transfer and pressure drop in OSF array. Experimental study was carried out in the first paper to study the effect for which they used the seven OSFs having different geometries and three working fluids with different Prandtl number. At the same time the effect of changing the Prandtl number of fluid with temperature was also investigated. The study was carried out in the range of Reynolds number varying from 10 to 2000 in both the papers. Liquid coolants used in the test were water and poly-alpha-olefin (PAO), for which the Prandtl number ranges from 3 to 150. They found that the Colburn factor for air is approximately twice the Colburn factor for the liquids at the same Reynolds number. The air model over predicts the heat transfer coefficient for liquids. The results of the two studies showed that the Prandtl number has a significant effect on heat transfer in OSF channel. Although there is no effect on the pressure drop.

Manglik et al. [4] carried out an experimental research on OSFs using air as medium. The effects of fin geometries as non-dimensional forms on heat transfer and pressure drop were investigated, for this study 18 different OSFs were used. Respective asymptotes for f and j were correlated by power law expressions in terms of Re and the dimensionless geometric parameters. After the analysis two correlations were developed, one for heat transfer and another one for pressure drop. The correlations were developed for all the three regions. Results were compared from the data obtained by other researchers in the deep laminar and fully turbulent regions. Correlations can be acceptable when comparing the results of the expressions to the experimental data obtained by Kays and London. Also it was indicated that more experiments should be done to extend the application of the correlations, especially for liquid working fluid.

Guo et al. [10] had developed empirical correlations for lubricant side heat transfer and friction characteristics of the High Pressure Direction (HPD) steel offset strip fin experimentally, for their study they used 36 different samples. The lubricant used was CD 15W/40 with Prandtl number of 140 at 90°C and low Reynolds number range of 30-500.The setup consisted of RTD and thermocouples for temperature measurement, Coriolis mass flow meter to measure mass flow rate. Constant heat flux was provided by two 2 KW electrical heater. Two measures were taken to reduce contact resistance, first by the use of high thermal conductivity paste and second using U shaped clamps. During tests, oil mass flow rate was set from 100 kg/h to 1000 kg/h. Applications of two correlations to different oil at different temperatures and media (Prandtl number) were not validated by them. The effect of fin manufacturing irregularities influencing the flow and heat transfer characteristics in actual heat exchanger had not been studied during investigation. Also correlations are limited to geometrical parameter ranges listed in the paper. Effect of geometrical parameters on f and j were studied.

Guo et al. [11] had used lubricant CD 15 W/40 as working fluid, 16 samples of offset strip fin cores were tested with different flow angles, fin height, fin wavelength and fin width. The angle between the fluid flow orientation and the fin surface is β, called the flow angle as shown in Fig. 5. The flow angles for the LPD type and the HPD type offset strip fins are respectively 0° and 90°. A general dimensionless parameter was used for judging global thermal–hydraulic performance as follows

$$JF=\frac{j/j_{ba}}{(f/f_{ba})^{1/3}}$$

where ‘‘ba’’ means bare channel without fin.

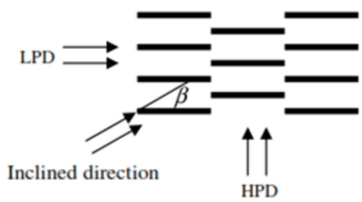

Fig. 5 Flow direction definition for offset strip fins [11]

The value of JF larger the better parameter, the enhanced heat transfer surface with larger JF factor has the better overall thermal–hydraulic characteristic. The effect of flow angle on f and j was studied and angle at which JF is maximum was also determined.

Bhowmik et al. [12] had used a steady-state, three-dimensional numerical model to study the heat transfer and pressure drop characteristics of an offset strip fin heat exchanger. Water was the heat transfer medium and the Re ranged from 10 to 3500. Fluent version 6.2 was used. A segregated solver was used to solve out the governing integral equations for conservation of mass, momentum and energy. GAMBIT was used to create a computational mesh. The finite volume technique and the semi implicit method for pressure linked equation (SIMPLE) were used to solve the basic conservation equations. A convergence criterion of 10^{-4} was selected for the continuity and momentum equations, and 10^{-8} was chosen for the energy equation. General correlations for the f and j factors have been developed and used to investigate flow and heat transfer in the laminar, turbulent and even transition regions.

A. Data Reduction Technique to Evaluate Fanning Friction Factor and Colburn Factor

To study the pressure drop and heat transfer in an offset fin, Colburn factor and fanning friction factor can be calculated using following procedure [10].

Hydraulic diameter D_h is defined in table 1. Reynolds number is calculated as follows

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

Friction factor for a length L is

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

The resistance of the bottom and top cover plate is also considered to calculate heat transfer coefficient of fluid.

$$\frac{1}{UA}=\frac{2t_b}{A_b K_f}+\frac{1}{A h_F \eta_o}$$

U is calculated by

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

Where,

$$\Delta T_m=\frac{T_{out}-T_{in}}{\ln(T_w-T_{in})-\ln(T_w-T_{out})}$$

$$Q=m_F C_F (T_{out}-T_{in})$$

Overall surface efficiency is defined as

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

The fin efficiency, η_f is calculated based on a one-dimensional fin model with an insulated tip.

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

h_F is used to calculate Nu

$$Nu = \frac{h_F D_h}{K_F}$$

Colburn factor is defined as

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

CONCLUSIONS

Following conclusion can be drawn:

1) Heat transfer and pressure drop characteristics are affected by geometrical and flow parameters.

2) Fanning friction factor and the Colburn factor decrease with the increase of Reynolds number.

3) With the increase of fin height, both the Colburn factor and the Fanning friction factor increase at the same Reynolds number.

4) With the increase of the fin length, both the Colburn factor and the Fanning friction factor increase at the same Reynolds number.

5) With the increase of fin wavelength, both the Colburn factor and the Fanning friction factor decrease at the same Reynolds number.

6) The effect of geometrical parameters is dependent on Re. There is relatively small effect due to changes in geometrical parameters in deep laminar region as compared to Re ≥ 1000.

7) When the flow angles of fins increase from 0° to 90° but with the same geometrical parameters, the j factor and the f factor increase. The thermal hydraulic performance JF of fins is the highest when the flow angle is 45°.

8) The Prandtl number has a significant effect on the Colburn factor of an offset fin arrays. Air models over predict the j factor for liquids. The Prandtl number have little effect on the Fanning friction factor.

9) Use of CFD is proved to be cost effective tool to solve these problems.

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