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Comprehensive Study of Compact Heat Exchangers with Offset Strip Fin

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Abstract

This chapter is aimed to address the performance of compact heat exchangers with offset strip fin, which have been studied by the researchers in detail for decades. The history and basic features of offset strip fins (OSF) are described first to introduce the fin geometry. Then, the effect of the fin geometry on the performance of the offset strip fin is given from experimental and numerical aspects, respectively. Flow streams evolution under varying offset strip geometries is summarized in order to demonstrate the physical impact on the flow. The thermohydraulic features of the flow in the offset strip fin are investigated by considering the Colburn j -factor and friction (f) factors in diverse flow regimes. Furthermore the criteria, flow area goodness factor j/f , the ratio $j/f^{3/3}$ and thermohydraulic performance factor JF , derived from the mentioned dimensionless factors, are also used as a scale of the performance of the structure and reported in the chapter.

Keywords: heat exchanger, offset strip fin

1. Introduction

The heat exchangers are one of the crucial components of machines since to remove the heat from the running system is the major concern, in order to improve the proper functioning. Numerous efforts have been performed by the researchers to overcome this issue by several methods and systems. The typical solution, which has been in use for decades, is the usage of heat exchangers, which are developed depending on the particular demands of the systems and have met the needs and improved the system's performance. But as in the other equipment, it came to a point where it is not sufficient enough to overcome the requirements. The compact heat exchangers emerged almost six decades back based on these particular demands. The most common definition of the compact heat exchangers in literature is "a heat exchanger having a surface area density greater than about $700\text{m}^2/\text{m}^3$ " [1]. The ultimate purpose of the

studies about the compact heat exchangers is to produce more efficient ones by reducing the physical sizes of the equipment for a given duty, which leads to use less coolant as well. There is not much possibility in order to get this goal, but one of these options is to have a higher heat transfer rate for particular conditions and the other one is to create a higher surface area and the last one is increasing both. The typical way to increase heat transfer surface area is using fins on the heat exchangers, which provide a higher surface area per unit volume ratio. The researchers endeavor to develop more efficient heat exchangers but small passage dimensions, nonuniformities and geometrical changes make it hard to characterize the heat transfer surface. The applications of the compact heat exchangers can be widely found in industry such as air conditioning, refrigeration, automotive and aerospace.

In this particular chapter, it is aimed to inform and address the offset strip fins, which have been studied by the researchers in detail for decades and still getting the attention due to its superior advantages. In the following sections, the history and fundamentals of this structure will be given first and in the following parts, the investigations will be summarized by considering their objectives, which are handled in the communications such as parametric effect of the structure, experimental and numerical research of the fin under varying flow regimes and conditions and the evolution of heat transfer and friction factors under different flow conditions by the change of the regime. The chapter will be concluded with the remarks that will outline the findings and will guide to the future studies.

2. History and basics of offset strip fin

Offset strip fin is one of the most preferred fin geometry in compact heat exchangers, which has a rectangular cross section cut into small strips of length, l and displaced by about 50% of the fin pitch in the transverse direction. The scheme of a typical strip fin is presented in **Figure 1**. The most significant variables of fin geometry are the fin thickness and strip length in the flow direction that leads to higher heat transfer coefficient and higher friction factors than plain fin geometries. The main reason of this improvement relies on developing laminar boundary layers (**Figure 2**). The enhancement is provided by the interruption of the flow

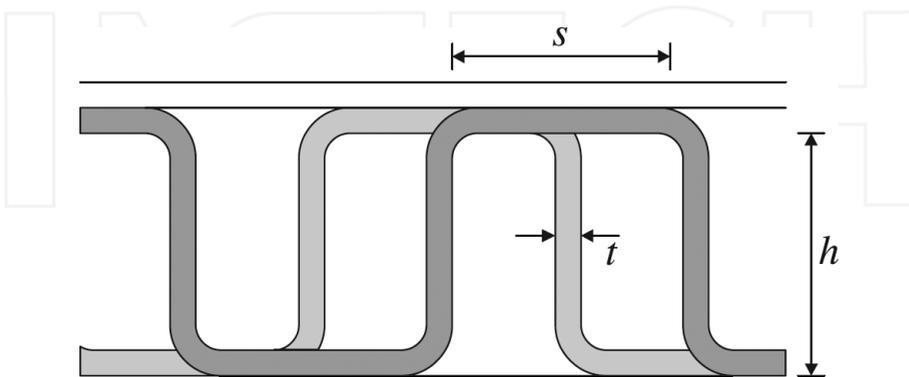


Figure 1. Offset strip fin schematic representation [2].

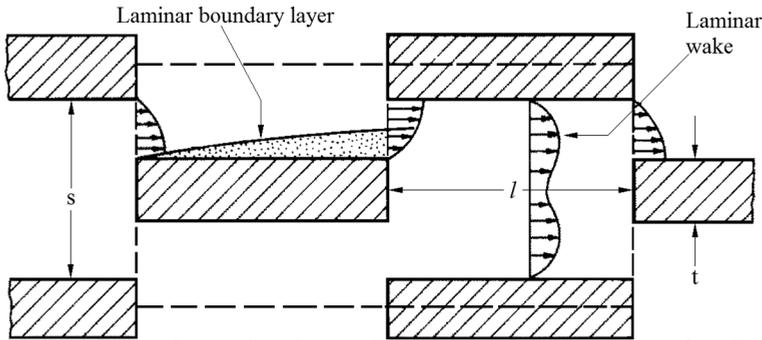


Figure 2. Flow patterns observed in the visualization experiments [3].

periodically, which does not only create fresh boundary layers but also generate greater viscous pressure drop due to higher friction factor (Figure 2).

The surface geometry in the given representation is described as follows: the fin length is l , height is h , transverse spacing is s and thickness is t . Even though nonuniform spacing is applicable and possible, generally the fin offset is the half-fin spacing and uniform. Furthermore, there are some other parameters that have influence on the flow and heat transfer like manufacturing irregularities (such as burred edges, bonding imperfections, etc.) [4].

Even though offset strip fins have been studied for decades by numerous researchers, Kays and London [5] and London and Shah [6] could be easily called as the pioneers of the offset strip fin researches by their valuable reports of their experiments about offset strip fins. The roots of their investigations rely on a test program at Stanford University in 1947 [5]. Since then, they have published their outcomes in several reports and papers, which still keep their importance to enlighten the path of researchers working in this field. In their study, they have examined several types of fins with regard to their varying parameters and operating conditions in order to explain and uncover blurry parts of the compact heat exchangers. In one of their very valuable publications [6], they shared their outcomes on the offset strip fins. The schematic representation of the fin structure is presented in Figure 3.

In that particular study, they have examined eight different fins, which differ from each by their fin numbers per inch, type of flow section, fin height, distance between plates, flow length, fin thickness and the material used in the study. Due to diverse fin type used in their studies, Kays and London [5] and London and Shah [6] have developed a coding method (explanation of the code is given in Table 1) in order to differentiate the fins from each other. An example of that coding might be given as in the following,

$$25.01.R(S) - 0.201/0.200 - 1/9(O) - 0.004(A1)$$

1 2 3 4 5 6 7 8 9

The numbers given underneath the code stands for, number of fins per inch, type of fin flow cross section (R=rectangular, T=triangular, U=U shape), fin sandwich construction

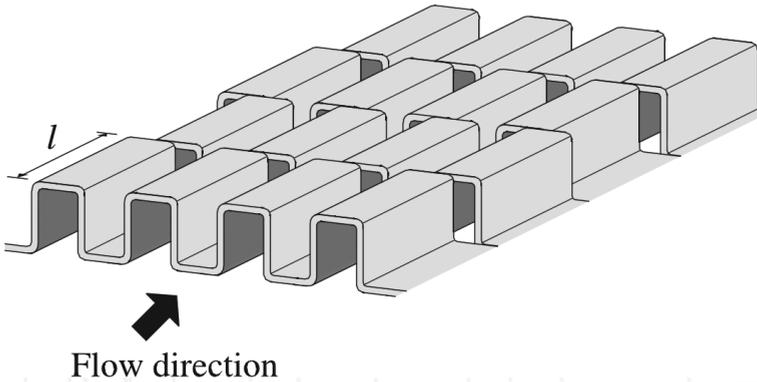


Figure 3. General features of the fin core material [6].

No	Explanation of the code
1	Number of fins per inch
2	Type of fin flow cross section
3	Fin sandwich construction
4	Fin height before brazing (inch)
5	Fin height after brazing (inch)
6	Uninterrupted fin length (inch)
7	Type of surface
8	Fin metal thickness (inch)
9	Fin material

Table 1. Explanation of codes.

(SD=single-double, S=single, D=double, T=triple), fin height before brazing, fin height after brazing, uninterrupted fin length, type of surface (L=louvered, O=offset, P=plain, S=strip), fin metal thickness, fin material (Al=aluminium, SS=stainless steel, Ni=nickel, etc.) respectively. Since their experiments are the basics of the OSF heat exchanger researches, it would be better to understand their terminology to distinct the varying geometries and structures in their investigations.

3. Data reduction of offset strip fins

The aim of the thermohydraulic analysis of the offset strip fins is to determine the pressure drop data and overall heat transfer coefficients of the structure. The pressure drop could be directly obtained from experiments, while the heat transfer rate could be found from experimental measurements by applying energy balance equations on either hot or cold streams. The typical equations used in data reduction could be derived as in the following.

In order to determine the heat transfer rate, the effectiveness- NTU method is used, which would be the ultimate purpose of the analysis.

The average heat transfer rate can be calculated by using Eq. (1)

$$\dot{Q} = (\dot{Q}_c + \dot{Q}_h)/2 \quad (1)$$

where \dot{Q}_c and \dot{Q}_h are the heat transfer rates of cold and hot stream, respectively. The heat transfer rates of each fluid can be calculated with Eq. (2) and Eq. (3)

$$\dot{Q}_c = \dot{m}_c c_{p,c} (T_{c,o} - T_{c,i}) \quad (2)$$

$$\dot{Q}_h = \dot{m}_h c_{p,h} (T_{h,i} - T_{h,o}) \quad (3)$$

The effectiveness for both unmixed fluid,

$$\varepsilon = 1 - \exp \left[\frac{NTU^{0.22}}{C_r} \{ \exp(-C_r NTU^{0.78}) - 1 \} \right] \quad (4)$$

where

$$\varepsilon = \dot{Q} / \dot{Q}_{max} \quad (5)$$

$$C_r = \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}} \quad (6)$$

With regard to these, the overall heat transfer coefficient (UA) can be obtained for the heat exchanger as,

$$UA = (\dot{m}c_p)_{min} NTU \quad (7)$$

where

$$UA = \dot{Q} / \Delta T_m \quad (8)$$

ΔT_m is the logarithmic mean temperature difference, which can be defined as,

$$\Delta T_m = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln \left((T_{h,i} - T_{c,o}) / (T_{h,o} - T_{c,i}) \right)} \quad (9)$$

The heat transfer coefficient (h_c) for the cold stream can be written as in the following,

$$\frac{1}{\eta_c A_c h_c} = \frac{1}{UA} - \frac{\delta_t}{k_t A_t} - \frac{1}{A_h h_h} \quad (10)$$

In this equation, h_h is the heat transfer coefficient of the hot stream. One of the other parameters in the given equation is the surface effectiveness (η_c) for a dry surface,

$$\eta_c = 1 - \frac{A_f}{A_c} (1 - \eta_f) \tag{11}$$

where η_f is,

$$\eta_f = \frac{\tanh(ml)}{ml} \tag{12}$$

where

$$m = \sqrt{h_c P_f / k A_{c,f}} \tag{13}$$

The flow regime is the crucial parameter of the thermohydraulic analysis of the heat exchanger studies since it has a big impact on the performance of the fin structure. Even though Reynolds number has a unique definition, it may be interpreted by the researchers from various aspects and could be written in different ways with regard to defined hydraulic diameter,

$$Re = \frac{u D_e}{\nu} \tag{14}$$

where hydraulic diameter D_e is,

$$D_e = \frac{2sh}{s + h} \tag{15}$$

As noted before, the major difference between the Re number relies on the definition of the hydraulic diameter and in order to provide a better look, some of the reported diameters and the corresponding Re numbers will be summarized in the following **Table 2**.

Reference	Hydraulic diameter	Reynolds number
[4]	$D_e = \frac{2 \cdot (s-t)h}{(s+t)+h/t}$	$Re = \frac{\rho v D_e}{\mu}$
[7]	$D_e = \frac{2 \cdot (H_f - t_f)(P_f - t_f)}{(H_f - t_f) + (P_f - t_f)}$	$Re = \frac{u D_e}{\nu}$
[8]	$D_e = \frac{4 \cdot s \cdot h_f \cdot l}{2 \cdot (s \cdot l + h_f \cdot l + h_f \cdot t) + s \cdot t}$	
[9]	$D_e = \frac{2 \cdot w \cdot L(F_f - t)}{w \cdot (L+t) + L \cdot (F_f - t)}$	
[10]	$D_e = \frac{4 \cdot s \cdot h \cdot l}{2 \cdot (s \cdot l + h \cdot l + h \cdot t) + s \cdot t}$	$Re_{D_e} = \frac{\rho u D_e}{\mu}$
[11]	$D_e = \frac{2 \cdot (s-b)l}{(l+b)}$	$Re = \frac{u_e D_e}{\nu}$

Table 2. Hydraulic diameter described in offset strip fin investigations from literature.

3.1. Heat transfer and pressure drop characteristics of offset strip fins

Like all other systems and devices, there are some nondimensional factors to evaluate the performance of the offset strip fins. The most common parameters used to decide the benefits of the structure are Colburn j -factor and friction factor (f), which corresponds to the heat

transfer and pressure drop, respectively. These two parameters could be defined as in the following,

$$j = StPr^{2/3} \tag{16}$$

$$f = \left(\frac{A_c}{A}\right) \left(\frac{2\Delta P}{\rho u^2} - k_c - k_e\right) \tag{17}$$

where A_c is the minimum free flow area for the external side and k_c and k_e are the coefficients of pressure loss at the inlet and the outlet of the heat exchanger. In addition to this, the Stanton number and the Prandtl number, which are used to define the Colburn j -factor, are,

$$St = \frac{h_c}{\rho u c_p} \tag{18}$$

$$Pr = \frac{\mu c_p}{\lambda} \tag{19}$$

where h_c is heat transfer coefficient, ρ is density, u velocity of the fluid, c_p specific heat of the fluid, μ dynamic viscosity and λ thermal conductivity. Since these two parameters are commonly preferred by the researchers in order to observe the performance of the offset strip fins, the results received at the studies with regard to these two parameters will be summarized in the coming sections where the published papers will be addressed.

One of the first friction factor and Colburn factor investigations was performed by London and Shah [6] as they were the leading investigators in this field. In the study, the effect of blockages either in steam side (the side where the steam is introduced) or in the air side (the side where the air flows across the offset fin) is underlined. The reason for the first one is explained as the poor brazing, which tends to condensation and bridge flow passage, whereas the latter one is because of bent fin edges and results in lower Colburn j -factor and higher friction factor, f . Furthermore, the effect of nondimensional parameters on j and f for different cores (different fin structures with different fin numbers) is highlighted as well. It is worth to note that, with regard to their findings when the aspect ratio gets lower, the friction factor and heat transfer are affected less and while the number of fins per unit size gets higher, they are affected more. In addition, offset spacing length is illustrated in **Figure 4**. The importance of the effect is emphasized in **Figure 4**, where increasing the fin spacing lowers the friction factor and heat transfer and vice versa and lowering the spacing makes the performance of the fin structure higher.

In addition to these most common performance criterions, new approaches are also suggested by the researchers. Bhowmik and Lee [7] adopted the j/f , $j/f^{1/3}$ and JF in order to examine the performance of the offset strip fins instead of conventional methods. j/f is known as “area goodness factor” [7] and $j/f^{1/3}$ is known as “volume goodness factor” [7]. JF number, which is related with the volume goodness factor, can be obtained by the following equation [7].

$$JF = \frac{j/j_R}{(f/f_R)^{1/3}} \tag{20}$$

where j_R and f_R are the reference values of Colburn j -factor and friction factor, respectively.

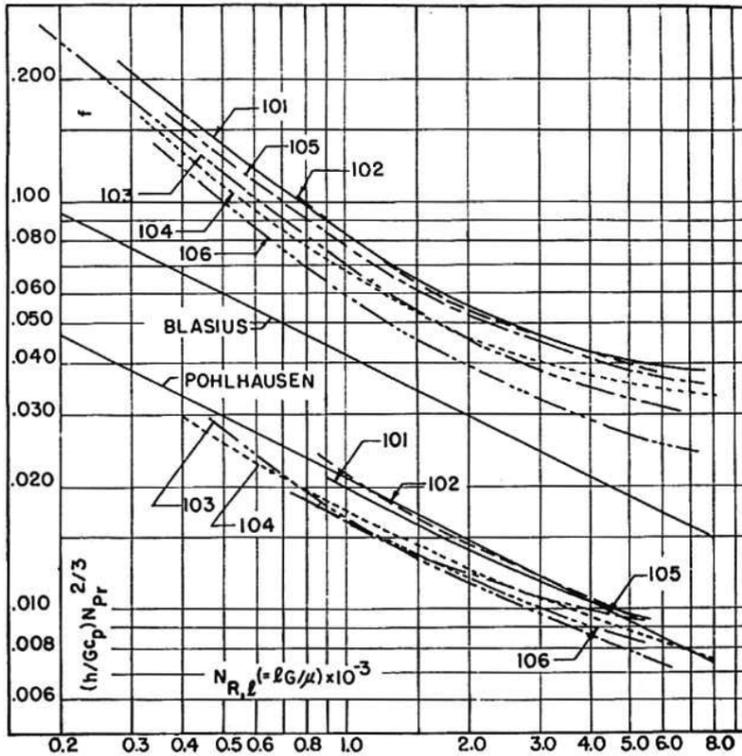


Figure 4. Offset spacing length vs. Reynolds number comparisons for varying surfaces [6].

The analyses are performed by a commercial computerized fluid dynamics (CFD) software in three dimensions. Thermohydraulic performance is studied as well for laminar, transition and turbulent flow regions, which lead to a general correlation that fits to all three regions. Since varying fluids are investigated as well, a Nu correlation including the Prandtl effect is also presented. Firstly, the numerical computations are validated by Manglik and Bergles [4] and Joshi and Webb [3] correlations and seen how they have a good agreement with them and due to their continuity, the results are claimed as a good candidate for a single correlation that covers all three flow regions.

Unlike the earlier reports, the performance of the offset strip fins are examined by flow area goodness factor j/f , the ratio $j/f^{1/3}$ and thermohydraulic performance factor JF . In these performance characteristics, a high j/f factor provides a low frontal area for the heat exchangers. When the JF factor is high, it refers to a good thermohydraulic performance and finally, high $j/f^{1/3}$ factor leads to a good heat transfer and pressure loss performance.

The effect of these three factors is observed for five Prandtl numbers ($Pr = 0.7, 7, 16, 33$ and 50), which will enable to decide which are the right working fluid conditions as shown in **Figure 5**. The Pr effect is more significant for j/f factor at the turbulent region; $j/f^{1/3}$ factor increases when

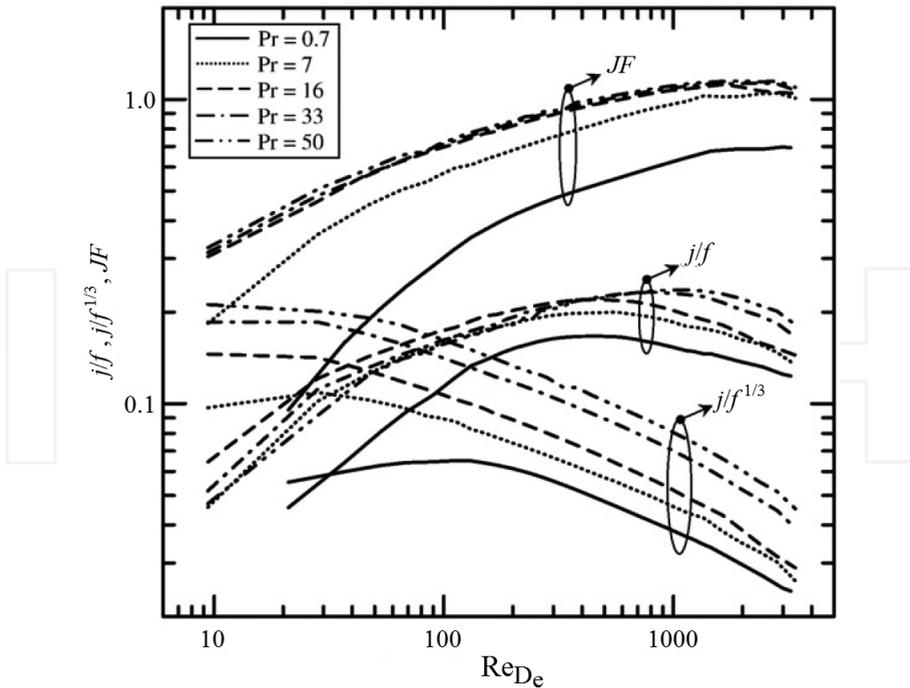


Figure 5. Evolution of performance factors [7].

the Pr ascends on the other side; unlike the other two factors, Pr does not have a distinctive effect on JF . According to these, it is underlined that JF factor could be useful for the water, while $j/f^{1/3}$ is more convenient for gas-oil liquid. In contrary to the given two factors, j/f could not be considered as a good performance criterion for fluids.

Moreover, these performance criteria are employed to provide a better comparison between seven common configurations namely plain, perforated, offset strip, louvered, wavy, vortex-generator and pin of plate-fin heat exchangers by Khoshvaght-Aliabadia et al. [8]. Water is used as the working fluid in Reynolds number range between 480 and 3770. The other purpose is to select an optimum plate-fin channel in which the best performance criteria evolution is observed. The significant enhancement is observed at the vortex-generator channel in the heat transfer coefficient and a proper reduction in the heat exchanger surface area (Figure 6). Moreover, the wavy channel displays an optimal performance at low Reynolds numbers, according to the referred criteria's. The offset strip fins, f factor, ascends by the increasing Re number as predicted but it is noted that the critical value for Re is 1800 and beyond that the friction factor rises 11.7%. As for the $j/f^{1/3}$, the offset strip fins got the highest values when the $Re > 1500$, while wavy has the highest for the low Re range. Furthermore, it is observed that the JF factor curves show the larger increment by ascending Re number for offset strip and vortex generators.

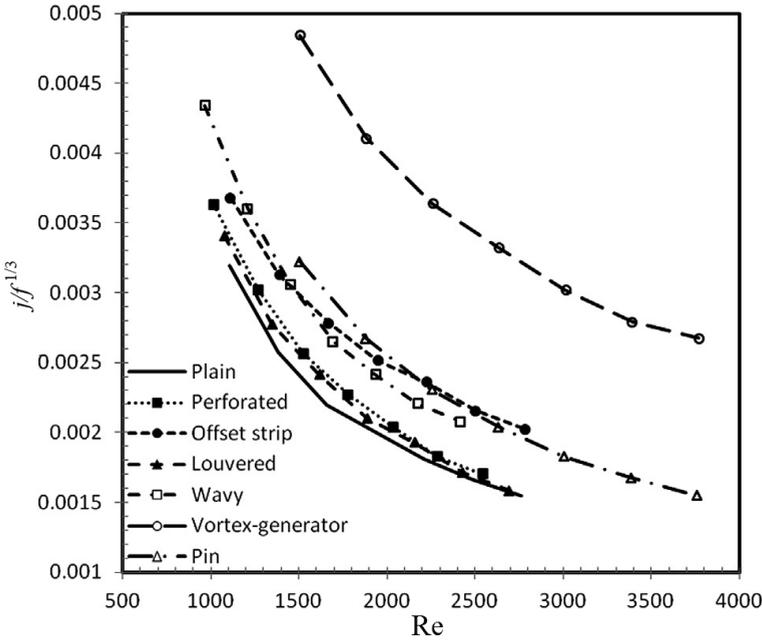


Figure 6. $j/f^{1/3}$ ratio vs. Re for different plate channels [8].

A different performance evaluation aspect is employed to the OSF heat exchangers by considering second law of thermodynamics [9]. A new parameter that is called as relative entropy generation distribution factor is proposed by the researchers. This new parameter represents a ratio of relative changes of entropy generation. The effect of parameters, which are commonly used nondimensional parameters for the OSF studies, is discussed. Optimization for the investigated parameters is carried out, which could provide sufficient information about the conditions that could maximize the performance. The proposed performance evaluation parameter bases on the entropy generation number (N_{s1}) and can be written as in the following,

$$\psi^* = \frac{(N_{s1,o,\Delta T} - N_{s1,a,\Delta T})/N_{s1,o,\Delta T}}{(N_{s1,a,\Delta P} - N_{s1,o,\Delta P})/N_{s1,o,\Delta P}} = \frac{1 - N_{s1,\Delta T}^*}{N_{s1,\Delta P}^* - 1} \tag{21}$$

where ‘a’ and ‘o’ refer to the augmented (OSF) and reference (plain plate-fin) channel, respectively and ‘ ΔT ’ and ‘ ΔP ’ stand for entropy generation due to the temperature difference and pressure drop, respectively.

The given performance is examined for different geometrical parameters in order to estimate the effective ones, by considering varying relative temperature difference. According to the obtained data, smaller thickness at lower flow rates gives better results.

4. Experimental studies for influence on flow and heat transfer

One of the fundamental research topics about offset strip fin is its geometry. The effect of the geometry on the performance of the fin and heat exchanger is the main concern of the investigators for decades. For this purpose, some researchers kept working on the effect of parameters on the performance of offset strip fin experimentally along the same path developed [5, 6]. One of these studies was released by Dong et al. [10]. Experiments are carried on air side of the heat exchanger to figure out the evolution of j and f for 16 different offset strip fins by flat tube heat exchanger. The flow of the air is ranging in $500 < Re < 7500$, while the water flow is kept constant at $2.5 \text{ m}^3/\text{h}$. The effect of fin geometry is presented for: space, height, length and flow length. In addition to the conventional approach to the fin structure, few researchers try to implement a new aspect to the studies. One of these investigations was reported by Peng et al. [11], which discusses the performance of a compact heat exchanger by an innovative offset strip fin array (**Figure 7**) at a range $Re = 500\text{--}5000$. With regard to the results, when fin pitch increases, the heat transfer reduces while the friction factor reaches to higher values. In addition, the fin length effect is presented as well, with regard to that heat transfer will be enhanced by the shorter lengths, while the friction factor increases as well. One other valuable analysis that can be pointed out is the effect of fin bending; according to them, by the increment of fin bending, the heat transfer increases and friction factor reduces.

Plate fin heat exchangers are commonly used in the air to air or air to gas applications, but not so appropriate to the liquid to liquid applications, which also have crucial importance for some applications. In order to fulfill this requirement, brazed plate heat exchangers are emerged. Commercially available ones are made up of stainless steel that is brazed by copper or nickel. Even though they meet the higher pressure and temperature conditions, they are not sufficient for the applications where corrosion issues appeared. In order to overcome this deficit, Fernández-Seara et al. [12] studied titanium brazed plate fin heat exchangers with OSF. The corrosion resistance lets titanium to be used in liquid-liquid applications, unlike the typical fin

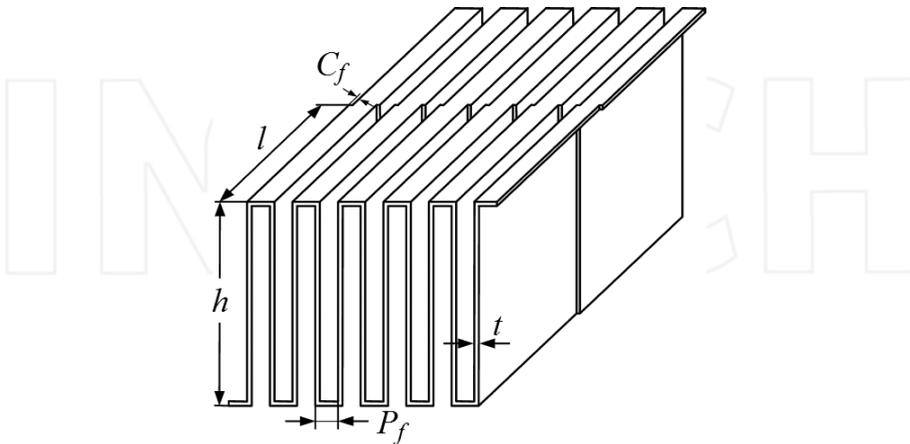


Figure 7. Scheme of the innovative offset strip fin [11].

structures. Across the heat exchanger, water-water and 10–30% ethylene glycol are used in the experiments. As a result of the investigations, empirical correlation to determine single phase convection heat transfer coefficient, as a function of Re is derived. The importance of this correlations are underlined since the experiments are carried out for water and ethylene glycol, which have higher numbers than the air and tends to provide more accurate predictions than the ones derived by air. The outcomes of the research are presented for varying Re number of water-water test and ethylene glycol mixture, individually. Pressure drop and overall heat transfer coefficients of the particular conditions are summarized with respect to the velocities or mass flow rate that they are tested. It is noted that the pressure drop gets higher as the ethylene glycol percentage increases when the flow regime drives up. In addition, the heat transfer is poor in ethylene glycol when it is compared with water and it gets worse when the mass flow reduces.

Offset strip fins are usually operated at low Reynolds numbers ($Re < 1000$), at which the flows are typically steady and laminar. Researchers have investigated the thermohydraulic performances of the offset strip fins widely at this regime. Unlike the reported studies, Michna et al. [13] tried to find the answer for the thermohydraulic analysis of the OSF at turbulent regime. The friction factor and heat transfer coefficient of the OSF are measured at Re number in the range $1000 < Re < 10,000$ in the experiments. In order to have a better approach, pressure and heat transfer tests are executed by aluminium and stainless steel, respectively, where the constant heat flux is provided by film heaters. For the confirmation of the friction factor, the correlation of Manglik and Bergles [4] (the correlation given in **Table 3**) is used, even though the correlation is for $Re < 20,000$, the correlation is extrapolated for the higher values of Re . An agreement with the correlation could be observed for friction factor at $Re < 20,000$ and it starts to oscillate beyond that range, while Colburn j -factor consistently has higher values than the correlation of Manglik and Bergles [4]. The evaluation of the modified Colburn j -factor is presented row by row for varying Re values as well. With regard to the vortex shedding and turbulent flow, the highest values of j and f are observed for the higher values of Reynolds [13].

Even though the uniform distribution is expected to obtain higher performances in compact heat exchangers, maldistribution could also be seen, especially for parallel channel heat exchangers. There are limited studies published on this topic for offset strip fins. One of those few studies is released by Saad et al. [14], in which the experimental distribution of phases and pressure drop in offset strip fin type compact heat exchangers is given. The experiments are accomplished in a flat vertical compact heat exchanger in which the flow of air and water are examined. The experiments are carried out for single phase and two phase flows by order. The single phase flow experiments are followed by CFD analysis of the same problem employed in the investigation. Furthermore, a correlation (**Table 3**) that belongs to single phase flow is developed as well for prediction of friction factor. The nonuniformity of the two phase flow regime is determined by the maldistribution parameters such as standard deviation (STD) and normalized standard deviation (NSTD). The pressure drop profiles and flow rate distribution are used to identify the phase distribution of the flow since it is hard to determine the solely phase distribution without other physical features. The results of the experimental and CFD

Reference	Correlation
[2]	$f = 9.6243Re^{-0.7422} \alpha^{-0.1856} \delta^{0.3053} \gamma^{-0.2659} \times [1 + 7.669 \times 10^{-8} Re^{4.429} \alpha^{0.920} \delta^{3.767}]$ $j = 0.6522Re^{-0.5403} \alpha^{-0.1541} \delta^{0.1499} \gamma^{-0.0678} \times [1 + 5.269 \times 10^{-5} Re^{1.340} \alpha^{0.504} \delta^{0.456} \gamma^{-1.055}]^{0.1}$
[4]	<p>Laminar range</p> $f = 8.12Re_D^{-0.74} (l/D_e)^{-0.41} \alpha^{-0.02}$ $j = 0.53Re_D^{-0.5} (l/D_e)^{-0.15} \alpha^{-0.14}$ <p>Turbulent range</p> $f = 1.12Re_D^{-0.36} (l/D_e)^{-0.65} \alpha^{0.17}$ $j = 0.21Re_D^{-0.4} (l/D_e)^{-0.24} \alpha^{-0.02}$
[6]	<p>$\beta < 20\%$</p> $f_{air} = \exp(7.91)(\alpha)^{-0.159} (\delta)^{0.358} (\gamma)^{-0.033} (Re_{D_e})^{(0.126 \ln Re_{D_e} - 2.3)}$ $j_{air} = 0.651(\alpha)^{-0.136} (\delta)^{0.236} (\gamma)^{-0.158} (Re_{D_e})^{(0.015 \ln Re_{D_e} - 0.623)}$ <p>$20\% \leq \beta < 25\%$</p> $f = \exp(9.36)(\alpha)^{-0.0025} (\delta)^{-0.0373} (\gamma)^{1.85} (Re_{D_e})^{(0.142 \ln Re_{D_e} - 2.39)}$ $j_{air} = 1.18(\alpha)^{-0.134} (\delta)^{0.0373} (\gamma)^{0.118} (Re_{D_e})^{(0.0445 \ln Re_{D_e} - 0.982)}$ <p>$25\% \leq \beta < 30\%$</p> $f = \exp(5.58)(\alpha)^{-0.36} (\delta)^{0.552} (\gamma)^{-0.521} (Re_{D_e})^{(0.111 \ln Re_{D_e} - 1.87)}$ $j_{air} = 0.49(\alpha)^{-0.23} (\delta)^{0.245} (\gamma)^{-0.733} (Re_{D_e})^{(0.049 \ln Re_{D_e} - 0.971)}$ <p>$30\% \leq \beta < 35\%$</p> $f = \exp(4.84)(\alpha)^{-0.48} (\delta)^{0.347} (\gamma)^{0.511} (Re_{D_e})^{(0.089 \ln Re_{D_e} - 1.49)}$ $j_{air} = 0.22(\alpha)^{-0.315} (\delta)^{0.235} (\gamma)^{-0.727} (Re_{D_e})^{(0.0313 \ln Re_{D_e} - 0.729)}$ <p>By Prandtl effect</p> <p>$\beta < 20\%$</p> $j_{Pr} = \exp(1.96)(\alpha)^{-0.098} (\delta)^{0.235} (\gamma)^{-0.154} (Re_{D_e})^{(0.00643 \ln Re_{D_e} - 1.3)} (Pr)^{0.00348}$ <p>$20\% \leq \beta < 25\%$</p> $j_{Pr} = 1.06(\alpha)^{-0.1} (\delta)^{0.131} (\gamma)^{-0.08} (Re_{D_e})^{(0.0323 \ln Re_{D_e} - 0.856)} (Pr)^{0.0532}$ <p>$25\% \leq \beta < 30\%$</p> $j_{Pr} = \exp(1.3)(\alpha)^{0.004} (\delta)^{0.251} (\gamma)^{0.031} (Re_{D_e})^{(0.0507 \ln Re_{D_e} - 1.07)} (Pr)^{0.051}$ <p>$30\% \leq \beta < 35\%$</p> $j_{Pr} = 0.2(\alpha)^{-0.125} (\delta)^{0.21} (\gamma)^{-0.069} (Re_{D_e})^{(0.0005 \ln Re_{D_e} - 0.338)} (Pr)^{0.0549}$
[12]	<p>$10 < Re < 3500$</p> $f_{cor} = 10Re_{D_e}^{-0.68}$ $j_{cor} = 0.489Re_{D_e}^{-0.445}$
[10]	$f = 20.362Re_D^{-0.7661}$
[15]	<p>Laminar range</p> $f = 7.6616Re_D^{-0.712} (x/D)^{-0.384} \alpha^{-0.092}$ $j = 0.483Re_D^{-0.536} (x/D)^{-0.162} \alpha^{-0.184}$ <p>Turbulent range</p> $f = 1.136Re_D^{-0.198} (x/D)^{-0.781} (t/D)^{0.534}$ $j = 0.242Re_D^{-0.368} (x/D)^{-0.322} (t/D)^{0.089}$
[16]	<p>$500 < Re < 7500$</p> $f = 2.092Re^{-0.281} \alpha^{-0.739} \beta^{0.972} \delta^{-0.78} \gamma^{-0.497}$ $j = 0.101Re^{-0.189} \alpha^{-0.488} \beta^{0.479} \delta^{-0.279} \gamma^{-0.315}$

Table 3. Heat transfer and friction factor correlations for offset strip fins.

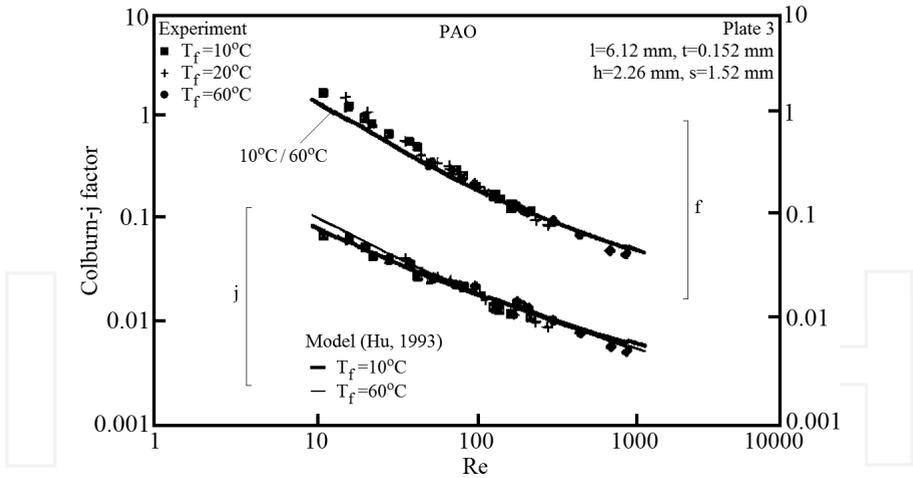


Figure 8. A comparison of test and numerical results of coolant polyalphaolefin [15].

simulation are confirmed by Manglik and Bergles [4] correlation (Table 3). A similar trend of that correlation, which is the descending of the friction factor by ascending Re number, is observed in the reported results. The correlated friction factor and the comparison of it with the reported ones are summarized and given in detail in correlations of j and f factors in OSF heat exchangers (HEX) section in order to give a better aspect to the findings.

Although the scope of the study is about OSF compact heat exchangers, which mostly deal with air, this is not the only fluid used with OSF. So, it is important to note the thermal hydraulic response of the fin structure with different Pr numbers. The effect of Prandtl is presented in a series of investigation by Hu and Herold, either experimentally [15] and numerically [16]. The experiments are corresponding to a liquid (water and polyalphaolefin) cooled offset fin compact heat exchanger. As for the given liquids, the Pr number that is used in the study ranges from 3 to 150. The obtained results compared by the air cooled models and a noticeable difference is observed for varying Prandtl values. A numerical analysis is performed to investigate uniformity of heat flux and temperature distribution and also for the validation of the tested conditions, where a good agreement could be seen (Figure 8). The model results were used to guide data reduction procedure.

5. Numerical studies for influence on flow and heat transfer

One of the leading numerical investigations about the OSF heat exchangers is carried out by Joshi and Webb [3]. The laminar and turbulent flow regimes are analytically modeled to predict the j and f factors of OSF. Moreover, the experiments are employed to visualize the flow structure, transition regime in particular. The ultimate purpose of this endeavor is to classify the regimes as the laminar, transition, or turbulent as best as it could be. The necessary equations for Nu and f are derived from energy and momentum balances. The character of

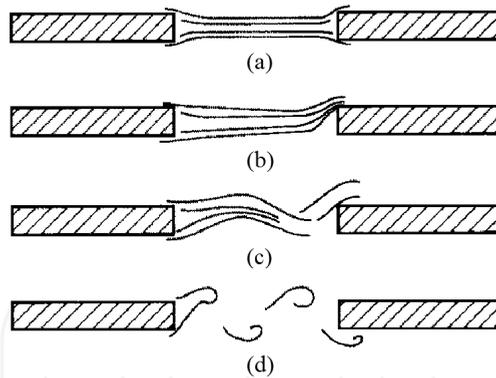


Figure 9. Flow patterns observed in visualization experiments [3].

flows on fins and in wakes is illustrated (**Figure 9**) with regard to the data attained from the numerical analysis, which are demonstrated with respect to velocity profiles. Since the effect of the fin length, thickness and spacing on the wake have not revealed in earlier communications, it is focused to find out this by either visualization or numerical analysis. The findings of the experiments are demonstrated by **Figure 9**, which starts to oscillate when the regime turns from laminar to turbulent (from (a) to (d) in **Figure 9**). Parametric study is included as well and the results of the friction factor are summarized. It is noted that when the thickness to length ratio ascends, the friction factor increases [3]. A correlation (**Table 3**) that bases on the wake width is developed to predict the Re at the transition points from the Re numbers of 21 different heat exchangers reported in literature.

Along the same path, the effect of thickness for the flow regime and performance is examined numerically [17]. And the flow pattern is investigated as presented in **Figure 10** (fin thickness ascends from top to bottom cartoon). With regard to that results when the fin is thin enough no particular difference could be observed from the low Re number. With the increment of the fin thickness recirculation zones emerge next to the plate.

Another study about the effect of parameters on the performance of the offset strip fin was reported by Kim et al. [2]. Even though the ultimate purpose of the study is to provide a better correlation for the offset strip fins at higher blockage ratios and different fluids, the parametric effect is also observed. It is remarked that the j and f factors descend when the spacing and length increases and in contrary it rises when the thickness increases. By the means of pressure drop, a similar trend could be observed with the j and f factors. It is highlighted that the thickness effect is more prominent on the performance than the spacing and length.

Even though most of the studies employed for the flow at horizontal heat exchangers, Suzuki et al. [18] operated the flow at vertical flat tubes used in the free-forced-mixed convection at low Re . The investigation has proceeded for a staggered array using numerical and experimental approaches. Three rows of flat plates are arranged in a staggered pattern where the air flows upward at a low speed.

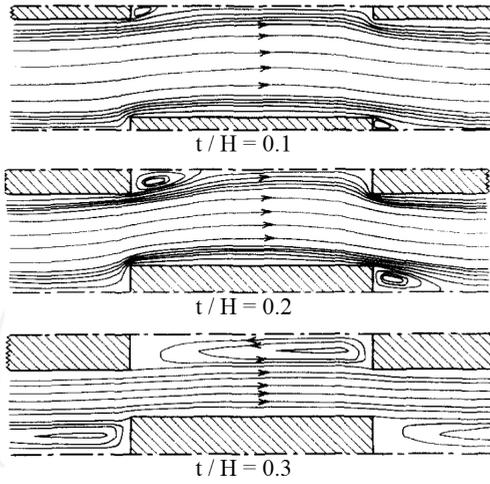


Figure 10. Flow patterns for different plate thickness [17].

In the numerical computations, two methods are used: the first one is α - ω method, the second one is U-V-P method (which solves axial and normal velocity components U and V together with pressure correction ΔP); in order to see the difference, both methods are run for the same case where it is seen they are almost identical. The effect of thickness is observed along the flow

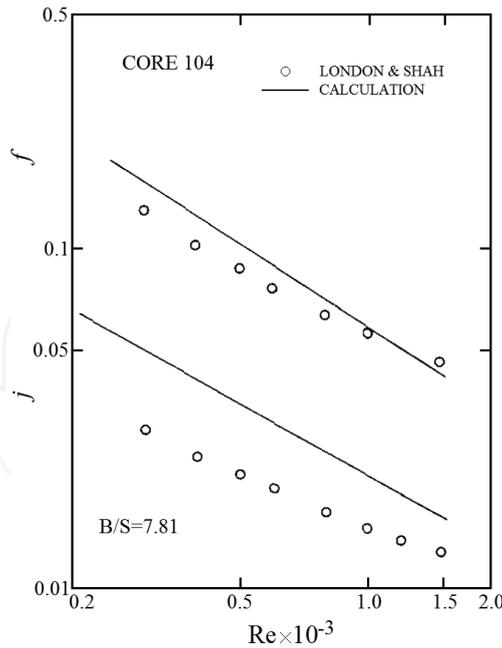


Figure 11. Comparison of the experiments by London and Shah [18].

length in the described array and any significant impact could be observed at the slower flow regimes. In all those trials, the highest heat transfer is obtained at the entrance of the array. A comparison between the published data from London and Shah [6] and the numerical data of the study is presented (**Figure 11**). It is noted that there is a good agreement between the calculated results and London and Shah for their offset geometry Core 104. Finally, it is concluded that the thicker fins are not necessary for the slower regimes since there is no distinctive effect on the performance. Even though the shorter fins give better heat transfer performance due to its higher production cost it is not efficient to replace with longer fins.

The physical phenomena for the heat transfer mechanism in offset strip fin geometries for the self-sustained oscillatory flow at high Re range is also considered by the researchers. Saidi et al. [19] noted that, due to the nature of the compact heat exchangers, in most of the studies, the flow regime is at lower Re range; in addition, in some rare applications, some researchers examine the flow at higher Re range in order to see the response at the turbulent. The flow regime at the intermediate Re range is not very common and there are unrevealed parts for this flow type. A numerical study based on finite volume method computed by a CFD code is given in the communication [20]. Friction factor and Colburn j -factor are compared by DeJong

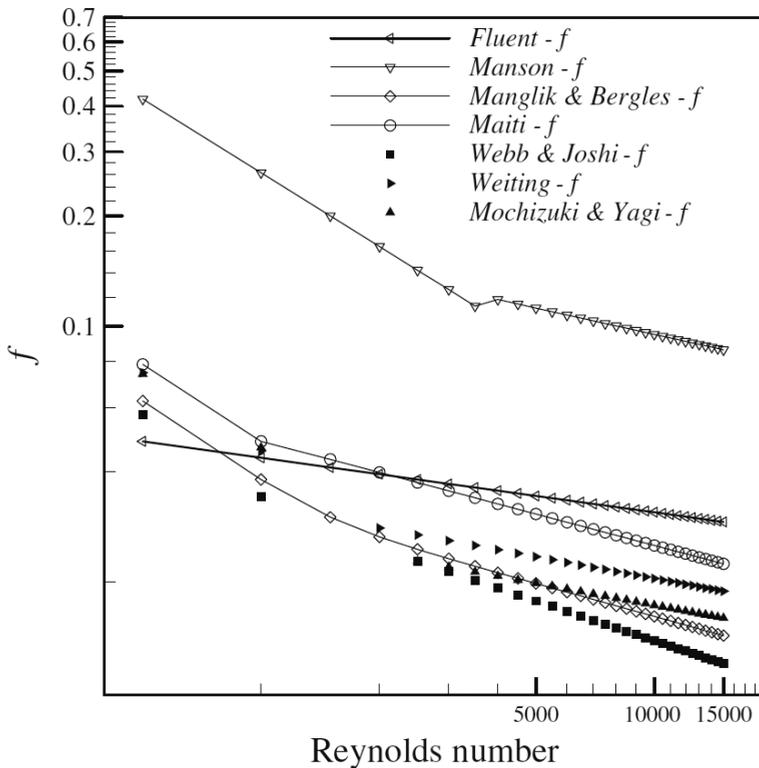


Figure 12. The comparison of Colburn j -factor of different heat transfer surface types [21].

et al. [20] for the confirmation of the findings. Then, the velocity field around the fin during a complete period of oscillation is presented by contour plots for various time steps in the sequence of development. Moreover, unsteadiness is presented in the same manner. The second moment correlation of fluctuating velocity components and second moment of temperature velocity fluctuations are demonstrated. It is noted that heat transfer and moment transfer are dissimilar.

Orientation of inlet and outlet headers plays a major role in performance of compact heat exchangers, especially in aerospace applications where the orientations are not straight. In order to understand the maldistribution at these components, Ismail et al. [21] studied these phenomena numerically. Three headers are developed to modify the distribution, which are combined with three offset strip fins and sixteen wavy fin geometries used in the heat exchangers. The computational results for wavy and offset strip fin structures are validated by analytical results and a good agreement is observed for j within -2% and for f within -9% , as it is illustrated in **Figures 12** and **13**. As for the headers, the flows are analyzed for either real (without modified headers) or ideal cases (with modified headers) and in all these conditions, the pressure drop is higher at real cases than at the ideal case. Among those, the biggest real case-ideal case difference is observed for the heat exchanger with baffle plate by 34%.

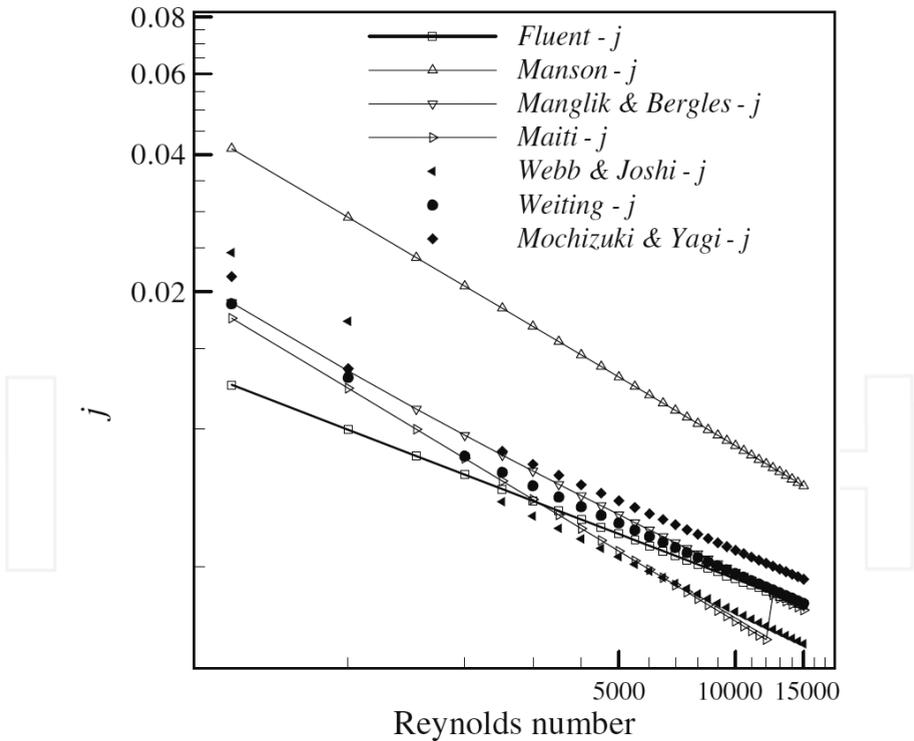


Figure 13. The comparison of friction factor of different heat transfer surface types [21].

6. Alternate applications benefit from offset strip fins

Even though the following applications are out of scope of this study, it is worth to note these in order to introduce the alternate ones to the readers who are not aware of. Solar air heater is one of these alternate systems that can transfer solar irradiant energy into airflow heat. These systems are typically used in space heating and food processing and they become popular for their cost-effectiveness and easy maintenance [22]. Like all other thermal systems, the development and research on these systems continue. In order to improve the performance of solar air heaters, researchers considered the use of offset strip fins [22–25] either numerically or experimentally. Results showed that, instantaneous thermal efficiency exceed 0.40 even at the lower air flow rates [22]. High thermal performance is obtained with low friction and as a result of all those power consumption of the fan descends [23]. In the laminar flow regime, the greatest temperature variation is recorded as 7°C at $Re = 479$ [24]. On another study, the total amount of the collected heat is 612 W/m² for an incident heat flux of 900 W/m² [25].

One of the other applications that uses offset strip fin is oil cooled heat exchangers where the need to improve heat transfer at the oil side is vital for the performance of the entire system. Due to the nature of the heat transfer medium, the fins are made up of steel instead of aluminium. The correlations for the system are derived from the experiments and compared with the experimental data; with regard to that, heat transfer is $\pm 10\%$ within the experiments with a mean deviation of 4.01%, while the friction factor can describe the experimental data within $\pm 15\%$ with a mean deviation 5.68% at low Re number [26]. Thermal hydraulic characteristics of offset strip fin are examined by considering the flow angle. Varying angles between 0 and 90° are observed and among those, the highest performance is observed at the angle 45° [27].

Ice slurry applications with its high transportation capacity by the latent heat, emerges as a promising alternate for the single phase coolant [28]. Pressure drop and heat transfer coefficient are investigated as common as in the OSF studies. But unlike the common approach, partial thermal resistance is examined as well. An empirical correlation for the Colburn j -factor is derived as well. The results revealing the instabilities of the flow may be seen due to total or partial blockages at isothermal conditions. In addition, a slight increase in the heat transfer rate and in the overall heat transfer coefficient is noted when the ice fraction increases at the flow.

Another two phase flow case is studied by some other researchers [29] for OSF heat exchangers. A vertical adiabatic channel OSF is tested for R113 (hydrofluorocarbon (trichloro-1,1,2-trifluoro-1,2,2-ethane), which is a refrigerant and now phased out in most of the developed countries) and not only pressure drop is measured but also two phase flow is visually photographed. A significant enhancement (40–50% higher) is observed for the two phase flow of R113 in OSF matrix than the conventional round tube.

7. Correlations of j and f factors in offset strip fin heat exchangers

The reliable prediction of the heat transfer and pressure drop in offset strip fin is vital and very hard to achieve. Various correlations have been developed by numerous researchers in decades to succeed and fulfill this requirement. Since most of the offset strip fin studies focus

to reveal the effect of the geometrical parameters, the large amount of data produced in the researches belong to parametric investigations. Most of the j and f correlations that could be found in literature bases on geometrical parameters as given in **Table 3**. The nondimensional parameters used in the correlations are, $\alpha = s/h$, $\delta = t/l$ and $\gamma = t/s$ where length is l , height is h , transverse spacing s and thickness t as noted before.

A particular comparison with respect to the Kays and London's experimental study [5] is taken to the following to see the differences between the approaches for the same particular problem. In the given comparison, the correlations pertain to Wieting [30], Joshi and Webb [4] and Muzychka et al. [31] are examined for a particular fin structure (uninterrupted fin length is

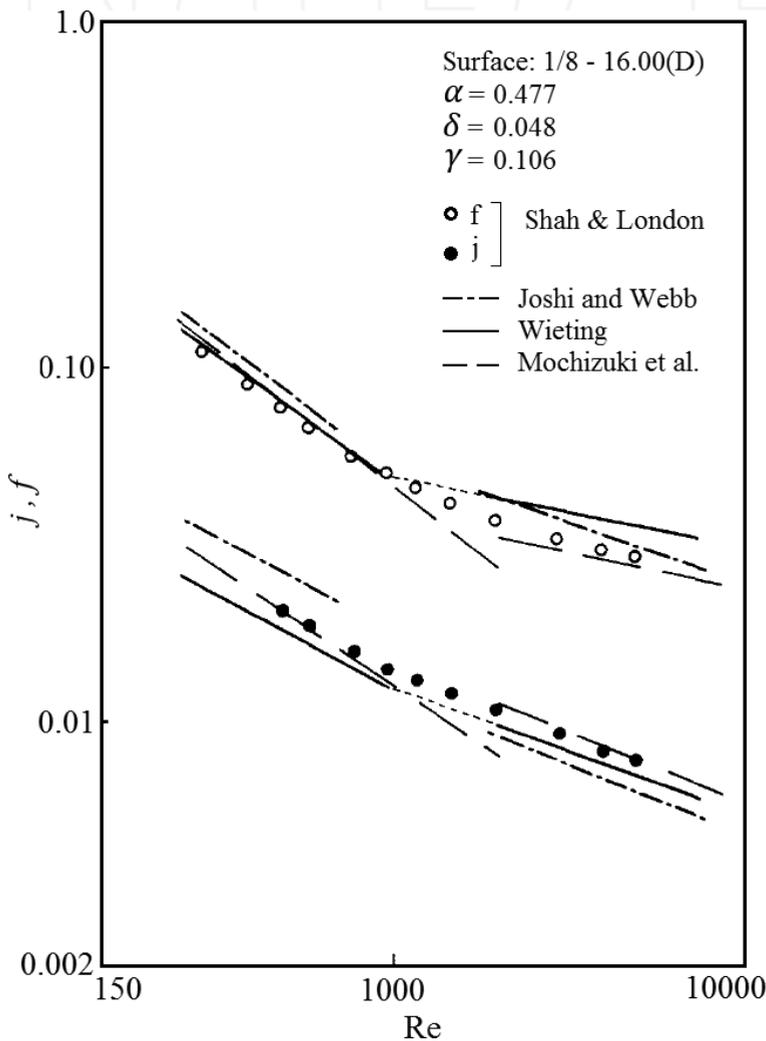


Figure 14. Comparison of j and f correlations with experimental data of Kays and London [4].

1/8 inch—number of fins per inch is 16.00 and double sandwiched fin structure) from Kays London experiments [3]. In **Figure 14**, it could be seen that, all three correlations, the most cited correlations in literature for this fin type can predict the j and f values in a good agreement, but they are not sufficient to present the transition between laminar and turbulent.

That inadequacy leads the researchers to find out a correlation, which can meet and correspond to all regimes at once. Manglik and Bergles tried to meet this with the correlation given in **Table 3**, which is claimed as giving a continuous solution for the flow regimes turning from laminar to turbulent. The calculations for the same, Kays and London problem (fin structure (1/8—16.00 D)), is also reported in the study (**Figure 15**).

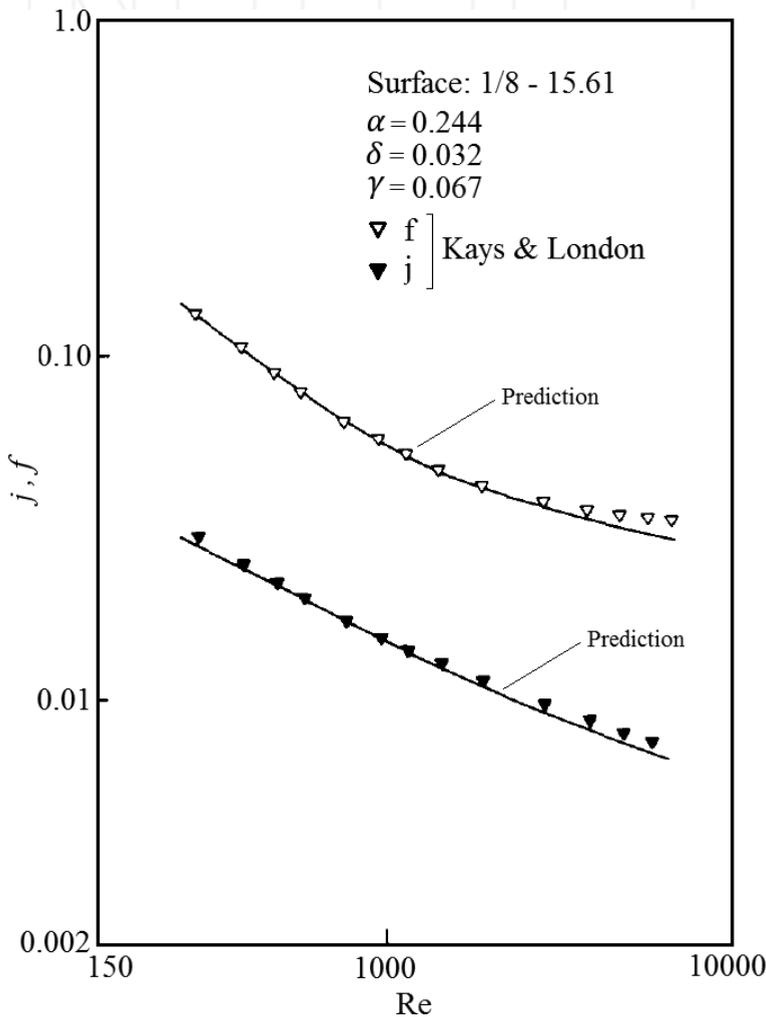


Figure 15. Comparison of correlation with experimental data of Kays and London [4].

Most of the correlations that could be met in literature are derived from the experimental studies, but unlike the common approach, some researchers try to extract the correlations from their numerical experience such as Kim et al. [2]. In the study, 39 different models are analyzed numerically by commercial CFD software where different geometric parameters and varying fluids are tried. It is also underlined that the earlier correlations are not sufficient to estimate the Colburn j -factor and friction factor when the blockage ratio (β) increases ($\beta = 12\text{--}27\%$). So there is a demand to find the answer of that question. The analysis are carried out by different turbulent models and the most suitable one is decided by comparing the findings with the most preferred correlations in literature [3, 4] and then the remaining calculations are performed to obtain a new correlation to attain the best fitting one not only for different fluids but also for different blockage ratios. The correlations are grouped as regarding to their blockage ratio (β). Since the blockage ratios below 20% are not efficient for offset strip fins and very high pressure drop could be seen beyond 35%, the blockage range investigated in the study is taken as 20–35%, the Colburn j -factor and friction f -factor can be correlated as in **Table 3**.

As mentioned earlier, the blockage ratio is not the only objective of the study [2], but the Prandtl effect is also observed by the usage of different fluids such as water; the correlations according to Prandtl can also be found as in the lower part of **Table 3**. These correlations are compared by the Manglik and Bergles for the blockage ratio of 12 and 27% in

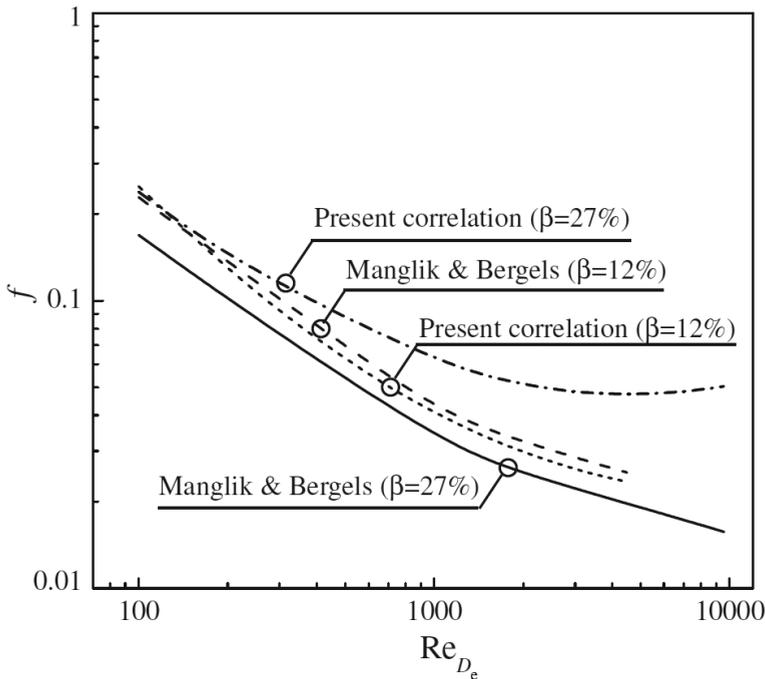


Figure 16. Comparison of Kim et al. correlation and Manglik and Bergles correlation for the blockage ratio of $\beta = 12\%$ and $\beta = 27\%$ for friction factor [2].

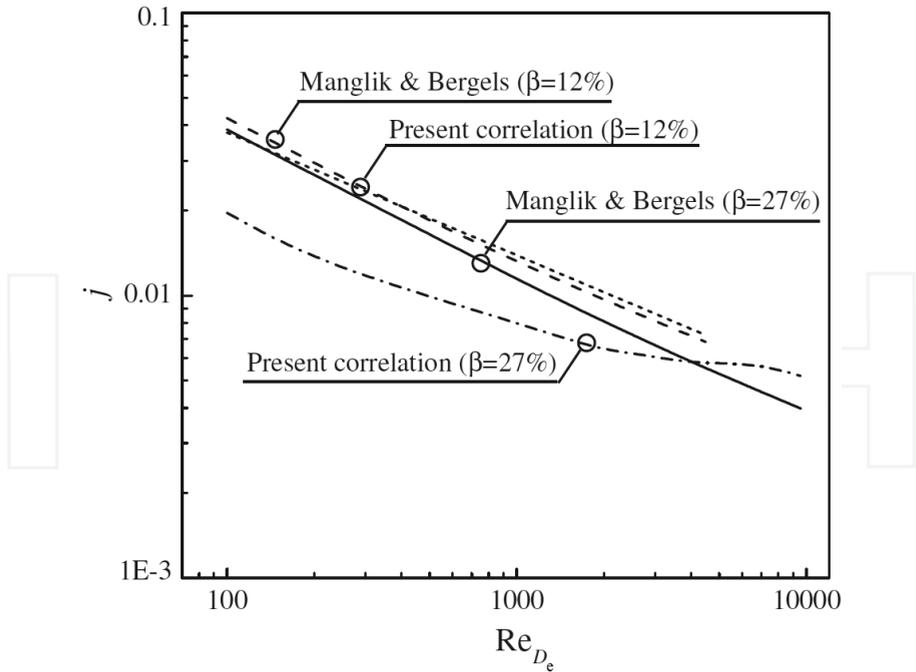


Figure 17. Comparison of Kim et al. correlation and Manglik and Bergles correlation for the blockage ratio of $\beta = 12\%$ and $\beta = 27\%$ for Colburn factor [2].

Figures 16 and 17. As it could be seen, the M&B correlation starts to overestimate when the blockage ratio is over 20%.

8. Concluding remarks

In this particular chapter, the ultimate purpose is to inform and address in detail about the studies on offset strip fins heat exchangers that have been performed by the researchers for decades and also to call a strong attention to the superior advantages of their use. The basics of this structure are reported; in addition, the investigations found in the literature are summarized by considering their objectives. The parametric effects of the structural, experimental and numerical research of the fin under varying flow regimes and conditions, the evolution of heat transfer and friction factors under different flow conditions are all given with regard to the objective of the investigations, which are cited. The outcomes that have to be highlighted at the end of this section can be listed as in the following:

- The highest heat transfer is observed at the entrance of the gate where the highest dislocations emerged.
- The shorter fin length leads to more interrupted flow, which provides a better heat transfer of this particular fin geometry.
- The Colburn factor j descends and the friction factor f ascends by the increase of the fin pitch.

- Both of these factors (j and f) increase with the decrease of fin length.
- Colburn factor j -factor decreases, while the friction factor f increases when fin bending distance decreases.
- The flow streams could be illustrated depending on the calculated data, which provide to visualize the regime evolution. In addition, the recirculation zones emerge with the increment of the thickness.

Other performance criteria derived and used for the offset strip fins are flow area goodness factor j/f , the ratio $j/f^{A/3}$ and thermal hydraulic performance factor JF . With regard to the investigations:

- JF factor could be useful for the water while $j/f^{A/3}$ is more convenient for gas-oil liquid. In contrary to the given two factors, j/f couldn't be considered as a good performance criterion for fluids.

The reliable prediction of the heat transfer and pressure drop in offset strip fin is crucial. The correlations derived in the studies are presented together in **Table 3** in order to provide a better comparison.

- With regard to the coverage, good agreement with the experimental data and mostly cited correlation pertains to Manglik and Bergles among the given equations.

Nomenclature

A_c	Heat transfer area of the cold side, m ²
$A_{c,f}$	Cross-sectional area of the fin, m ²
A_f	Frontal area, m ²
A_h	Heat transfer area of the hot side, m ²
A_t	Heat transfer area of the tube, m ²
C_f	Fin bending distance, mm
$c_{p,c}$	Specific heat of the cold stream, kJ/(kg °C)
$c_{p,h}$	Specific heat of the hot stream, kJ/(kg °C)
C_r	Heat capacity ratio
D	Diameter, mm
D_c	Hydraulic diameter, mm
f	Fanning friction factor
f_R	Reference value of the friction factor
h	Height of the fin, mm
h_c	Heat transfer coefficient of the cold (external) stream, W/(m ² °C)
h_h	Heat transfer coefficient of the hot (internal) stream, W/(m ² °C)

j	Colburn j -factor
j_R	Reference value of the Colburn j -factor
JF	Performance evaluation criteria related with volume goodness factor
k	Thermal conductivity of the material, W/(m °C)
k_c	Pressure loss coefficient at the inlet of the heat exchanger
k_e	Pressure loss coefficient at the exit of the heat exchanger
k_t	Thermal conductivity of the tube material, W/(m °C)
l	Uninterrupted fin length, mm
m	Parameter for the calculation of η_f related with $h_c, A_{c,f}, P_f$ and k, m^{-1}
\dot{m}_c	Mass flow rate of the cold stream, kg/s
\dot{m}_h	Mass flow rate of the hot stream, kg/s
N_s	Entropy generation number
Nu	Nusselt number
NTU	Number of transfer unit
$NSTD$	Normalized standard deviation
P	Pressure, Pa
P_f	Perimeter of the fin, m
Pr	Prandtl number
\dot{Q}	Average heat transfer rate, W
\dot{Q}_c	Heat transfer rate of the cold stream, W
\dot{Q}_h	Heat transfer rate of the hot stream, W
\dot{Q}_{max}	Maximum heat transfer rate, W
Re	Reynolds number
Re_{D_e}	Reynolds number based on the equivalent diameter
s	Transverse spacing, mm
t	Thickness of the fin material, mm
T	Temperature, °C
$T_{c,i}$	Inlet temperature of the cold stream, °C
$T_{c,o}$	Outlet temperature of the cold stream, °C
$T_{h,i}$	Inlet temperature of the hot stream, °C
$T_{h,o}$	Outlet temperature of the hot stream, °C
ΔT_m	Logarithmic mean temperature difference, °C
St	Stanton number
STD	Standard deviation
u	Free velocity of the external fluid, m/s
U	Overall heat transfer coefficient, W/m ² °C
UA	Overall thermal conductance, W/°C

Greek letters

α	Nondimensional parameter, s/h
β	Blockage ratio
γ	Nondimensional parameter, t/s
δ	Nondimensional parameter, t/l
δ_t	Thickness of the flat tube material, mm
ε	Effectiveness
η_c	Surface effectiveness of the cold side
η_f	Fin efficiency
μ	Dynamic viscosity, kg/ms
ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
λ	Thermal conductivity of the fluid, $W/(m\ ^\circ C)$
ψ^*	Performance evaluation criteria

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