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Numerical Investigation of Plate-fin Heat Exchanger with Wavy Channel and Wavy Vortex-generator Channel

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ABSTRACT

This paper aims to enhance the efficiency of the plate-fin heat exchanger with wavy plate fins (WPFs) and a passive heat transfer improvement technique, namely transverse vortex-generator. With the aim of enhancing efficiency in heat transfer, a new model of a wavy plate-fin channel, transversely mounted rectangular winged vortex-generator channel, is introduced and investigated by computational fluid dynamics (CFD) simulations. Comparative studies have been carried out among wavy plate-fin, wavy plate-fin with rectangular transversely mounted vortex-generator, and plain plate-fin, Reynolds number ranging from 285 to 1800 at laminar flow regime. The effect of different key parameters, including Colburn factor (j), pressure drop, and friction factor (f) along with Area goodness factor is investigated thoroughly with respect to the array length and Reynold number. The outcomes depict that values of f and j factor rise with the increment of pressure drop led by the peaks at WPFs and wings. Due to the effective blending of the streaming working fluid through the winglets, it is determined that the overall heat transfer efficiency of the winged WPFs is greater than that of the plain plate-fin and WPFs; on the other hand, Winged WPFs has a very high friction factor. At lower Reynold number, the results also depict, winged WPFs have a larger j factor value and larger area goodness factor value.

Keywords: Plate-fin heat exchanger, Rectangular Vortex-generator, CFD, Heat transfer, Wavy plate-fins

1. Introduction

Plate Fin Heat Exchangers (PFHEs) are frequently used compact heat exchangers that consist of different layers of corrugated fins and separator plate sheets. Different types of PFHEs are broadly used in automobiles, cryogenic industries, and aerospace because of their compactness (i.e., higher effective surface-area-to-volume ratio), weight, concentrated space, thermal efficiency, low energy requirement, and mostly reduced cost [1]. PFHEs are now frequently used in a wide variety of chemical industrial processes as well as other industrial functions, such as evaporators, radiators, condensers, as well as air-conditioning systems [2]. To enhance the thermal conductivity and heat transfer rate, various traditional techniques, including different shapes of the plate-fin channels, including plain, offset strip, corrugated, perforated, pin, and vortex-generator, have been used in the PFHEs[2]. Experimental investigations on different fin models were carried out earlier by Kays and London [3]. Later these experimental data formulated the basis for numerical and theoretical investigations and various empirical correlations. From the comparative studies [4], it is clear that the vortex-generator channel is an advantageous surface to gain the desired enhanced heat transfer rate in the PFHEs. For different winglet designs and angles of attack, Biswas et al. [5], [6] showed a significant advantage in enhancing the heat transfer rate of PFHEs and fin-tube heat exchangers for the longitudinal vortex-generators[7], [8]. Khoshvaght-Aliabadi[9] studied seven standard arrangements of channels used in PFHEs experimentally, and the outcome represented that vortex-generator channels are very compatible to be applied as a high-quality fluid flow interruption surface, and WPFs channels exhibited satisfactory performance in low Reynolds numbers at laminar flow regime. Wavy PFHE generally has a higher heat transfer rate, j factor, and f factor as it has a more effective surface area than plane PFHEs. Dong[10] carried out various experimental research and numerical simulations based on heat transfer and flow characteristics in wavy fin; air as the working fluid. The result showed the significant effect of waviness amplitude on the pressure drop and heat transfer of WPFs. In order to anticipate the changing pattern of the Fanning friction factor and Colburn factor for the sinusoidal wavy channels within a spectrum of Re(from 600 to 7000), Khoshvaght-Aliabadi et al.[11] examined a model using a 3D computational fluid dynamics (CFD) simulation along with a neural network. In-case of heat transfer rate improvement and other parameters, mainly f factor and j factor, vortex-generators, named as winglets, literally show outstanding performance. An experimental investigation led by Fiebig et al.[12] was the first systematic analysis of the output of various types of vortex generators (Re 1360 to 2270). Zhu et al.[13] led a 3D simulation on the heat transfer along with the laminar flow in four fundamental fins of PFHEs numerically, where apparently, the wavy fin has more j factor as well as f factor values in low values of Re(Re<1500). Zhu et al.[14] inspected flow loss and heat transfer in turbulent channel flow, considering the four shapes of longitudinal vortex generators. Among the various studies, most of them are based on as well as related to plain PFHEs with different vortex-generators and their performance. However, complete three-

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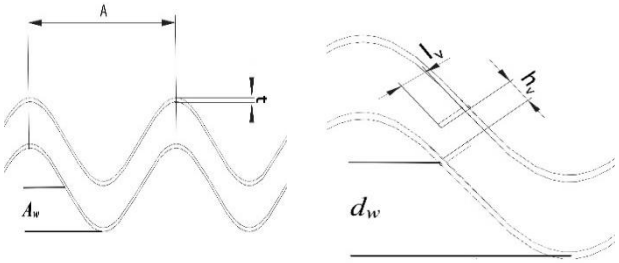
dimensional heat transfer numerical simulations of other PFHEs with enhancement techniques except plain winged PFHEs have received less attention in literature while using water as the working medium. Moreover, a 3-D numerical simulation-based analysis of the wavy PFHEs with transverse rectangular vortex-generator channels has not been accomplished previously.

The objective of the present study is to examine the heat transfer and the laminar flow in the plane PFHEs, WPFs, and WPFs with transversely positioned rectangular winged channels. The heat transfer and flow process and its performance enhancement are obtained; using 3-D numerical simulations (Computational Fluid Dynamics-CFD); illustrated and discussed carefully, focusing on the variations of the f-factor and j-factor

2. Geometric structure and numerical model

The top view of the wavy fin channel and the winged wavy channel is shown in Fig. 1. In order to have convenience in the comparative study, all the parameters of all types of fins are considered to be the same, which are provided in Table 1. The three fin models are modeled in a 3-D environment, as shown for the wavy fin channel in Fig. 2. Meshing was done with Ansys Meshing 2020 R1 Academic. The Plain fin channel model is meshed with 324968 hexahedral cells. The wavy fin channel model has 348706 hexahedral cells. And winged WPFs model has 324461 tetrahedral cells. In-case of all three models, a finer meshing is done near the wall which is the interface between the fin wall and fluid domain. In this experiment, the Reynold Number ranges from 285 to 1800, and the Prandtl number is set to 3.

Water is the working fluid. Its properties are assumed to be constant as follows:
density=983 kg/m3, specific heat Cp=4180 J/(kg K), viscosity= 4.71e-4 Pa s, thermal conductivity= 0.655 W/(m K). As the material of fin and cover plate, aluminium with thermal conductivity of 202 W/(m K) is used.
30,000 W/m^2 constant heat flux was applied to the bottom surface of the fin. The inlet velocity of the working fluid was chosen according to the Reynolds number, which is given in Table 2. Here, the inlet temperature was kept fixed at 60 °C.



(a) **(b)**

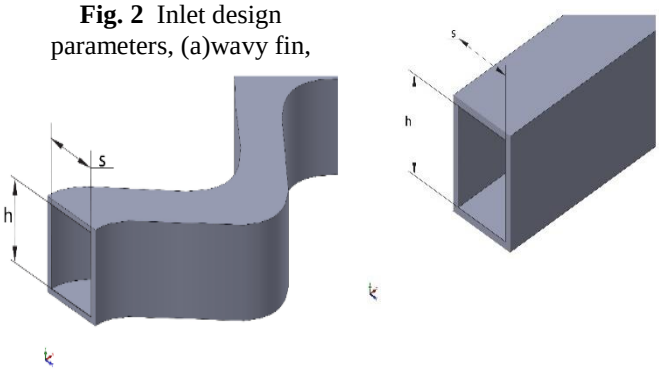
Fig. 1 2D design parameters of (a) Wavy fin, (b) Vortex-generators

Table 1 Fin geometries

Input parameters (in mm)	Terminology	Reference values(mm)
Fin thickness	t	0.152
Fin height	h	2.26
Fin spacing	s	1.52
Fin array length,	L	306
Length of winglet	l_v	0.70
Thickness of winglet	t_v	0.1
Height of winglet	h_w	0.1
Distance between winglet	h_v	15.2
Distance of winglet from the peak	d_w	1.52
Amplitude of the wavy fin	A_w	1.52
Distance between two peaks	A	7.6
Number of winglet pairs		20

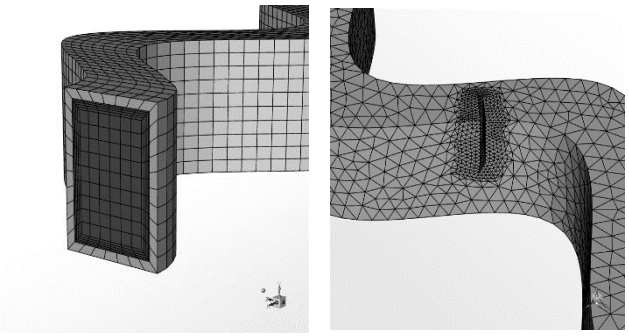
(a) **(b)**

Fig. 2 Inlet design parameters, (a)wavy fin,



(b)plain fin

In Fluent, the SIMPLE algorithm is used to have the solution. And the K-Epsilon(k-e) model is used as the turbulence model. Results are collected by doing area-weighted average on different planes across the domain.



(a) **(b)**

Fig. 3 Computational Domain**, (a)** Hexahedral mesh **(b)** Tetrahedral mesh

Table 2 Inlet constant velocities(m/s)

Re	Plain fin	WPFs	Winged WPFs
285	0.07497	0.0788	0.0788
400	0.10522	0.1104	0.1104
800	0.2104	0.2209	0.2209
1200	0.3156	0.33147	0.33147
1800	0.47352	0.4972	0.4972

3. Data Reduction

The hydraulic diameters can be obtained from Zhu et al.[13]
For plain fin,

$$D_h = \frac{2sh}{(s+h)} \tag{1}$$

For wavy fin and winged wavy fin,

$$D_h = \frac{2sh}{(s+\zeta h)} \tag{2}$$

The j factor, Reynolds number, Nusselt number :

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

$$Re = \frac{U_m D_h}{v} \tag{4}$$

$$Nu = \frac{h_c D_h}{\lambda_f} \tag{5}$$

The heat absorbed from the water flowing in the plate-fin heat exchanger can be obtained from,

$$Q_{conv} = \rho V C_p (T_{b,out} - T_{b,in}) \tag{6}$$

The convective heat transfer coefficient in the fin channel is calculated by:

$$h_c = \frac{1}{\eta_0} \frac{1}{\frac{1}{U_m} - \frac{b}{\lambda_s}\frac{A_w}{2A_{w,cl}}} \tag{7}$$

The overall heat transfer coefficient U is obtained from:

$$U_m = \frac{Q}{A_w \Delta t_m} \tag{8}$$

The log mean temperature difference(LMTD) :

$$\Delta t_m = \frac{\triangle T_{wall-b,out} - \triangle T_{wall-b,in}}{\ln(\frac{T_{wall,in} - T_{in}}{T_{wall,out} - T_{out}})} \tag{9}$$

The surface efficiency of fin channel (η_0) :

$$\eta_0 = 1 - \frac{A_2}{A_w}\left(1 - \eta_{f,id}\right) \tag{10}$$

The ideal fin efficiency ($\eta_{f,id}$) in fin channel:

$$\eta_{f,id} = \tanh(\tfrac{1}{2} m_e h) / (\tfrac{1}{2} m_e h) \tag{11}$$

$$m_e = \sqrt{\frac{2h_c}{\lambda_s t}} \tag{12}$$

Friction factor f is calculated by:

$$f = \frac{\triangle P D_h}{2\rho u^2 \mathrm{L}} \tag{13}$$

4. Model Validation and Grid Independence Test:

With the aim of verifying the reliability of CFD software (Ansys Fluent2020 R1 Academic), a rectangular plane fin channel was compared with the experimental values from the literature at Re 285, and solutions are well established, showed in Figure 4. The experimental data used here was obtained from Zhu et al.[13]. And about 7% variation is found compared to the experimental data.

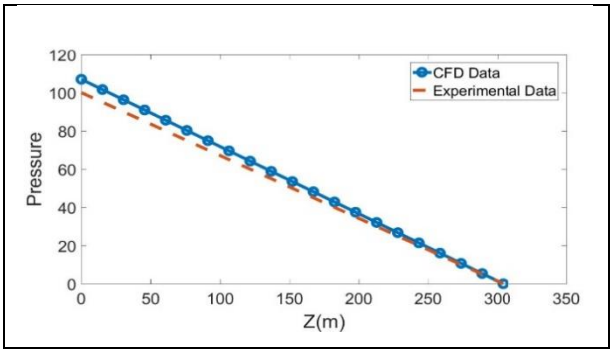


Fig. 4 Comparison of pressure change along the flow direction between experimental data from Zhu et al.[13] and CFD data

The grid independence test was conducted for wavy fin channel before other calculations at Re=400 so that the output values from the simulation becomes reliable and constant even for larger grid number.

Two types of meshing were done: hexahedral and tetrahedral. Seven sets of grid numbers were taken both for hexahedral and tetrahedral meshing and the simulation results are shown in figure 5. From figure 5, it is clear that after grid or element number 0.3 million the two curves collapse with each other and the deviation between the values are calculated 0.35% only. So to make the solution grid independent, 0.3 to 0.45million cells or elements are used in this study.

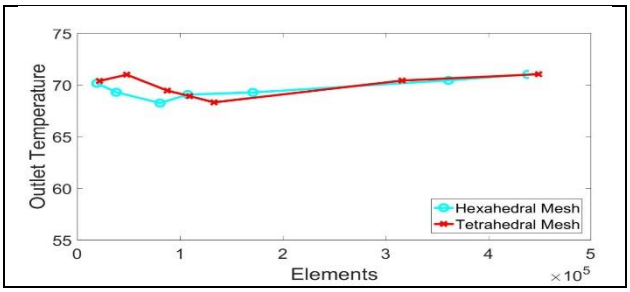


Fig.5 Change of outlet temperature with element numbers at Re 400

As both the outlet temperature values from the simulation using both hexahedral mesh and tetrahedral mesh are almost the same with less than 1% deviation, plane fin channel and wavy fin channel are meshed using hexahedral meshing and winged WPFs channel was meshed using tetrahedral meshing to avoid complexity and reduce the overall computation time.

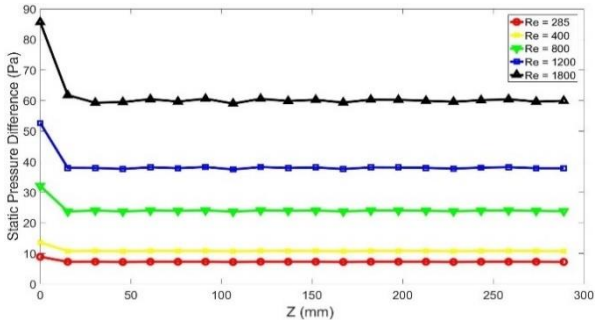
5. Results and Discussions

The performance enhancement comparison is described in terms of the static pressure drop, temperature difference, j, and f factor value change with respect to Reynold number (Re) and area goodness factor.

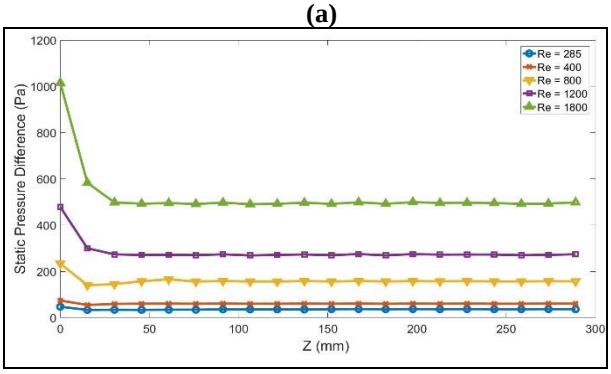
The heat transfer rate and the pressure drop are the deci ding parameters in comparing different heat exchangers. The goal of the design of the heat exchanger is to achieve a high heat transfer combined with a pressure drop as small as possible. The *j* factor or the Colburn-Chilton *j* factor is a dimensionless factor for heat transfer. From the correlations stated before, if the j-factor increases, overall heat transfer increases; which is the key factor in-case of enhancing the performance of heat exchangers.

The plain PFHE channel has the smallest j and f values, at different Re, among three fin channels which are verified in the literature [13]. The static pressure drop difference is showed in Fig. 6 for the three different fin channels at different Re. Here, it is clear from Fig. 6(a), 6(b), and 6(c) that the plain fin channel shows the lowest pressure decrease, while compared to the other two at all value of Re; As the fluid flow is not being interrupted inside plain fin channel. In-case of the other two channels, the winged wavy channel shows more than twice the pressure drop values than the traditional WPFs.

Here in-case of the wavy channel, at the inlet, the pressure remains high, which can be a result of recirculation of water due to small fin spacing and minor turbulence generated due to the corrugation-peaks of the wavy passage. The thermal entry effect is also countable. The eddy shear stress also affects the inlet pressure. But after a certain distance, the pressure drop becomes constant and also higher the Re, the constant values of pressure drop is also higher than that of a plain plate-fin as the fluid flow velocity, as well as mass-flow volume, is increased in higher Re.



(a)



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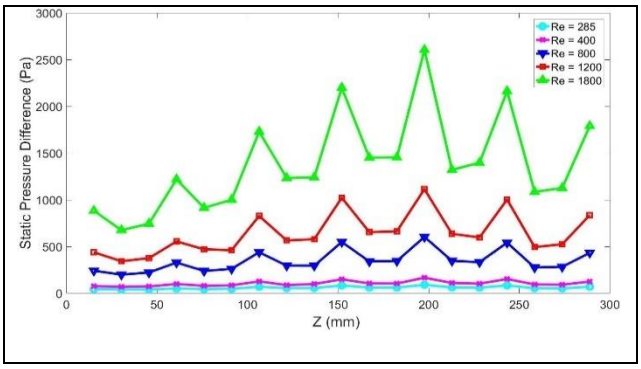
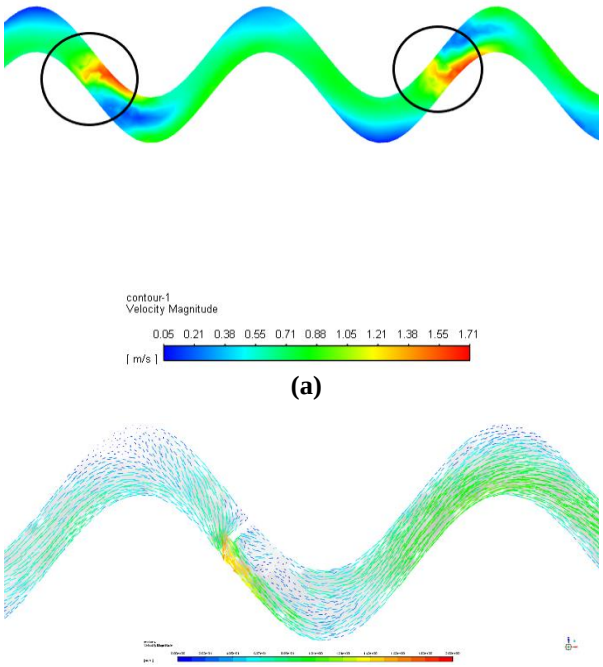


(c)

Fig. 6 Pressure drop with the array length of the three fins at different Re: (a) Plain fin, (b) Wavy fin, (c) Winged wavy fin

Fig. 6(c) depicts that for winged WPFs; in low Re(Re<400), the water flow remains almost in a common pressure drop region, which is almost the same as wavy fin shows under Re<800. But when Re≥400, the velocity increases along with mass flow rate, which is interrupted by the wall-corrugation peak, and the velocity of the water drops faster along with pressure which creates some smaller eddy.

But when it comes to the winged wavy channel, the vortex-generator winglets prevent a huge amount of flow, which creates transient turbulence shown in Fig. 7(a) and 7(b).



(a)

(b)

Fig. 7 Fluid flow interruption through vortex-generator channel due to rectangular transversely positioned winglets, (a) fin positions, (b) vector contour

Behind the fins across the flow, an eddy is created by the sudden reduced pressure which is clearly shown in figure 7(b) by the blue colored vector contours. This eddy has particular shear stress which increases the friction of the fluid and thus increasing the friction factor which should be low for heat transfer enhancement. If the friction force increases, the heat exchanger needs a higher pumping force which will eventually reduce the efficiency compared to other fin channels.

Fig. 8 shows a clear picture of the increased friction factor of the vortex-generator channel over others.

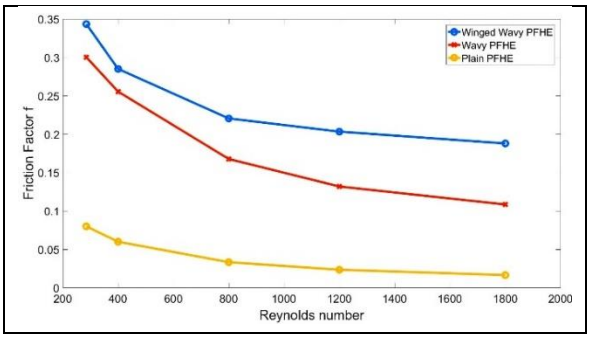


Fig. 8 Change of Friction factor f with respect to the change of Re for three fin arrays

From the Darcy-Weisbach formula if Re increases the friction factor decreases. Because, if Re increases the flow becomes turbulent from laminar. As a result, the velocity increases, and eddies created by the corrugation peak and vortex-generators become distorted and the shear-stress can not increase the friction force against the increased fluid flow.

Though the temperature values maintain a constant slow rise in upper Reynold number (Re>800)(because of the speedy stream the water cannot absorb heat properly), it fluctuates in lower Reynold number(Re≤ 285), causing a rapid instant rise in the plot shown in Fig. 9.

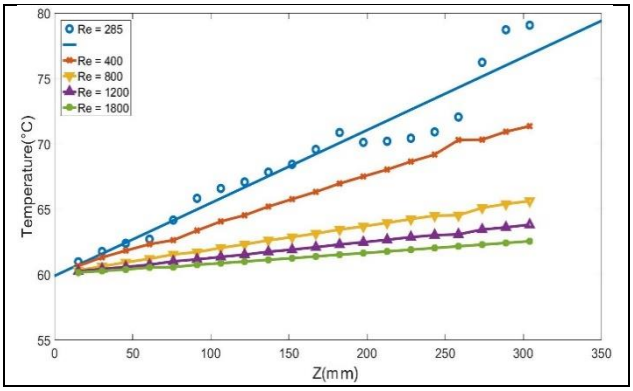


Fig. 9 Comparison of fluid temperature with the array length of Winged Wavy Fin at different Re

This phenomenon might be considered as the result of turbulence generated by the vortex-generators -slowing the water stream and letting it absorb more heat from the heated surface. Low velocity can also be assumed to be a cause for water to absorb more heat from the fins.
An enormous amount of increased value of the Colburn j-factor is also clearly noticeable from Fig. 10.

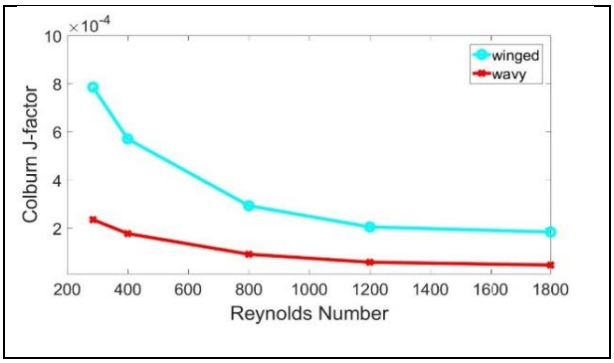


Fig. 10 Comparison of Colburn factor(j) with respect to the Reynolds Number between winged WPFs and WPFs

The figures clearly show that at low Re(Re<400), the value of j factor is too much higher (about 72.6%) for the winged WPFs than normal WPFs. The fluid is blocked and separated accordingly by the leading surfaces and tailing surfaces, At the particular

locations, resulting in an improvement in the j factor. And if the j factor value rises, the heat transfer coefficient, and heat transfer rate, also increase. In-case of Fig.10, j factor value of the winged wavy fin channel decreases exponentially with Re, which is faster than that of the wavy fin channel. It indicates that at lower values of Re (Re<400), the heat transfer rate is drastically improved which is caused by the thermal entry effect. Though the rise of fluid temperature at higher Reynold numbers is slower, due to the enhanced pressure drop, j factor increases.

Performance evaluation criterion:

Performance evaluation of a complex heat exchanger depends on different parameters like heat duty increase, initial cost, reliability, maintenance cost, compact fabrication, and so on. To avoid complexity, a single-phase comparison system is used named "Flow area goodness factor"; based on the 1st law analysis of thermodynamics. This criterion includes the j factor, f factor, Reynolds number Re, along with the geometry specification. "Area goodness factor" is used to evaluate pressure drop and heat transfer performance. Figure 11 shows the comparison between wavy and winged wavy fin channels using the area goodness factor which is simply the ratio of j and f factor.

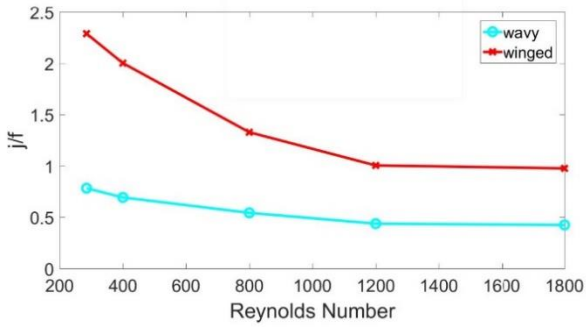


Fig. 11 Area goodness factor for WPFs and winged WPFs

From figure 11 it is crystal clear that the area goodness factor is much higher at low Re (as the mass flow velocity is lower in low Re) for both the WPFs and winged WPFs but for winged WPFs, it is higher than that of WPFs. This is a phenomenon caused by the much pressure drop and increased j factor value(due to increased heat transfer) led by the vortex-generators. With the increase of Re, the J/f ratio falls for both fin channels but for wavy WPFs, the value remains higher than the normal WPFs which is expected from this study.

Analyzing the j-factor, f-factor, and also j/f-factor plots, it can be assured that the laminar flow region or lower Reynold number is better for heat transfer inside heat exchangers as the values decline rapidly with the increase of Re which is shown in the figures clearly.

From the representation of the pressure distribution, f-factor, and j-factor in the three fin channels, and also the Area goodness factor comparison, it is crystal clear that a larger flow disruption mainly at lower Reynold numbers(laminar flow regions) always leads to a higher pressure reduction along with a higher heat transfer rate.

6. Conclusions

Comparative studies among plain PFHE, WPFs, and rectangular winged WPFs channels have been conducted, and heat transfer is carefully considered through the value change of j and f factor values. CFD simulations are considered for these three fin channels of PFHE for various Reynold number. The legitimacy is established by comparing CFD results for a plane fin channel with the correlating experimental data from the literature.

Concluding remarks are given below:

- The laminar flow region is better than the turbulence flow in-case of heat transfer optimization for all types of fin channels used in this study.
- J and f-factor values increased both in low and high Re for winged WPFs channel
- The maximum j value is 8E-4 among the three fin channels
- The maximum f value is 0.34 among the three fin channels, and the minimum f value is 0.025 showed by the plain PFHE channel.
- After the Area goodness factor comparison, the main objective of this study (to enhance heat transfer performance using a new fin channel design with vortex-generator) has been achieved. And the results show that the winged WPFs has a greater heat transfer performance(around 33% to 42.5%) than plain fin channels and wavy fin channels.

7. References

[1] L. S. Ismail and R. Velraj, "Studies on Fanning Friction (f) and Colburn (j) Factors of Offset and Wavy Fins Compact Plate Fin Heat Exchanger–A CFD Approach," Numer. Heat Transf. Part A Appl., vol. 56, no. 12, pp. 987–1005, Dec. 2009, doi: 10.1080/10407780903507957.

[2] Z. Wang and Y. Li, "Layer pattern thermal design and optimization for multistream plate-fin heat exchangers - A review," *Renewable and Sustainable Energy Reviews*. 2016, doi: 10.1016/j.rser.2015.09.003.

[3] W. M. Kays and A. L. London, *Compact Heat Exchangers*, vol. null.

1984.
[4] M. Khoshvaght-Aliabadi, S. Zangouei, and F. Hormozi, *Performance of a plate-fin heat exchanger with vortex-generator channels: 3D-CFD simulation and experimental validation*, vol. 88. 2015.
[5] G. Biswas, K. Torii, D. Fujii, and K. Nishino, "Numerical and experimental determination of flow structure and heat transfer effects of longitudinal vortices in a channel flow," *Int. J. Heat Mass Transf.*, vol. 39, no. 16, pp. 3441–3451, 1996, doi: 10.1016/0017-9310(95)00398-3.
[6] P. Deb, G. Biswas, and N. K. Mitra, "Heat transfer and flow structure in laminar and turbulent flows in a rectangular channel with longitudinal vortices," *Int. J. Heat Mass Transf.*, vol. 38, no. 13, pp. 2427–2444, 1995, doi: 10.1016/0017-9310(94)00357-2.
[7] A. Jain, G. Biswas, and D. Maurya, "Winglet-type vortex generators with common-flow-up configuration for fin-tube heat exchangers," *Numer. Heat Transf. Part A Appl.*, vol. 43, no. 2, pp. 201–219, 2003, doi: 10.1080/10407780307325.
[8] G. Biswas, N. K. Mitra, and M. Fiebig, "Heat transfer enhancement in fin-tube heat exchangers by winglet type vortex generators," *Int. J. Heat Mass Transf.*, vol. 37, no. 2, pp. 283–291, 1994, doi: 10.1016/0017-9310(94)90099-X.
[9] M. Khoshvaght-Aliabadi, F. Hormozi, and A. Zamzamian, "Role of channel shape on performance of plate-fin heat exchangers: Experimental assessment," *Int. J. Therm. Sci.*, 2014, doi: 10.1016/j.ijthermalsci.2014.01.004.
[10] D. Junqi, C. Jiangping, C. Zhijiu, Z. Yimin, and Z. Wenfeng, "Heat transfer and pressure drop correlations for the wavy fin and flat tube heat exchangers," *Appl. Therm. Eng.*, 2007, doi: 10.1016/j.applthermaleng.2006.11.012.
[11] M. K. Aliabadi, M. G. Samani, F. Hormozi, and A. H. Asl, "3D-CFD simulation and neural network model for the j and f factors of the wavy fin-and-flat tube heat exchangers," *Brazilian J. Chem. Eng.*, 2011, doi: 10.1590/S0104-66322011000300016.
[12] M. Fiebig, P. Kallweit, N. Mitra, and S. Tiggelbeck, "Heat transfer enhancement and drag by longitudinal vortex generators in channel flow," *Exp. Therm. Fluid Sci.*, 1991, doi: 10.1016/0894-1777(91)90024-L.
[13] Y. Zhu and Y. Li, *Three-dimensional numerical simulation on the laminar flow and heat transfer in four basic fins of plate-fin heat exchangers*, vol. 130, no. 11. 2008.
[14] J. X. Zhu, N. K. Mitra, and M. Fiebig, "Effects of longitudinal vortex generators on heat transfer and flow loss in turbulent channel flows," *Int. J. Heat Mass Transf.*, 1993, doi: 10.1016/S0017-9310(05)80118-8.

NOMENCLATURE

C_p	: specific heat at constant pressure
A	: distance between the sine wave peaks
A_F	: total fin area (m^2)
D_h	: hydraulic diameters, m
$T_{b,out}$, $T_{b,in}$	: Bulk input, output temperature K
T_{wall}	: fin wall temperature K
h_c	: heat transfer coefficient
U_m	: overall heat transfer coefficient
b	:base plate thickness(m)
A_w	: total heat transfer area of fin channel, (m^2)
$A_{w,cl}$	: wall area of the clipboard in fin channel,
u	: velocity(ms^{-1})
λ_s, λ_f	: thermal conductivity of surface, fluid
A_2	: secondary heat transfer area in fin channel, (m^2)
ζ	: wavy fin passage length to wavy fin array length ratio
Nu	: Nusselt number
Pr	: Prandtl number
J	: Colburn factor
U_m, U	: Overall heat transfer coefficient
v	: fluid velocity
ρ	: density
V	: volumetric flow rate