Study of thermal–hydraulic performance of fin-and-oval tube compact heat exchangers with corrugated fin patterns

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CERTIFICATE OF RESEARCH

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Declaration

This is to certify that the work presented in this thesis is an outcome of the analysis, simulation & research carried out by the author themselves under the watchful supervision of Dr. Arafat Ahmed Bhuiyan.

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List of Acronyms

A	-	Total heat transfer surface area [m2]
A_c	-	Minimum cross-sectional [m2]
A_o	-	Total surface area [m2]
C_p	-	Specific heat [J/kg K]
D_c	-	Tube outside diameter [m]
D_o	-	Fin collar outside diameter [m]
D_h	-	Hydraulic diameter [m]
f	-	Friction factor
F	-	Pumping power factor
F_1	-	Fin length [m]
F_w	-	Fin width [m]
F_p	-	Fin pitch [m]
h	-	Heat transfer coefficient [W/m2 K]
j	-	Colburn factor
J	-	Heat transfer performance factor
JF	-	Overall thermal hydraulic performance
k	-	Thermal conductivity [W/m K]
N	-	Number of tube row
Nu	-	Nusselt number (h Dh/k)
m_f	-	Mass flow rate [kg/s]
D_p	-	Pressure drop [pa]
P_1	-	Longitudinal pitch [m]
Pr	-	Prandtl number
P_t	-	Transverse pitch [m]
Q	-	Heat transfer rate [w]
X_1	-	Tube position from inlet [m]
Re	-	Reynolds number ((q Um Dh)/l)
Re_{Dc}	-	Reynolds number based on Dc
Re_{Dh}	-	Reynolds number based on Dh
$S_{\emptyset}$	-	Volumetric source term

T	-	Temperature [K]
U_m	-	Mean velocity at the minimum flow cross-sectional[ms^{-1}]

Greek letters

δ_f	-	Fin thickness [m]
μ	-	Dynamic viscosity [kg/ms]
ρ	-	Density [kg/m^3]
Γ	-	Diffusion coefficient
ϕ	-	Transport property

Subscript

in	-	Inlet
m	-	Mean
out	-	Outlet
w	-	Wall

Abbreviations

LVGs	-	Longitudinal vortex generators
RVGs	-	Rectangular vortex generators
VGs	-	Vortex generators
CFOT	-	Corrugated fin and oval tube
CHE	-	Compact heat exchanger
FTCHE	-	Fin-and-tube compact heat exchanger
CVGPF	-	Corrugated/vortex-generator plate-fin
OC	-	Fins one-corrugated fins
TC	-	Fins three-corrugated fins

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Abstract

In the recent years the development of application of vortex generator have increased the heat transfer enhancement in compact heat exchanger. But it has been a challenge to improve heat transfer without using vortex generator. The main motivation of this research is to study thermal–hydraulic performance characteristics in an oval shaped tube compact heat exchanger with new design of fins and tube by using computational fluid dynamics approach. One-corrugated and three-corrugated fins with oval tubes are used of the FTCHE (fin-and tube compact heat exchanger) that takes a huge step in the development of FTCHEs with increased thermal efficiency. The major advantage of such design is its ability to produce high thermal efficiency and performance criteria of FTCHE. I have investigated the thermal and hydraulic characteristics in the range of Reynolds number 200-800. This study proves that there is difference of flow between plain and corrugated fins and it gives better thermal and hydraulic performance. I have found that three corrugated fin patterns show better results than one corrugated fin patterns. I have found potential improvement over the baseline case.

Chapter 1: Introduction

1.1 Background

The device that transfers heat from one medium to another is basically called heat exchanger. The subject of heat transfer enhancement has increased the interest of compact thermal devices such as compact heat exchanger.

Heat exchangers are portrayed by a huge heat exchange surface territory per unit of volume of the exchanger. This brought about lessened space, weight, supporting structure, vitality prerequisite and cost; enhanced process configuration, plant format, and preparing conditions; and low liquid stock contrasted with customary outlines, for example, shell-and-tube exchangers. This proportion (wetted confront range per unit of volume) ought to be ordinarily higher than 600– 700 m2/m3 or a pressure driven width trademark Dh under 6 mm for the warmth exchanger gadget to be dealt with as smaller. [1,2].

There are different sorts of compact heat exchangers in assembling process. Heat exchangers with tube bank in cross stream are of good connected consideration in different warm and concoction designing procedures. The primary reason for utilizing them is to increment or lessening the temperature and heat exchange starting with one liquid then onto the next. The changes in the execution of the heat exchangers have pulled in numerous specialists for quite a while as they are of extraordinary specialized, temperate, and environmental significance. Execution change ends up noticeably basic especially in heat exchangers with gasses in light of the fact that the heat protection of gasses can be 10– 50 times as vast as that of fluids, which requires expansive heat exchange surface per unit of volume on the gas side. [3].

A fin-and-tube compact heat exchanger (FTCHE) is a model of compact heat exchanger (CHE) that involves a block of alternating layers of fins that can have different geometry and forms. The augmentations in the thermal–hydraulic performance and energy efficiency of the FTCHEs have been of a great interest for numerous occasions for a long time for technical designing and applied importance [4].

The conventional techniques for lessening the airside thermal resistance are by expanding the surface territory of the heat exchanger, or by decreasing the boundary layer (thermal) thickness on the surface of the heat exchanger. Expanding the surface zone is compelling however it brings about the expansion in material cost and increment in mass of the heat exchanger. One of the techniques to diminish limit layer thickness is by the age of uninvolved vortices. In this system, an impediment to produce a vortex situated toward the stream adjusts the stream field. The subsequent change in the stream because of a snag alters the nearby warm limit layer. The net effect of this manipulation is an average increase in the heat transfer for the required area. The main challenge of using vortex generators as an alternative is the significant pressure drop increase and pumping power due to increase of pressure drag and instability of flow over the tube bank. At the point when the liquid streams over the tube bank into the FTCHEs, there will be a rise in the pressure drop over each of the tubes, due to partition focuses and the back vortices of the tubes where, the properties of the liquid are shaky and change quickly.

Ameel et al. [5] used a numerical approach to investigate the effect of length in the FTCHEs with several of fin geometries. The results indicated that in order to be able to fabricate the FTCHE, the value of the number of tube rows, the longitudinal tube pitch must be compatible with the heat exchanger length as described by the performance evaluation criterion (PEC). Gong et al. [6] conducted a three dimensional numerical simulation of fluid flow and heat transfer characteristics in the FTCHEs with circular tubes and curved form of the rectangular vortex generators, which was punched on the fin surfaces. The results indicated that the average Nusselt number, friction factor and secondary flow intensity are larger than that of the reference conventional plain fin at different Reynolds number, respectively. It was concluded that the vortex generators can increase the intensity of secondary flow and reduce the size of wake regions behind circular tubes.

Previous studies of enhancement techniques in the FTCHEs mostly focused to improve heat transfer performance with smooth fin surface and the flow manipulators like vortex generators. When a fluid passes Vortex Generators, due to the friction and pressure difference on the leading and trailing edges of the winglet Vortex Generators longitudinal vortices are generated. The longitudinal vortex is three-dimensional in nature as the flow spirals around the main flow direction, and there occurs complicated flow structure. It has been found in recent decades, from reviewing the literature that the effect of geometrical parameters on heat transfer and pressure drop for various fin types and tube geometries of the FTCHEs have been investigated. However, the effective factors of in-lined corrugated patterns on the heat transfer and pressure drop across in a heat exchanger have not been analyzed numerically.

In this study, the corrugated fins and oval tubes are presented as high potential strategy as at the point when the liquid streams over the tube bank into the FTCHEs, there will be a rise in the pressure drop over each of the tubes, due to partition focuses and the back vortices of the tubes where, the properties of the liquid are shaky and change quickly. The corrugated fins is considered to improve uniformly pressure distribution over tube surfaces. Increase in heat transfer rate due to the enhancement of mixing processes of the flow in the rear of tube and reduce the wake region. Moreover, the oval tube shapes is considered to moderate the pressure loss and decrease the drag force as results of reducing pressure differential in the flow direction and resulting of the separation point delay and formation of the poor wake region.

1.2 Motivation

So far as per my knowledge this kind of investigation of thermal and hydraulic performance in the FTCHEs with oval tube and such fin patterns is showing better result. The outline of geometric model, usage and approval of the new example of balances, as one corrugated and three-corrugated fins particularly concocted for alteration of property of the convective warmth exchange and warm effectiveness in the FTCHEs. The newly designs has changed the structure of fluid flow over tube banks in the FTCHEs. In addition, the corrugated fin patterns and oval tube shapes of the FTCHEs are anticipated to improve the thermal-hydrodynamic performance of fluid flow with decreasing of pumping power and increasing the heat transfer over a wide range of Reynolds number. I am using Nusselt number, j-colburn factor & friction factor to indicate performance. This study is based on “Thermal–hydraulic performance of fin-and-oval tube compact heat exchangers with innovative design of corrugated fin patterns” by Ahmadali Gholami a, Mazlan A. Wahid a, H.A. Mohammed

Chapter 2: Model Description

2.1 Physical and computational model

The front view of one third of the computational domain is presented in Fig. 1.

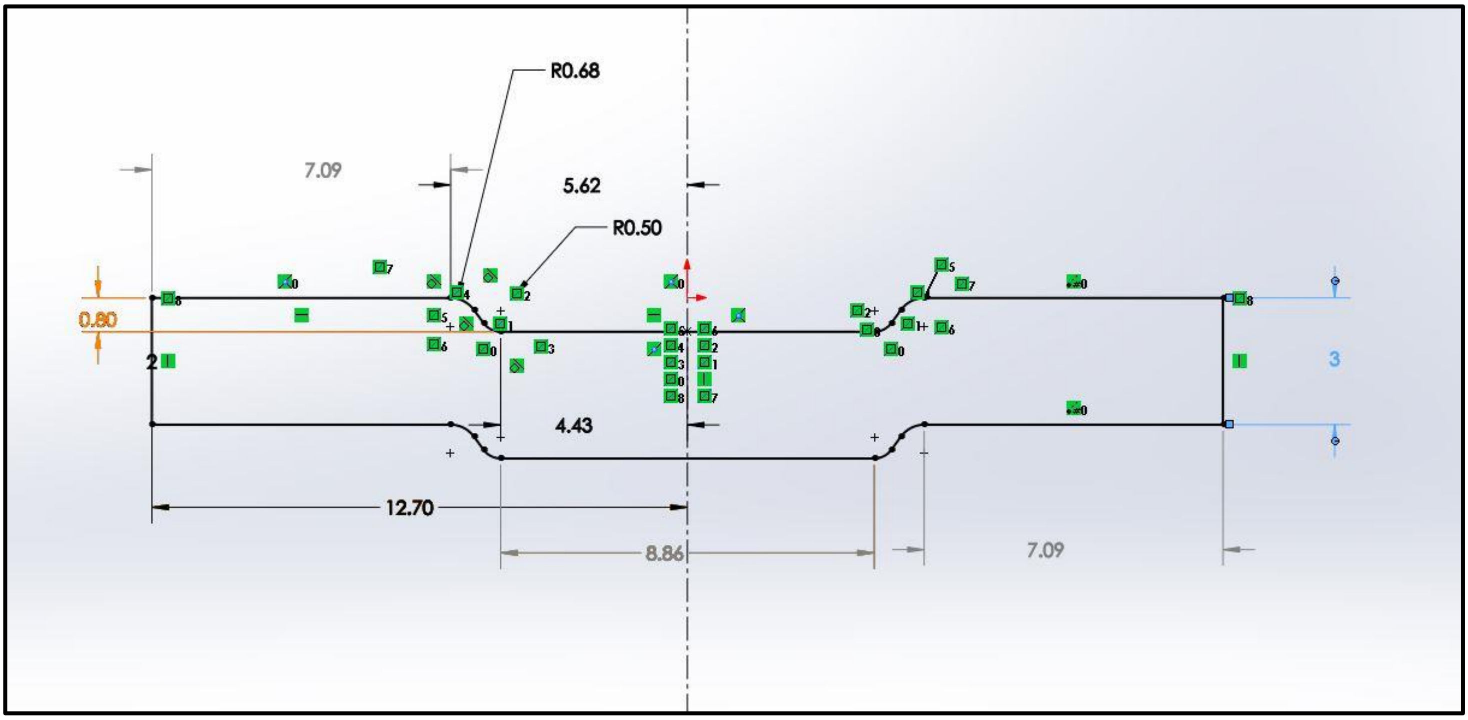


Figure 1 : 2D Front View of computational domain

These forms of fins influence on both sides of each fin to fluid flow behavior. In the present study, two kinds of corrugated design of fins are adopted, specifically one-corrugated and three-corrugated fin patterns are introduced in this work. Corrugated patterns resemble wavy structures having a progression of bends with radii 0.68 mm and 0.50 mm and included points 45 degree for the all over sides of fins profile, separately. The corrugated patterns is oriented in tube arrangement along the flow direction. These types of balances are acquainted with increment the quickening of liquid into the ebb and flow locales of folded balance at that point to make the stream destabilized, boundary layer change and mass liquid blending. The destabilized stream is produced by the corrugated fin and it lessens the thickness of the thermal boundary layer close to the balance surface and after that diminishes the airside thermal resistance. In Table 1 various comprehensive geometric parameters are listed:

Table 1: Comprehensive geometric parameters

Parameter	Symbol (unit)	Value
Major axes of oval tube	D_a(mm)	12.7
Minor axes of oval tube	D_b(mm)	7.94
Fin collar diameter	D_c(mm)	10.459
Transverse tube spacing	P_t(mm)	25.4
Longitudinal tube spacing	P_1(mm)	25.4
Fin pitch	F_p(mm)	3
Fin thickness	δ_f(mm)	0.1
Fin length	F_1(mm)	101.6
Fin width	F_w(mm)	25.4
Tube position from inlet	X_1(mm)	12.7
Number of tube	N	4
D_c=D_{tube}+2δ_f		

In Fig. 2 and Fig. 3 the computational field with three subdomains and the perpendicular coordinate system are shown in the isometric view, top view and front view where X, Y, Z stood for the span wise direction, fin pitch direction and the stream wise direction respectively. The computational domain consists of three subdomains with the names upstream side extended region, main region and downstream side extended region. Here both one corrugated and three corrugated design is shown. The upside and downside boundaries of the computational solid domains considered two neighbors corrugated fins, which consists of the airflow channel with four inline oval holes in the main domain. The position of oval tubes is indicated in Fig. 2 from the entrance of flow and exterior sides of the fins. The upstream boundary can be established at a distance of one tube diameter($D_{c)}$. in front of the leading edge of the fin to provide uniformity of the inlet velocity [49,50]. In this study, the real computational field was stretched by Fw/2 (Fw/2 > D_c) with small modification compared with Dc to maintain the inlet velocity with more consistency and uniformity. The exit domain was stretched by Fl to ensure a flow pattern to avoid recirculation flow and ensure for applying the completely developed boundary conditions at the outlet domain according to Ref. [7]. The extended domain is presented in scale with the fin length in Fig. 2. In this study, the joins between fin flange and tube is assumed in perfect contact and thus the thermal resistance in the tube and flange is neglected by employing isothermal tube walls of outside diameter D_c. The thickness of the corrugated fin is the same as the base fin thickness d_f according to [8,9,10,11].

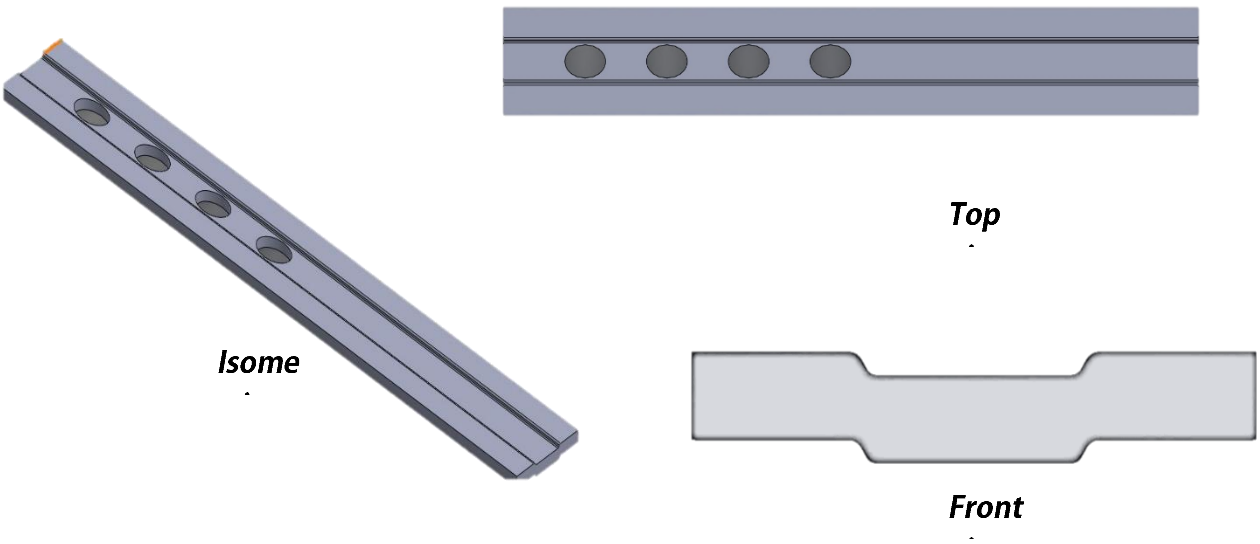


Figure 2 : Different views of computational domain for one corrugated designs

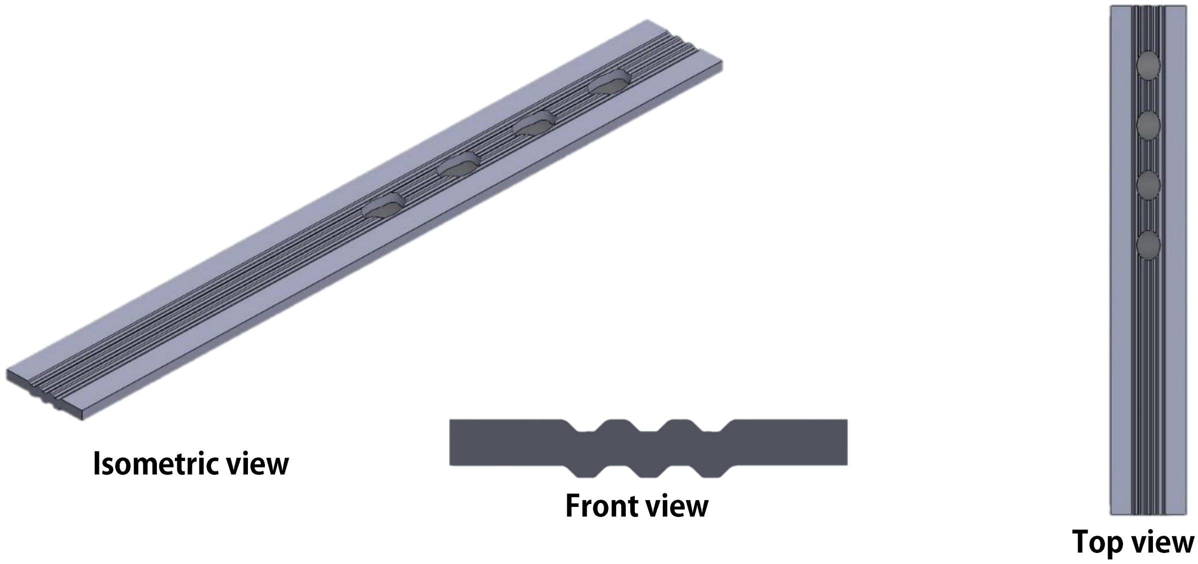


Figure 3 : Different views of computational domain for three corrugated designs

Properties of the fins and air flow are shown in Table 2 by the commercial CFD software package ANSYS Fluent.

Table 2 : Physical properties of air and aluminum

	Density(kgm^{-3})	Specific heat($Jkg^{-1}K^{-1}$)	Thermal conductivity(W $m^{-1}K^{-1}$)	Dynamic viscosity(kg $m^{-1}s^{-1}$)
Air	1.23	1006.43	.0242	.00001789
Fin	3600	765	36	-

2.2 Mathematical modeling and governing equations

The following assumptions were used:

- The incompressible flows in the airside passages are limited as laminar flow because of the small fin pitch and the low fluid velocity
- The physical properties of the working fluid and the fin material are constant
- The heat transfer is via sensible heat and no mass transfer is being taken place
- Smooth surface conditions for the fin-and-tube
- Effects of heat dissipation and thermal radiation are negligible
- The joins between fin flange and tube is assumed to be in a perfect contact
- Buoyancy force is neglected
- The flow and heat transfer are in steady state
- The viscous dissipation is negligible

Employing the basic laws of mechanics to a Newtonian fluid yields the transport principal equations for a fluid. Due to difficulty of the fluid flow over tube banks in the FTCHEs, the direct solution of the transport governing equations for mass, momentum, energy is not possible.

Generally, the small volume or element of the given fluid in the motion may undergo two changes. First, the fluid element can translate or rotate in the domain, and second, it can be distorted and consequently the element shape change is generated by a simple stretching along one or more axes, or by an angular distortion. The transport equations expressed in tensor form consists of continuity, momentum and energy equations in the computational domains are presented in Table 3.

Table 3 : Governing equations

Continuity equation: $\frac{\partial(\rho u_i)}{\partial x_i} = 0.0$	(1)
Momentum equation: $\frac{\partial(\rho u_i u_j)}{\partial x_i} = \frac{\partial}{\partial x}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial \rho}{\partial x_j}$	(2)
Energy equation: $\frac{\partial(\rho u_i T)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right)$	(3)
Paper pads $\frac{\partial(\rho u_i \emptyset)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \emptyset}{\partial x_i}\right) - S_\phi$	(4)

2.3 Boundary conditions

The boundary conditions used in the simulation are listed in the following Table. Thermal and hydraulic characteristics were investigated in all surfaces using these.

Table 4 : Boundary conditions

In the upstream side in the extended region (inlet domain)	**In the downstream extended region (Outlet domains)**	**For the surface of fins as coil and main region of model**	**In the outlet region, Neuman boundary condition (One-way)**
At the entrance area as inlet side of boundary $U = U_{in} = Const$ $U_x = U_y = 0$ $T = T_{in} = 300K$	At the up and down side boundaries $\frac{\partial U_x}{\partial y} = 0$ $U_y = 0$ $\frac{\partial U_z}{\partial y} = 0$ $\frac{\partial T}{\partial y} = 0$	In the upside and bottom side of boundaries $U_x = U_y = U_z = 0$ $\frac{\partial p}{\partial n}$=0	$\frac{\partial U_x}{\partial z} = 0$ $U_y = 0$ $\frac{\partial U_z}{\partial z} = 0$ $\frac{\partial T}{\partial z} = 0$ $\frac{\partial P}{\partial z} = 0$
In the up and bottom side of boundaries $\frac{\partial U_x}{\partial y} = 0$ $U_y = 0$ $\frac{\partial U_z}{\partial y} = 0$ $\frac{\partial T}{\partial y} = 0$ $\frac{\partial P}{\partial y} = 0$	For the lateral of boundaries $\frac{\partial U_y}{\partial x} = 0$ $U_x = 0$ $\frac{\partial U_z}{\partial x} = 0$ $\frac{\partial T}{\partial x} = 0$	For the lateral boundaries of fluid region $\frac{\partial U_y}{\partial x} = 0$ $U_x = 0$ $\frac{\partial U_z}{\partial x} = 0$ $\frac{\partial T}{\partial x} = 0$	
For the lateral of boundaries $\frac{\partial U_y}{\partial x} = 0$ $U_x = 0$ $\frac{\partial U_z}{\partial x} = 0$ $\frac{\partial T}{\partial x} = 0$ $\frac{\partial P}{\partial x} = 0$	For the surfaces on tube walls $U_x = U_y = U_z = 0$ $T = T_{wall} = 350K$ $\frac{\partial p}{\partial n}$=0 ; n signifies the normal direction	For the lateral side of boundaries at the fin surface region $U_x = U_y = U_z = 0$ $\frac{\partial T}{\partial x}$=0	

2.4 Parameter definitions

Reynold's number, $Re = \frac{\rho U_m D_h}{\mu}$

- The flow condition can be characterized by Reynolds number. It is the ratio of inertial forces to viscous forces
- It is used to predict the transition from laminar to turbulent flow.

J-coulbarn factor, $\mathrm{j} = \frac{h}{\rho U_m C_p} Pr^{\frac{2}{3}}$

- It helps to predict unknown properties.

Nusselt Number, $Nu = \frac{hD_h}{k}$

- The Nusselt number (Nu) is the ratio of convective to conductive heat transfer across (normal to) the boundary.

Fanning friction factor, $\mathrm{f} = \frac{2\Delta P A_c}{\rho U_m^2 A_0}$

- It gives us the significance of friction.

Pressure drop, $\Delta P = (P_{inlet} - P_{outlet})$

Logarithmic mean temperature difference, $\Delta T_m = \frac{(T_{wall} - T_{inlet}) - (T_{wall} - T_{outlet})}{\ln[\frac{T_{wall} - T_{inlet}}{T_{wall} - T_{outlet}}]}$

Total Heat Transfer rate, $Q = \dot{m} C_p (T_{outlet} - T_{inlet})$

Convective heat transfer coefficient, $h = \frac{Q}{A_0 \Delta T_m}$

2.5 Grid generation and grid independency

Grid generation or meshing of the physical domain is the base of many computational methods in which complicated partial differential equations on fluids case studies are being solved. I have used tetrahedron shape as my element shape. The details of the meshing nodes and elements are shown in the following table.

Table 5 : Mesh and Physics Report

Mesh Report

Domain	**Nodes**	**Elements**
Part	331413	1711315

Physics Report

Domain - Part	
Type	cell

Domain	Boundaries	
Part	Boundary – bottom surface	
	Type	Wall
	Boundary – hot tubes	
	Type	Wall
	Boundary – inlet	
	Type	Velocity inlet
	Boundary – outlet	
	Type	Pressure outlet
	Boundary – Top surface	
	Type	Wall
	Boundary – wall	
	Type	Wall

ANSYS Workbench was used to solve the governing equations. [12] The meshes of computational domains were generated by using ANSYS Meshing. The validation of the grid independency for the numerical solution is carried out for different grid numbers. In order to validate the solution independence of the grid, four different numbers of grid points were tested.

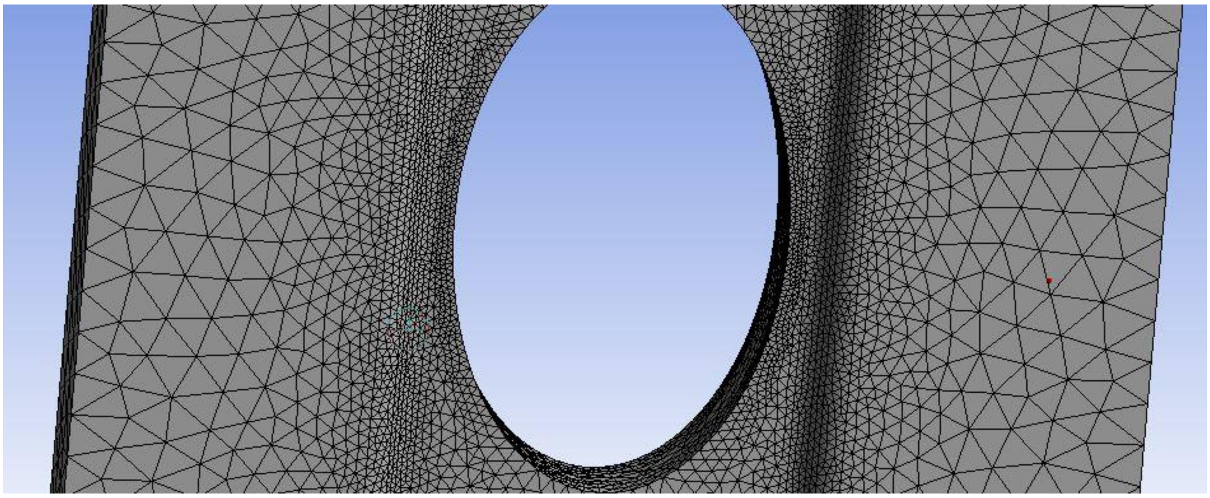


Figure 4 : Grid generation on the computational model

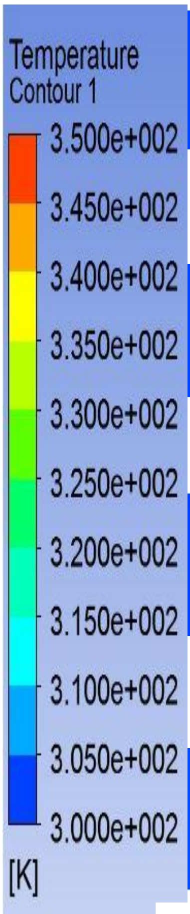
At last the elements and nodes used in the analysis showed minimal error. So simulation was carried out using this.

Chapter 3: Results and discussion

The enhancement of heat transfer in a fin surface is critical to improve the overall performance of heat exchangers. The corrugated fins and oval tubes are introduced to increase the heat transfer coefficient and reduce the thermal resistance on the airside by interrupting thermal boundary layer growth, thereby increasing the convective heat transfer coefficients and reducing the airside resistance. By using these fin shapes, fluid is accelerated to curvature regions of corrugated fin then to make destabilized flow, boundary layer modification and bulk fluid mixing. The corrugated fins enhance the mixing of the hot and cold fluids and improve the poor heat transfer in the wake region. The destabilizing flows are generated by the corrugated fin and would reduce the thickness of the thermal boundary layer near the fin surface and then reduce the airside thermal resistance. In addition, an oval shaped tube presented in this study was selected to improve the heat transfer and the pressure loss. Moreover, it delays the separation in comparison with the circular cross section, which helps decrease the friction factor and the flow resistance leading to a major decrease in the pressure loss. Another main advantage in delaying the separation and reattaching the flow after the separation is the decrease in the pressure drop mainly due to friction drag in the case of using the oval shaped tubes.

3.1 Temperature distribution

The variation of the temperature is shown in Figs. 7 and 8, respectively. The range of Reynolds number used is from 200 to 900.



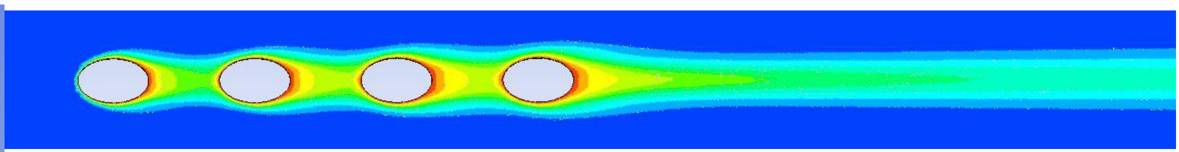


Figure 6 : Temperature contour of one corrugated fin at Re=200

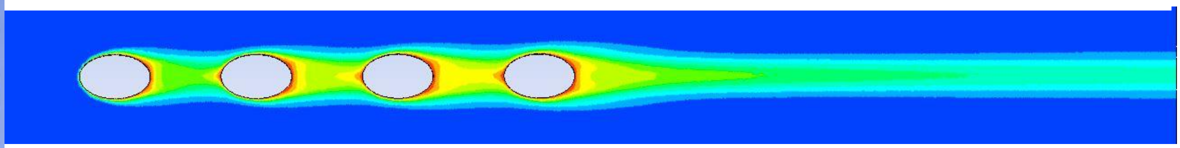


Figure 5 : Temperature contour of one corrugated fin at Re=400

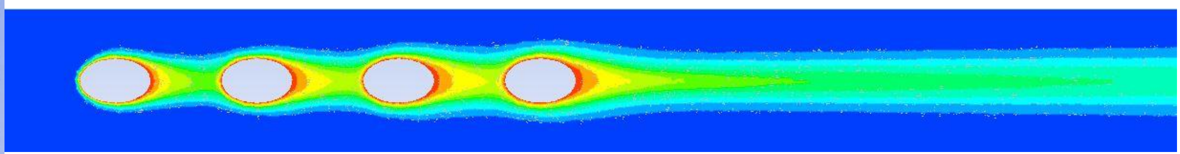


Figure 7 : Temperature contour of three corrugated fin at Re=200

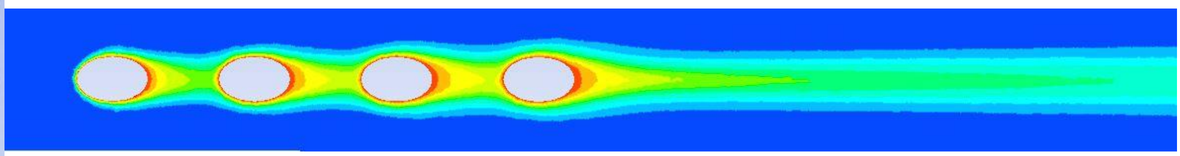


Figure 8 : Temperature contour of three corrugated fin at Re=600

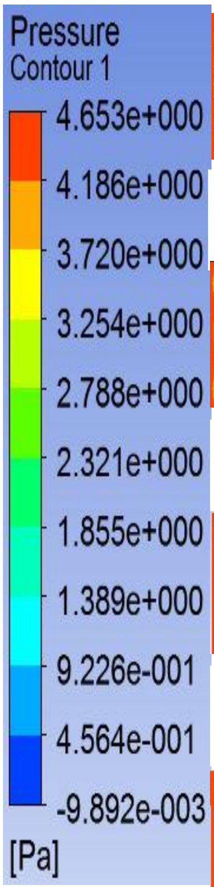
We see that just behind the hot tubes the temperature is high. I got ~335^0 C in the outlet region which is very good for heat exchanger. Since we are considering laminar flow the temperatures at sides are not as behind tubes. We have taken the surface just middle of the XZ plane.

3.2 Pressure Field

The variation of pressure distribution along the flow direction is displayed in Fig. 9 & 10. The difference between pressure contours in two configurations of the FTCHEs is evidently shown the effects of using the corrugated fins and oval tubes geometry on the dynamic flow characteristic along the physical domains. These results presented the pressure characteristic in the surface area of air flow at h = 1.6 mm.

The oval tube is introduced as a streamlined body, whereas circular tube is considered as a bluff body and conventional form of tube. For streamlined bodies, the frictional drag is the dominant source of air resistance. For a bluff body, the dominant source of drag is the pressure drag. For a given frontal area and velocity, a streamlined body will always have a lower resistance than a bluff body. Cylinders are considered bluff bodies because at large Reynolds numbers the drag is dominated by the pressure losses in the wake region. In this study, four tubes between fin surfaces through an air fluid experience a drag force, which is usually divided into two components: the frictional drag and the pressure drag. The frictional drag comes from friction between the fluid and the surfaces with flowing of air over them. The pressure drag comes from the eddying motions that are set up in the fluid by the passage of the body. This drag is associated with the formation of a wake, which can be readily seen behind each tube of the FTCHE, and it is usually less sensitive to Reynolds number than the frictional drag. The pressure drag is important for separated flows, and it is related to the cross-sectional area of the body. Furthermore, the pressure inside the wake region remains low as the flow separates and a net pressure force (pressure drag) is produced between the stagnation point in the front of tube and back of tube.

For the oval shape, the boundary layer on the top and bottom surface experience only mild pressure gradients and they remain attached along almost the entire major axis length. The wake area is very small and the viscous friction dominates the drag force inside the boundary layers. However, for the round tube, the pressure gradients over the surface increase in magnitude. In particular, the adverse pressure gradient on the top and bottom rear portions of the tube may become sufficiently strong to produce a separated flow. This separation will increase the size of the wake and the pressure losses in the wake due to eddy formation and therefore the pressure drag increases. For the tube with round shape a larger fraction of flow over surface may be separated that in this stage the pressure drag is much greater than the viscous drag.



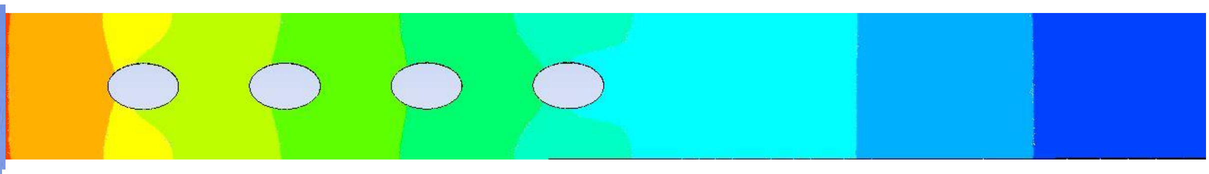


Figure 9 : Pressure contour of one corrugated fin at Re=200

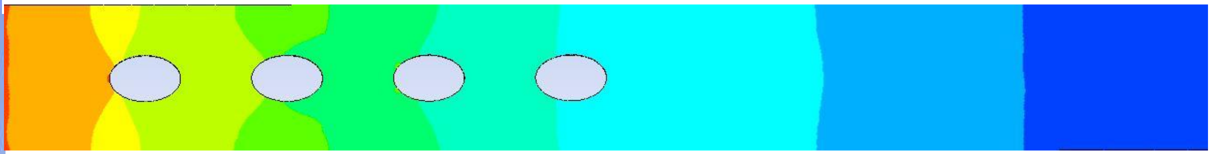


Figure 10 : Pressure contour of one corrugated fin at Re=400

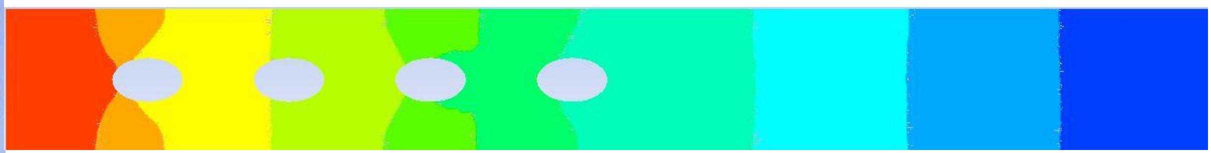


Figure 11 : Pressure contour of three corrugated fin at Re=200

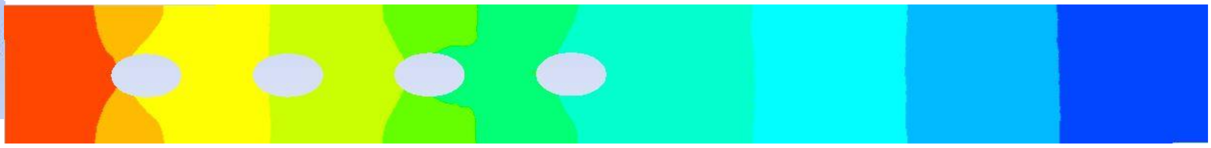
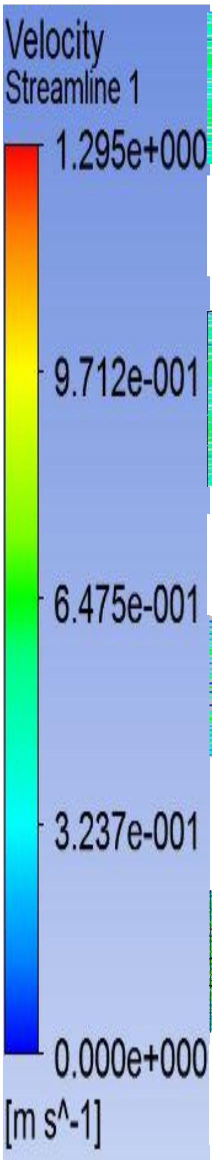


Figure 12 : Pressure contour of three corrugated fin at Re=600

3.3 Velocity streamline

Figs. 13 and 14 display the velocity contour for the physical domain in the surface plane 1.6 mm for one-corrugated fins. The effects of fins and tubes show how the new designs affected the local velocity contour in the flow field of FTCHE. The corrugated fins can change the flow patterns over tube surfaces and increased the acceleration of fluid into wake region. The flow acceleration delays the separation from the tube, reduces the drag across the tube and aids the fluid into the wake recirculation zone.

When the fluid flows through the corrugated patterns, causing the bulk flow mixing, boundary layer modification, and flow destabilization; heat transfer is enhanced due to these forms of fins. The reduction of wake regions and improve mixing process are the basic mechanism for enhanced convective heat transfer.



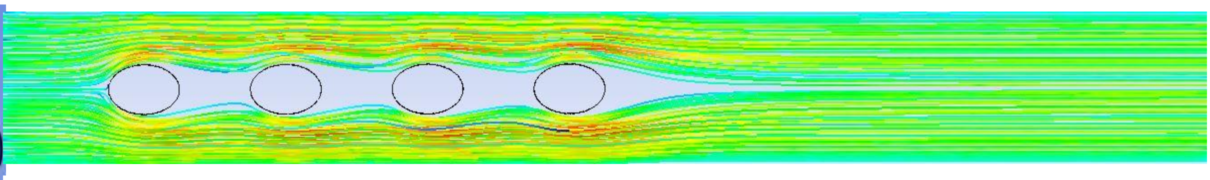


Figure 13 : Velocity streamline of one corrugated fin at Re=200

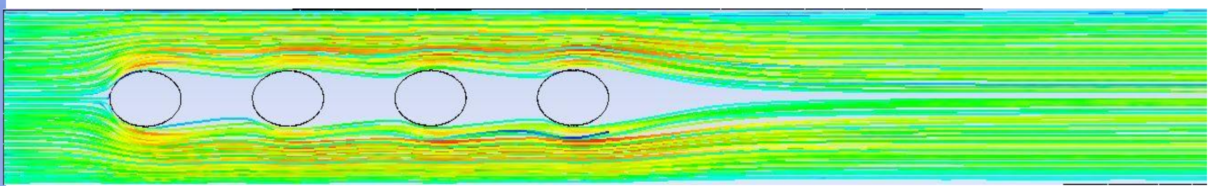


Figure 14 : Velocity streamline of one corrugated fin at Re=400

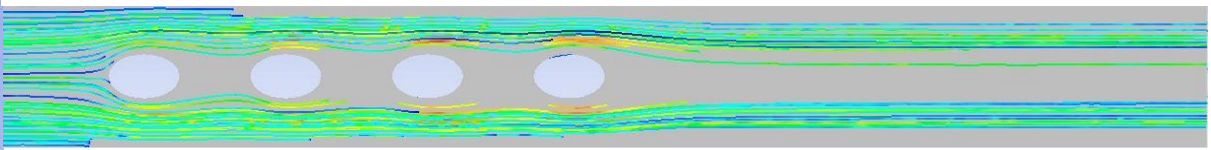


Figure 15 : Velocity streamline of three corrugated fin at Re=200

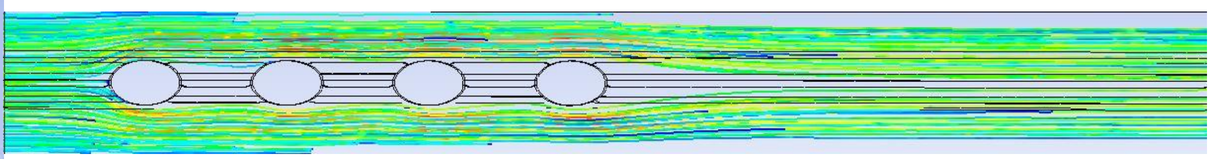


Figure 16 : Velocity streamline of three corrugated fin at Re=600

3.4 Friction factor

In the following figures we the difference of friction factors for one corrugated and three corrugated fin patterns with plate fin & circular tube. I also compared the results with the literature referred here.

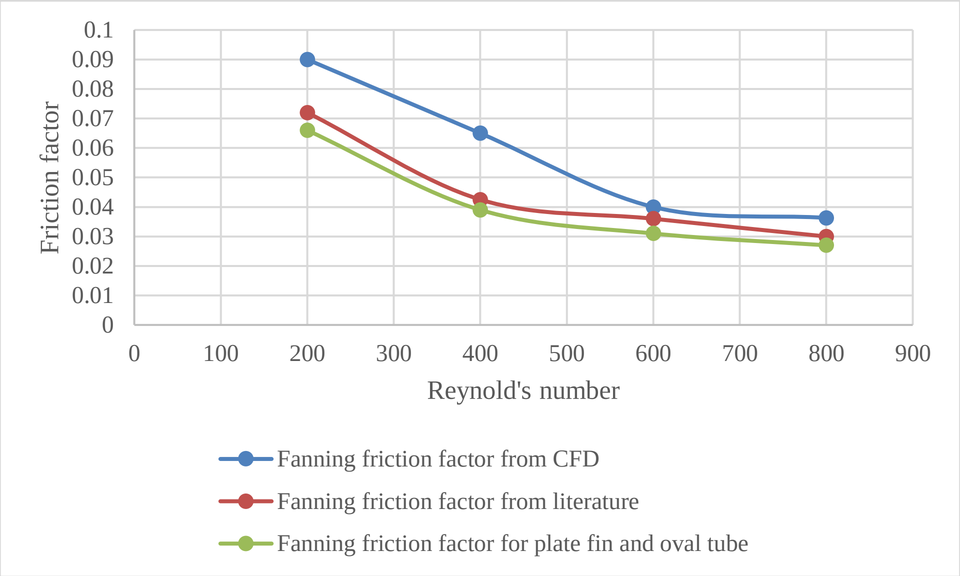


Figure 17 : Friction factor comparison for one corrugated fins

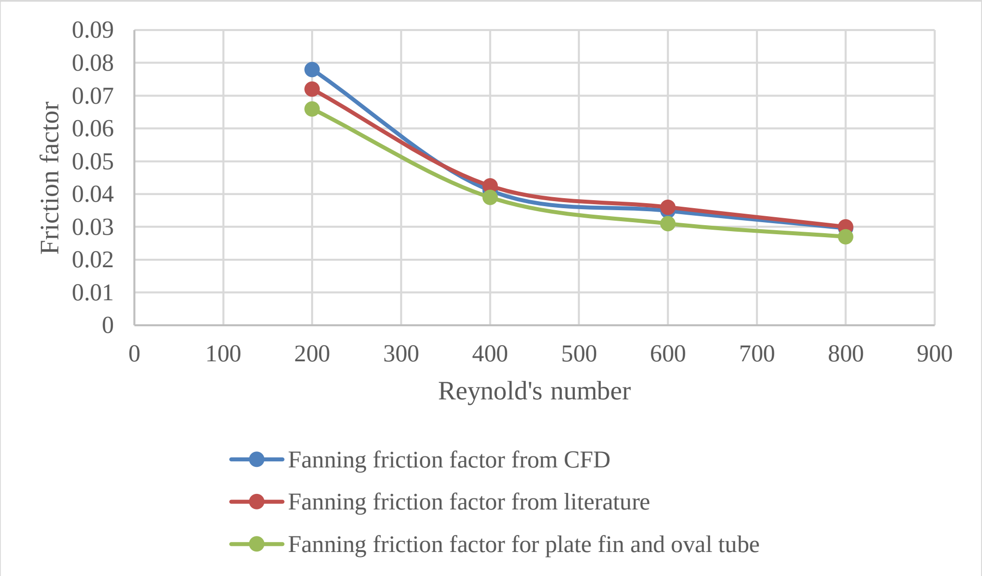


Figure 18 : Friction factor comparison for three corrugated fins

3.5 Nusselt number

In heat transfer at a boundary (surface) within a fluid, the Nusselt number (Nu) is the ratio of convective to conductive heat transfer across (normal to) the boundary. In this context, convection includes both advection and diffusion. Named after Wilhelm Nusselt, it is a dimensionless number [19]. In Figure 15 & 16 we see significant improvement with our designs. The oval tubes and corrugated patterns have better result than plate fin & circular tube.

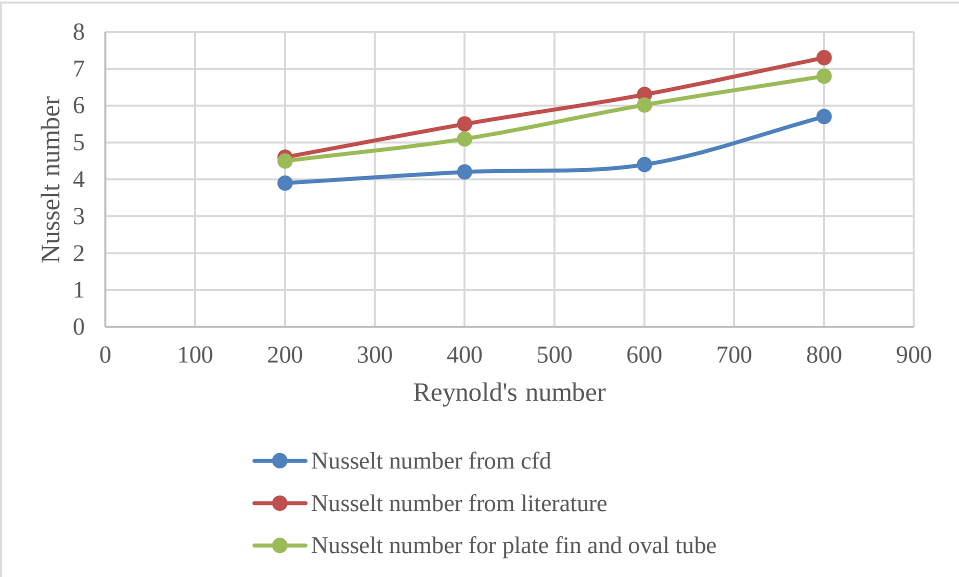


Figure 19 : Nusselt number comparison for one corrugated fins

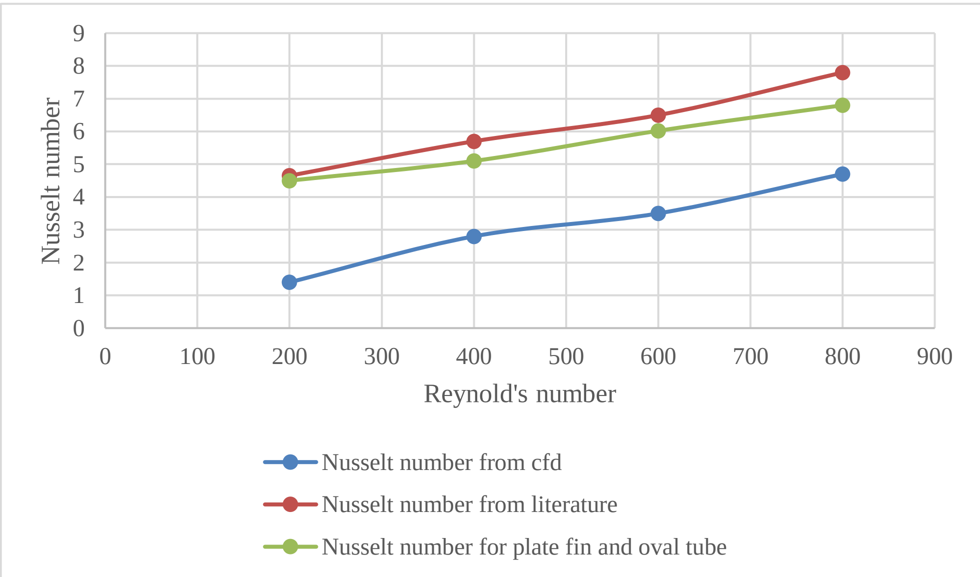


Figure 20 : Nusselt number comparison for three corrugated fins

3.6 J-colburn factor

I have used another parameter to evaluate performance of this design. The j-Colburn factor is a non-dimension quantity, and shows a relation between the convective heat transfer, fluid properties, flow conditions and geometry. The outcome of simulation is presented in the following Fig. 17 & 18 for one and three corrugated design respectively as the variation of the j-Colburn factor as criteria analogy between heat and momentum with different Reynolds numbers. The numerical results reveal that the fins configurations have a significant effect on the thermal and hydrodynamic characteristic. The j-factor increases with increasing number of curvature regions in corrugated fins. In addition, the tube shapes with oval forms had a positive effect on the j-factor.

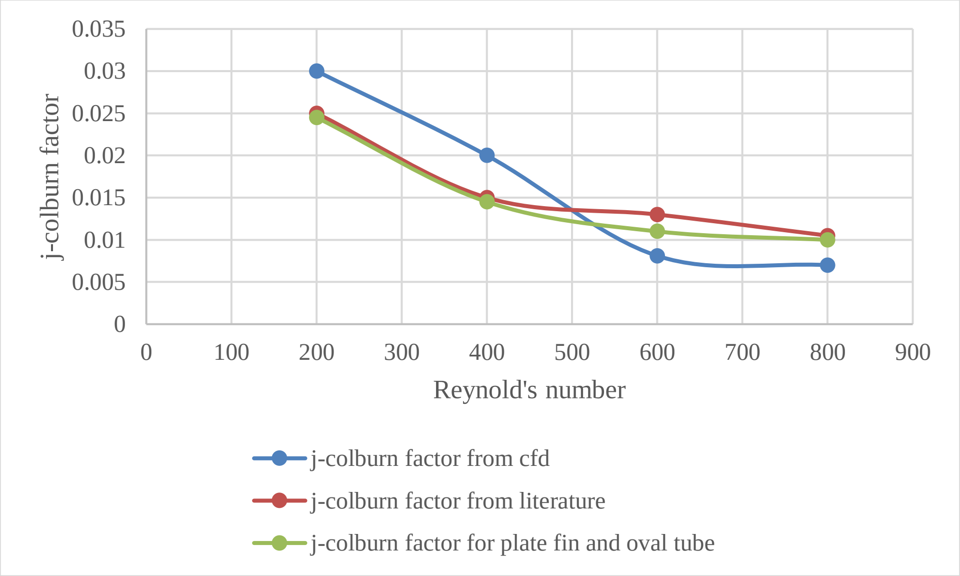


Figure 21 : J-colburn factor comparison for one corrugated fins

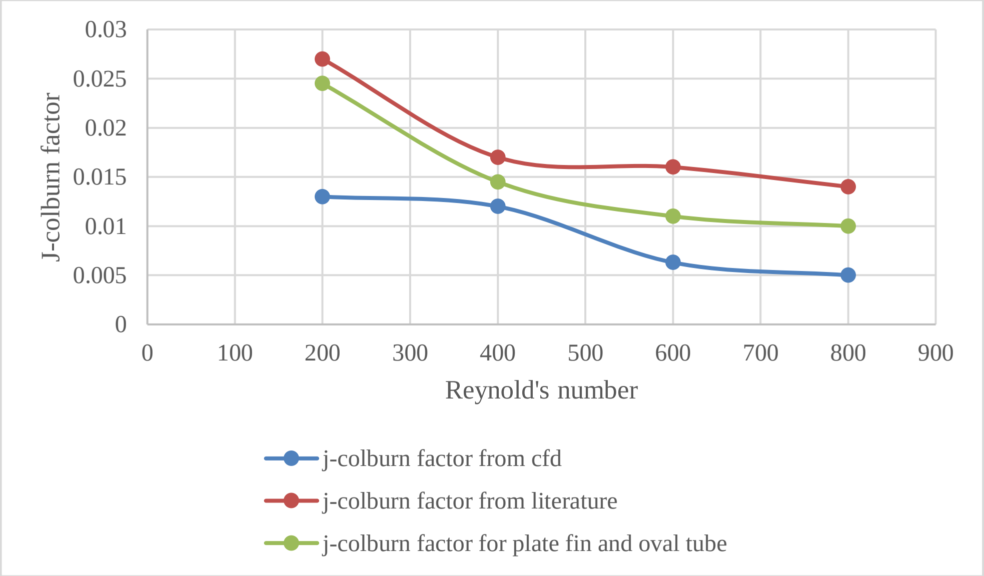


Figure 22 : J-colburn factor comparison for three corrugated fins

Chapter 4: Conclusion

Three-dimensional CFD simulation results of the thermo-hydraulic characteristics of the corrugated fin-and-oval tube compact heat exchangers with a new types of fin patterns such as one-corrugated and three-corrugated designs considering curvature regions of corrugated patterns in the tube directions were presented in this study. In the last few decades, literature records revealed that using vortex generator as principal strategies have been used to augment the heat transfer performance in this field. New design of fins without vortex generators as central approach was introduced in this study. Following remarks were found:

- The corrugated fins with one and three curve regions improved uniformly the pressure distribution and increased the convective heat transfer over tube surfaces.
- Oval tubes are better than circular tubes in terms of performance.
- Compared with the baseline case, the corrugated fin with oval tube shows potential improvement of heat transfer performance and moderate pressure loss in the FTCHE.
- Some errors were found comparing with the literature CFD as the elements number and meshing were not same
- Still improved performance was investigated.

Chapter 5: Bibliography

[1] R.K. Shah, D.P. Sekulic, Heat exchangers, in: Handbook of Heat Transfer, 1998, p. 3.

[2] A. Stankiewicz, J.A. Moulijn, Re-Engineering the Chemical Processing Plant: Process Intensification, CRC Press, 2003.

[3] S.A.E. Sayed Ahmed, O.M. Mesalhy, M.A. Abdelatief, Flow and heat transfer enhancement in tube heat exchangers, Heat Mass Transfer 51 (11) (2015) 1607–1630.

[4] S. Tiggelbeck, M. Fiebig, Flow structure and heat transfer in a channel with multiple longitudinal vortex generators, Exp. Therm. Fluid Sci. 5 (4) (1992) 425–436.

[5] B. Ameel, J. Degroote, C. T'Joen, H. Huisseune, S. De Schampheleire, J. Vierendeels, M. De Paepe, Accounting for the effect of the heat exchanger length in the performance evaluation of compact fin and tube heat exchangers, Appl. Therm. Eng. 65 (1–2) (2014) 544–553.

[6] B. Gong, L. Wang, Z. Lin, Heat transfer characteristics of a circular tube bank fin heat exchanger with fins punched curve rectangular vortex generators in the wake regions of the tubes, Appl. Therm. Eng. 75 (2015) 224–238.

[7] M. Behfard, A. Sohankar, Numerical investigation for finding the appropriate design parameters of a fin-and-tube heat exchanger with delta-winglet vortex generators, Heat Mass Transfer (2015).

[8] B. Gong, L. Wang, Z. Lin, Heat transfer characteristics of a circular tube bank fin heat exchanger with fins punched curve rectangular vortex generators in the wake regions of the tubes, Appl. Therm. Eng. 75 (2015) 224–238.

[9] Z.M. Lin, C.P. Liu, M. Lin, L.B. Wang, Numerical study of flow and heat transfer enhancement of circular tube bank fin heat exchanger with curved delta winglet vortex generators, Appl. Therm. Eng. 88 (2015) 198–210.

[10] Z.M. Lin, L.B. Wang, Y.H. Zhang, Numerical study on heat transfer enhancement of circular tube bank fin heat exchanger with interrupted annular groove fin, Appl. Therm. Eng. 73 (2) (2014) 1465–1476.

[11] Y.L. He, H. Han, W.Q. Tao, Y.W. Zhang, Numerical study of heat-transfer enhancement by punched winglet-type vortex generator arrays in fin-andtube heat exchangers, Int. J. Heat Mass Transfer 55 (21–22) (2012) 5449–5458.

[12] ANSYS Inc, ANSYS Fluent Theory Guide, 2013.

[13] K.M. Kwak, K. Torii, K. Nishino, Simultaneous heat transfer enhancement and pressure loss reduction for finned-tube bundles with the first or two transverse rows of built-in winglets, Exp. Therm. Fluid Sci. 29 (5) (2005) 625–632.