IMPROVEMENT OF THERMAL- HYDRAULIC PERFORMANCE OF COMPACT HEAT EXCHANGERS WITH MULTI-CORRUGATED FIN AND OVAL TUBE ARRAYS

By

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SUPERVISED BY
DR. ARAFAT AHMED BHUIYAN

A Thesis Submitted to the Academic Faculty in Partial Fulfillment of the Requirements for the Degree of

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

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DECLARATION

It is hereby declared 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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CERTIFICATE OF RESEARCH

The thesis titled **"Improvement of thermal-hydraulic performance of compact heat exchangers with multi-corrugated fin and oval tube arrays"** submitted by Saiyed Tasnim Md. Fahim (141446) & Rayhanul Islam (141447)) has been accepted as satisfactory in partial fulfillment of requirement for the Degree of Bachelor of Science in Mechanical Engineering on November 2018.

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List of Acronyms

Nomenclature

A	total heat transfer surface area [m^2]
A_c	minimum cross-sectional [m^2]
A_o	total surface area [m^2]
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_l	fin length [m]
F_w	fin width [m]
F_p	fin pitch [m]
h	heat transfer coefficient [W/m^2 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 ($h\ D_h/k$)
m_f	mass flow rate [kg/s]
ΔP	pressure drop [pa]
P_l	longitudinal pitch [m]
Pr	Prandtl number
P_t	transverse pitch [m]
Q	heat transfer rate [w]
X_l	tube position from inlet [m]
Re	Reynolds number ($(\rho\ U_m\ D_h)/\mu$)
Re_{Dc}	Reynolds number based on D_c
Re_{Dh}	Reynolds number based on D_h
S_ϕ	volumetric source term
T	temperature [K]
U_m	mean velocity at the minimum flow cross-sectional [$m\ s^{-1}$]

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

Acknowledgements

The thesis was carried out by the authors themselves under the close supervision and guidance of DR. ARAFAT AHMED BHUIYAN, Department of Mechanical & Chemical Engineering, Islamic University of Technology (IUT). We would like to thank him from the deepest of our heart, for helping us all the way. He dedicated his valuable time & effort to solve our problems & guided us in such a nice way that is really beyond imaginations. His vast knowledge in the field of Heat Transfer also enhanced our venture to a great extent. Last but not the least we express our gratitude to our Parents and ALLAH, THE ALMIGHTY.

Abstract

Compact heat exchanger (CHXs) is one of the top choices in residential and industrial applications for its higher thermal efficiency, lower pressure drop, low cost, light weight characteristics. Among various types of compact thermal devices, fin and tube compact heat exchangers are stimulating enormous interest in thermal engineering field. In the present study, airside performance of CHXs having corrugated fin and oval tube are evaluated using a CFD code ANSYS 12.0. Main motivation of this investigation is to find out the pattern of gradual improvement of thermal-hydraulic performance by showing the comparison between plane and corrugated fin geometry. This study has been performed for Reynolds number ranging from 200 to 1000. Here we have analyzed the performance among plane, one, two & three corrugated fin geometry. Also, the variation of tube shape changing the circular tube geometry to oval shape has also been studied. The significant increase of Nusselt number, Friction factor, Colburn factor has been found for the corrugated compact heat exchangers. Results indicate that incurring a moderate pressure loss the corrugated fin patterns can ensure more thermal efficiency. The pattern of gradual improvement of heat transfer rate and slacken pressure loss with the increase of corrugation in a fin with oval shaped tubes has been experienced.

Keywords:

Compact Heat Exchanger, Corrugated channel, friction factor, Colburn factor, Reynolds Number.

Chapter 1

1.1 Introduction

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.

Compact heat exchangers are characterized by a large heat transfer surface area per unit of volume of the exchanger. This resulted in reduced space, weight, supporting structure, energy requirement and cost; improved process design, plant layout, and processing conditions; and low fluid inventory compared to conventional designs such as shell-and-tube exchangers. This ratio (wetted face area per unit of volume) should be typically higher than 600–700 m2/m3 or a hydraulic diameter characteristic Dh less than 6 mm for the heat exchanger device to be treated as compact [1,2].

There are various types of compact heat exchangers in manufacturing process. Heat exchangers with tube bank in cross flow are of great applied attention in various thermal and chemical engineering processes. The main purpose of using them are to increase or decrease the temperature and heat transfer from one fluid to another. The improvements in the performance of the heat exchangers have attracted many researchers for a long time as they are of great technical, economical, and ecological importance. Performance improvement becomes essential particularly in heat exchangers with gases because the thermal resistance of gases can be 10–50 times as large as that of liquids, which requires large heat transfer 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) involving 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 researchers for a long time for fundamental technical designing and various applied importance [4].

The traditional methods of reducing the airside thermal resistance are by increasing the surface area of the heat exchanger, or by reducing the thermal boundary layer thickness on the surface of the heat exchanger. Increasing the surface area is effective but it results in the increase in material cost and increase in mass of the heat exchanger. One of the methods to reduce boundary layer thickness is by the generation of passive vortices. In this technique, an obstacle to generate a vortex oriented in the direction of the flow alters the flow field. The resulting change in the flow due to an obstacle modifies the local thermal boundary layer. The net effect of this manipulation is an average increase in the heat transfer for the affected area. The main challenge of using vortex generators as passive methods is the significant increase of pressure drop and pumping power due to increase of pressure drag and instability of flow over the tube bank. When the fluid flows over the tube bank into the FTCHEs, there will be a rise in the pressure drop over each of the tubes, because of separation points and the rear vortices of the tubes where, the properties of the fluid are unstable and change rapidly.

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 known as vortex generators. When a fluid passes VGs, longitudinal vortices are generated due to the friction and pressure difference on the leading and trailing edges of the winglet VGs. The longitudinal vortex is invariably three-dimensional as the flow spirals around the main flow direction, and the flow structure is complicated. It has been found, 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 during the recent decades. 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 introduced as high potential strategy to evaluate the functioning of specially designed of fin and tube for moderate pressure loss and enhancement of the thermo-hydraulic performance of the FTHEs. The corrugated fins is considered to improve uniformly pressure distribution over tube surfaces. Increment of the heat transfer rate due to the enhancement of mixing processes of the flow in the rear of tube and reduce the wake region. In addition, 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 Background and Motivation

To the best of the authors' knowledge, the investigation of thermal-hydrodynamic performance of airflow in the FTCHEs with innovative design of fin patterns and oval tube is new and it is not available in literature so far. The design of geometric model, implementation and validation of the new pattern of fins, as one corrugated and three-corrugated fins especially devised for modification of property of the convective heat transfer and thermal efficiency 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. We are using Nusselt number, j-colburn factor & friction factor to indicate performance.

Chapter 2

Literature review

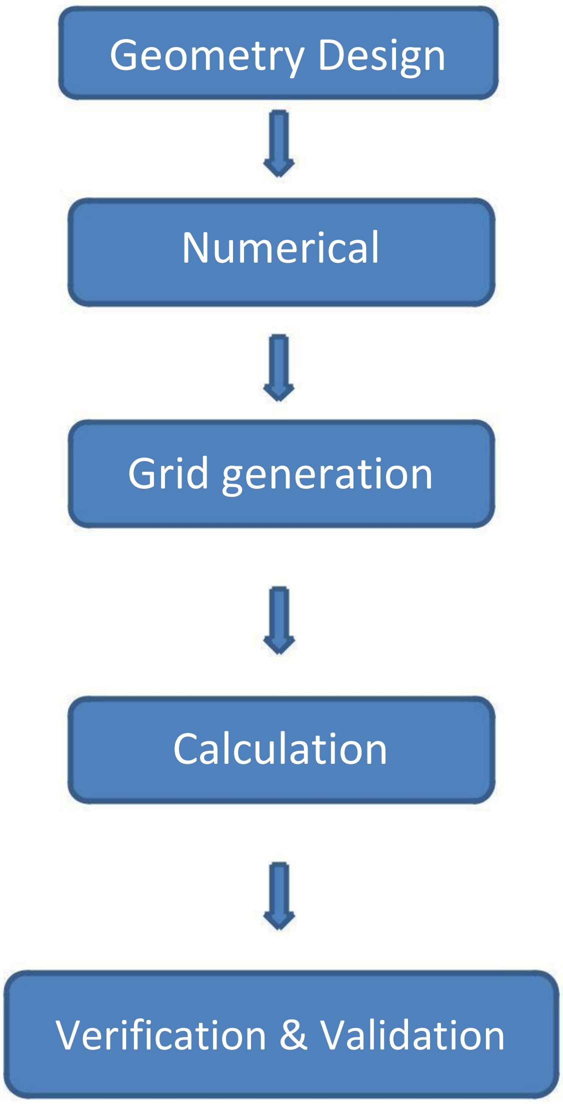
This study is based on "Thermal–hydraulic performance of fin-and-oval tube compact heat exchangers with innovative design of corrugated fin patterns"

by

Ahmadali Gholami a, Mazlan A. Wahid a, H.A. Mohammed b

Chapter 3

Methodology



Chapter 4
Model Description

4.1 Physical and computational model

Usual core regions with one and three-corrugated fin patterns is presented in Fig. 1. Two designs which are different, of the FTCHEs introduced in this study are shown in this figure, it consists in many corrugated fins with parallel passages of airflow area between them. Fig. 2(i) & (ii) points out the schematic representation of front view of the corrugated fin shapes with two different patterns and the corrugated profiles.

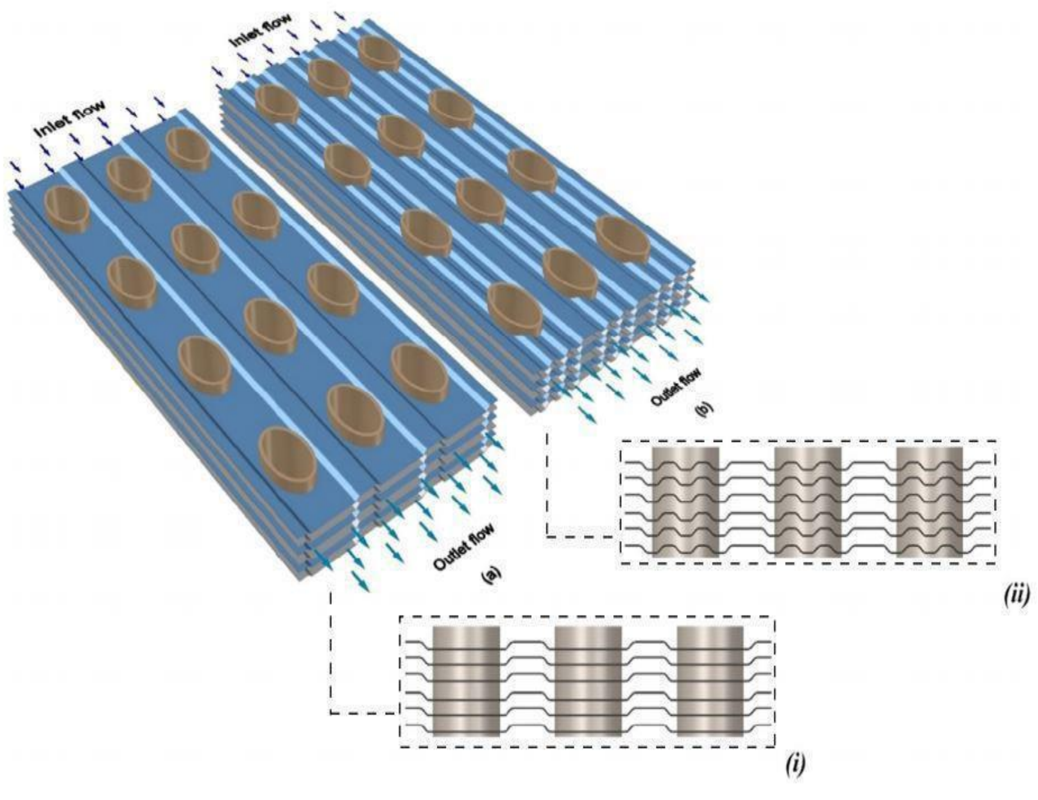
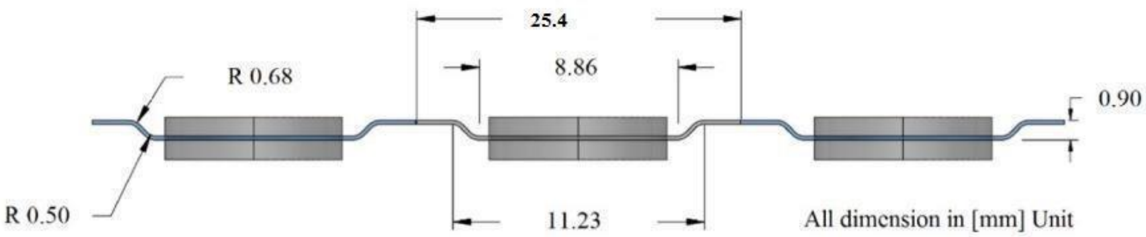
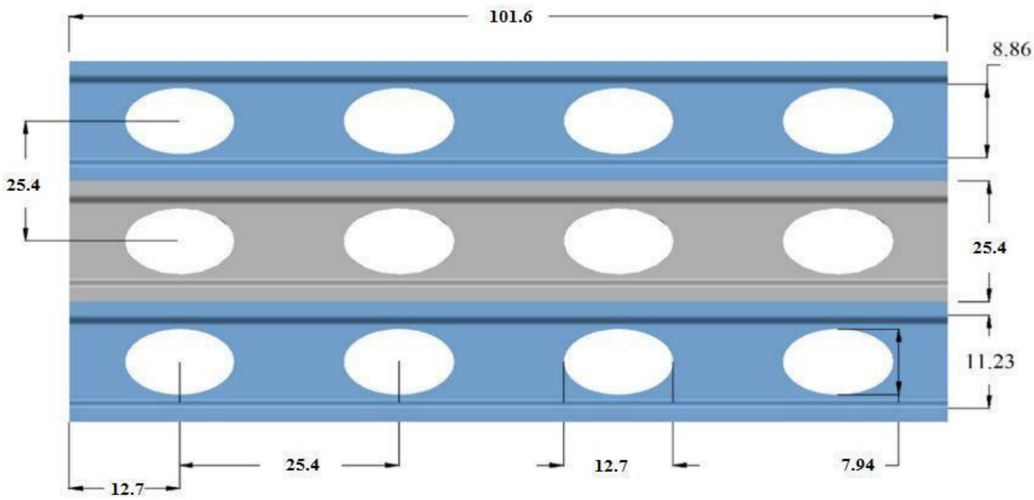


Figure 1 : Schematic diagram of corrugated-fin-and-oval tube compact heat exchangers: (a) one-corrugated fin patterns; (b) three-corrugated fin patterns

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 are like wavy forms having a series of curves with radii 0.68 mm and 0.50 mm and included angles 45 degree for the up and down sides of fins profile, respectively. The corrugated patterns is oriented in tube arrangement along the flow direction. These forms of fins are introduced to increase the acceleration of fluid into the curvature regions of corrugated fin then to make the flow destabilized, boundary layer modification and bulk fluid mixing. The destabilized flow is generated by the corrugated fin and it reduces the thickness of the thermal boundary layer near the fin surface and then reduces the airside thermal resistance. The top and front views of the computational domain of FTCHEs involve the corrugated fins and oval tubes with various geometrical parameters consist in one-corrugated patterns and three corrugated patterns are shown in Figs. 2 and 3, respectively.



Front view



Top view

Figure 2 : Geometric characteristics of one-corrugated fin patterns: (a) top view of one-corrugated fin pattern; (b) front view of one-corrugated fin profile.

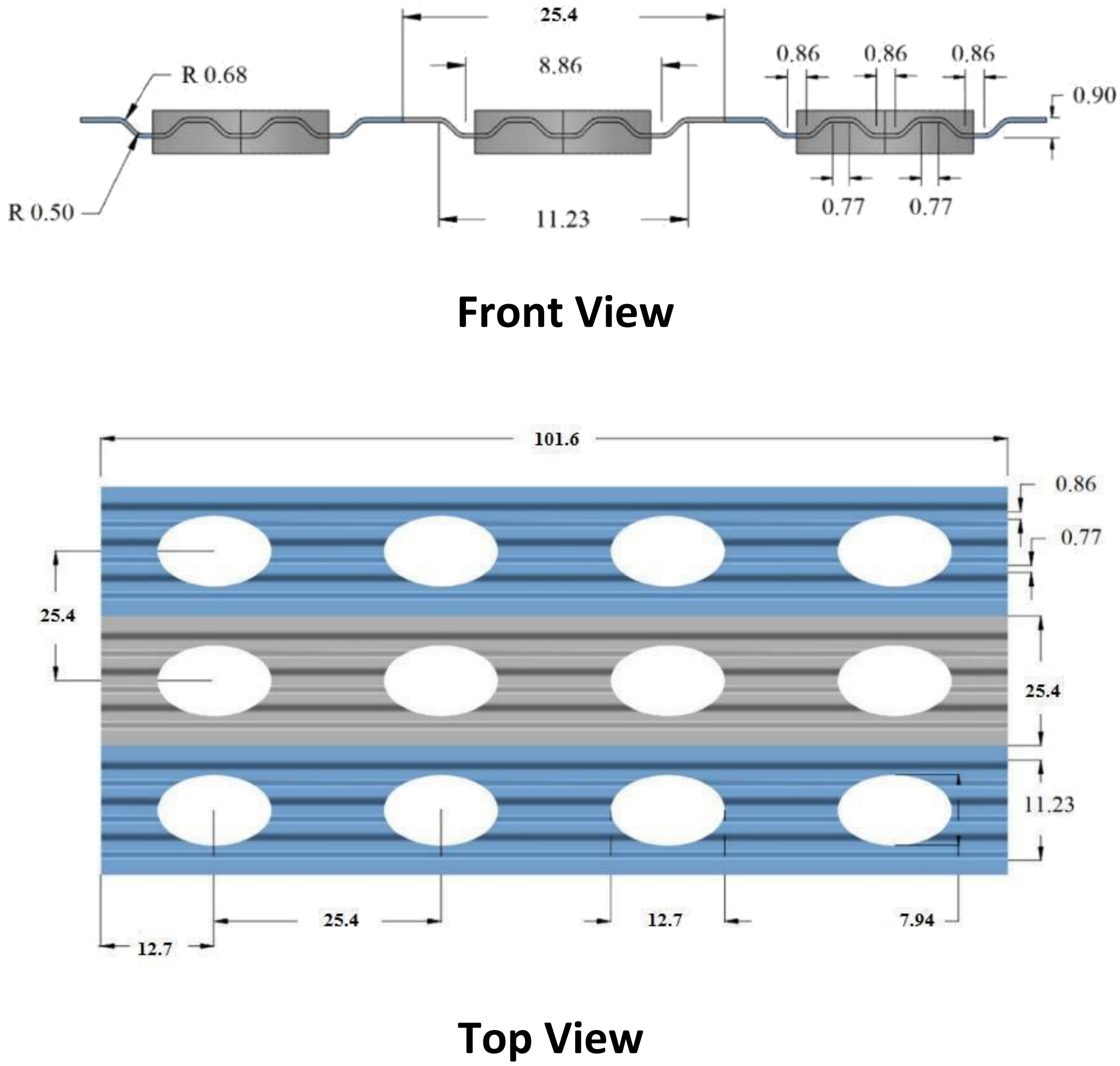


Figure 3 : Geometric characteristics of three-corrugated fin patterns: (a) top view of three-corrugated fin pattern; (b) front view of three-corrugated fin profile

Fig. 3. Geometric characteristics of Three-corrugated fin patterns: top view of three-corrugated fin pattern & front view of three-corrugated fin profile.

The first design introduced with one-corrugated fins pattern and four inline arrangement of oval shape of tubes is illustrated in Fig. 2. The second design is shown in Fig. 3 which consists of three-corrugated fins pattern with four inline oval tube banks. Due to the symmetric arrangement in the models of FTCHEs, one third of the regions employed in Figs. 2 and 3 are selected as the main computational domain. The main geometrical parameters as base values for performing the CFD simulations of onecorrugated and three-corrugated fin designs in the novel models of FTCHEs are listed in Table 1.

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_l (mm)	25.4
Fin pitch	F_p (mm)	3.0
Fin thickness	δ_f (mm)	0.1
Fin length	F_l (mm)	101.6
Fin width	F_w (mm)	25.4
Tube position from inlet	X_l (mm)	12.7
Number of tube	N	4
$D_c = D_{tube} + 2\delta_f$		

In Fig. 4 and Fig. 5the 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 spanwise direction, fin pitch direction and the streamwise 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. 4 from the entrance of flow and exterior sides of the fins. The upstream boundary can be established at a distance of one tube diameter( ). 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 > Dc) 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]. To save the space, the extended domain is not presented in scale with the fin length in Fig. 4. 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 . The thickness of the corrugated fin is the same as the base fin thickness according to [8,9,10,11].

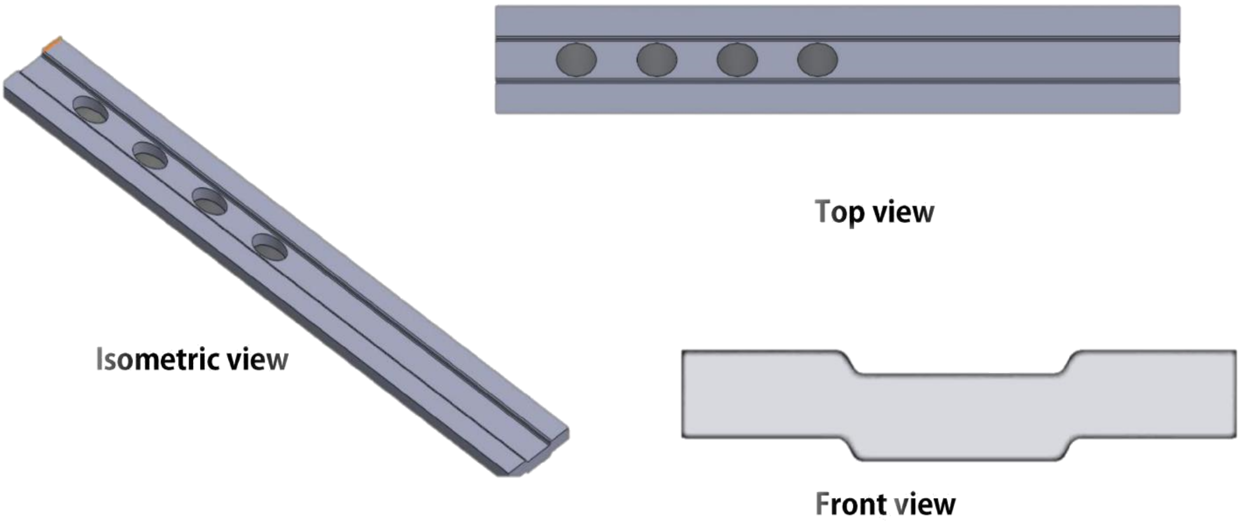


Figure 4 : Different views of computational domain for one corrugated designs

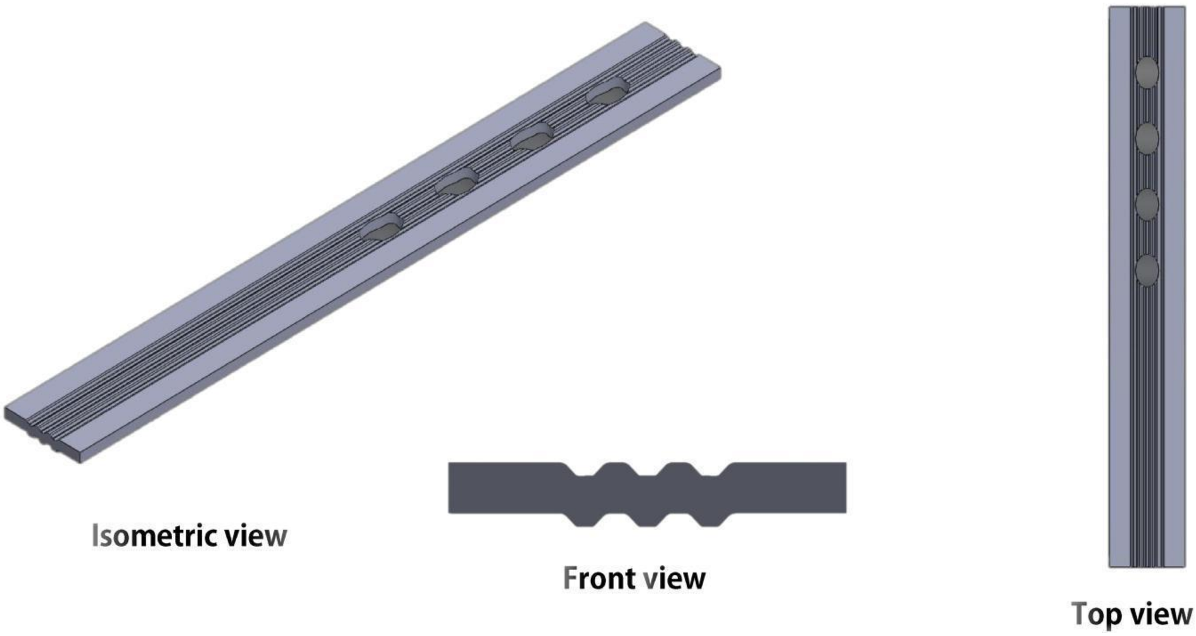
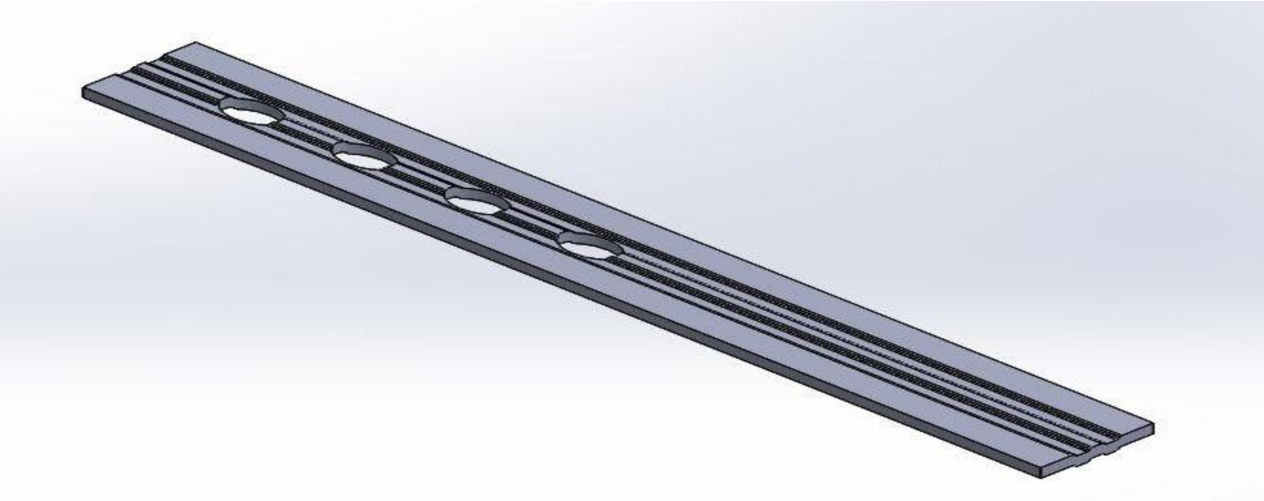


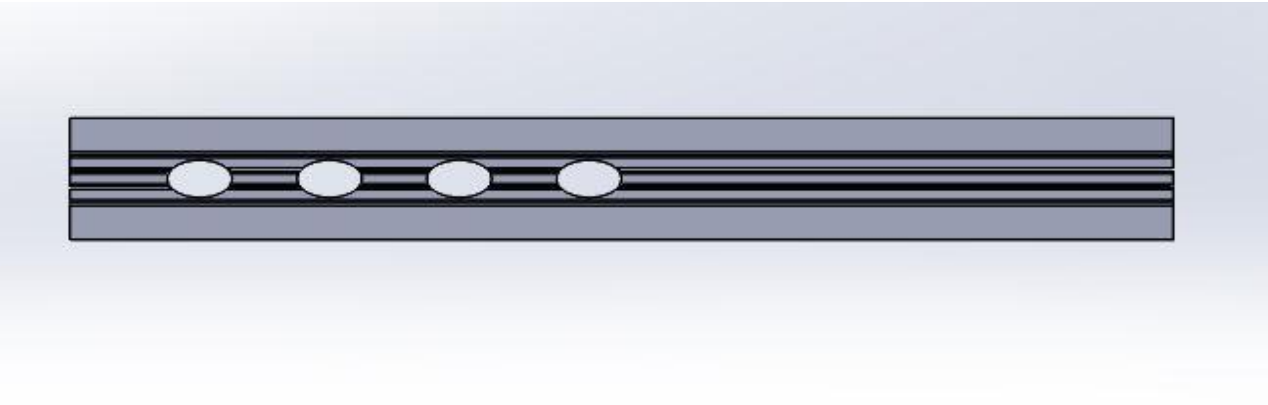
Figure 5 : Different views of computational domain for three corrugated designs

Tw

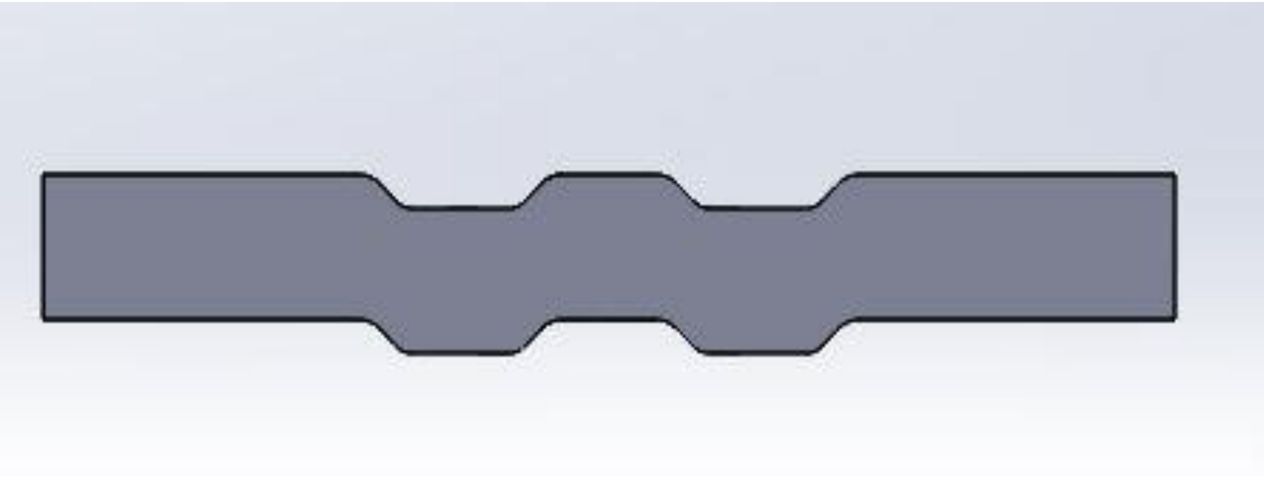
Newly Designed Two Corrugated Fin Geometry with Oval Tubes



Isometric View



Top view



Right view

Fig 6: Different views of computational domain for two 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 ($kg\ m^{-3}$)	Specific heat ($J\ kg^{-1}\ K^{-1}$)	Thermal conductivity ($W\ m^{-1}\ K^{-1}$)	Dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
Air	1.23	1006.43	0.0242	0.00001789
Fin	3600	765	36	–

4.2 Mathematical modeling and governing equations

I have used the following assumptions listed in below.

1. The incompressible flows in the airside passages are limited as laminar flow because of the small fin pitch and the low fluid velocity
2. The physical properties of the working fluid and the fin material are constant
3. The heat transfer is via sensible heat and no mass transfer is being taken place
4. Smooth surface conditions for the fin-and-tube
5. Effects of heat dissipation and thermal radiation are negligible
6. The joins between fin flange and tube is assumed to be in a perfect contact
7. Buoyancy force is neglected
8. The flow and heat transfer are in steady state
9. 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

Governing equations used for numerical simulation in the FTCHE.

Continuity equation:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0.0 \qquad (1)$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial p}{\partial x_j} \qquad (2)$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right) \qquad (3)$$

General transport equation:(for scalars):

$$\frac{\partial}{\partial x_i}(\rho u_i \varnothing) = \frac{\partial}{\partial x_i}\left(\Gamma_{\varnothing}\frac{\partial \varnothing}{\partial x_i}\right) - S_{\varnothing} \qquad (4)$$

4.3 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	

4.4 Parameter Definitions

Reynold’s Number:

- The flow condit $Re = \frac{\rho U_m D_h}{\mu}$ cterized 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-colburn factor:

$$j = \frac{h}{\rho U_m C_p} Pr^{\frac{2}{3}}$$

- It helps to predict unknown properties.

Nusselt Number:

$$Nu = \frac{h D_h}{k}$$

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

Fanning Friction factor:

$$f = \frac{2 \Delta P A_c}{\rho U_m^2 A_0}$$

- 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, $Q = \dot{m} Cp (Toutlet - Tinlet)$

Convective heat transfer coefficient:

$$h = \frac{Q}{A_0 \Delta T_m}$$

4.5 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 Physical report

Mesh Report

Mesh Information for FFF

Domain	Nodes	Elements
part_1	331413	1711315

Physics Report

Domain Physics for FFF

Domain - part_1	
Type	cell

Table 4. Boundary Physics for FFF

Domain	Boundaries	
part_1	**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 part_1	
	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.

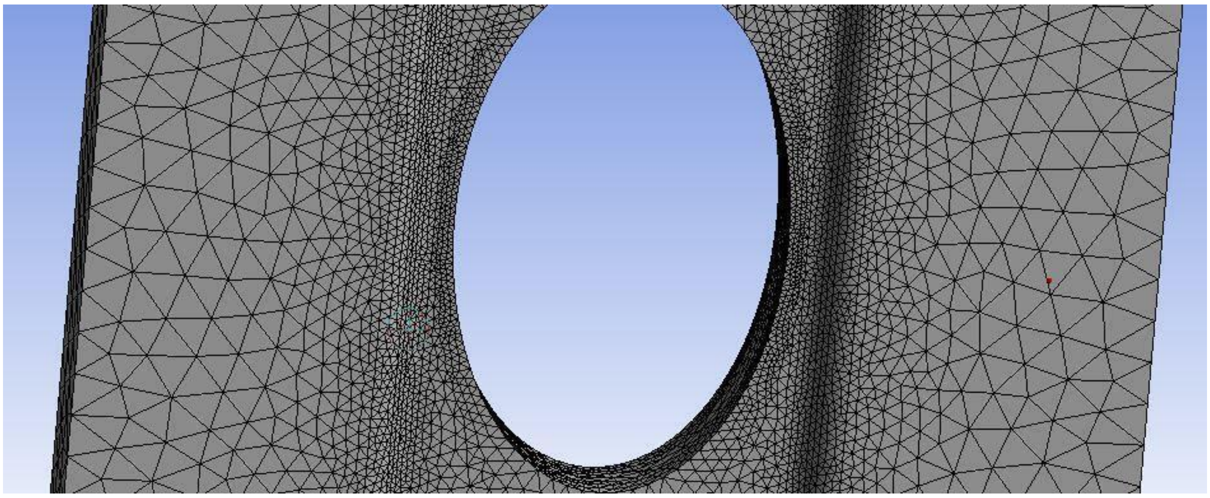


Figure 7: Grid generation on the computational model

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

Chapter 5
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.

5.1 Temperature Distribution

The variation of the temperature is shown in Figs. 8,9,10 and 11, respectively.
The range of Reynolds number used is from 200 to 900.

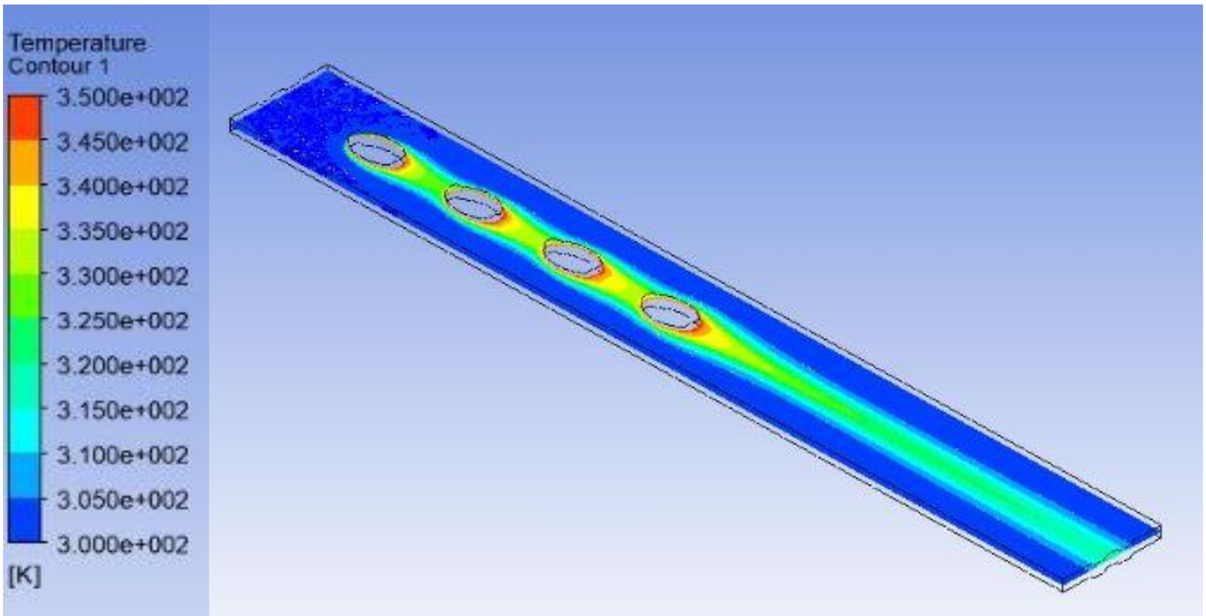


Fig 8: 2 corrugated Re 200 temperature distribution

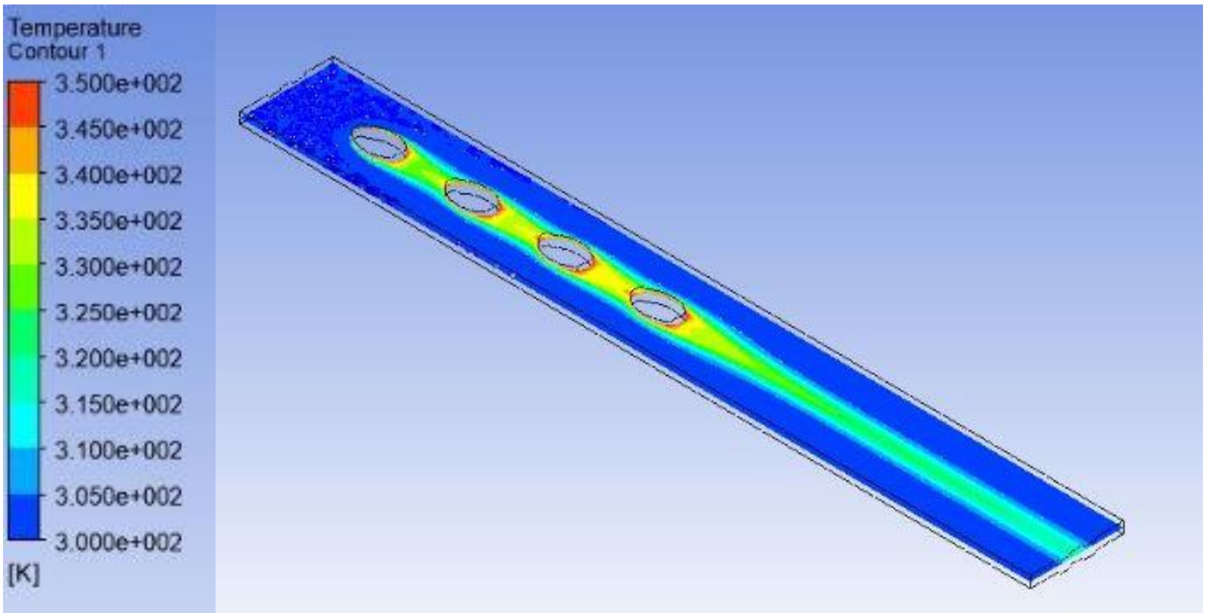


Fig 9: 2 corrugated Re 400 temperature distribution

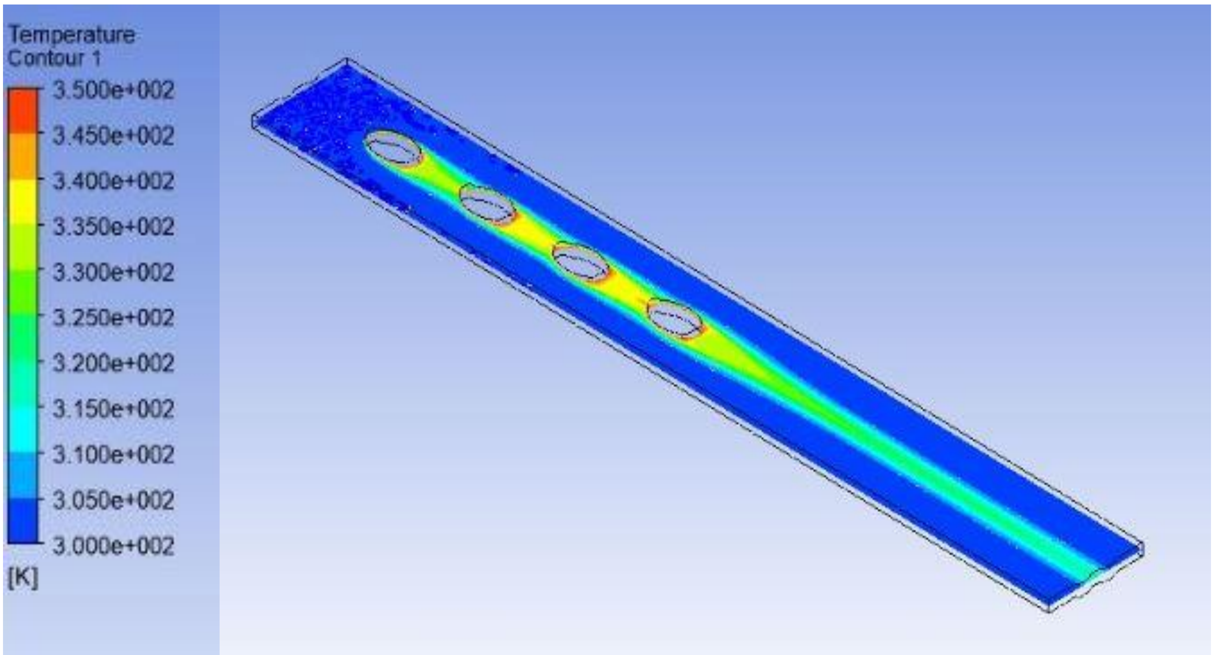


Fig 10: 2 corrugated Re 600 temperature distribution

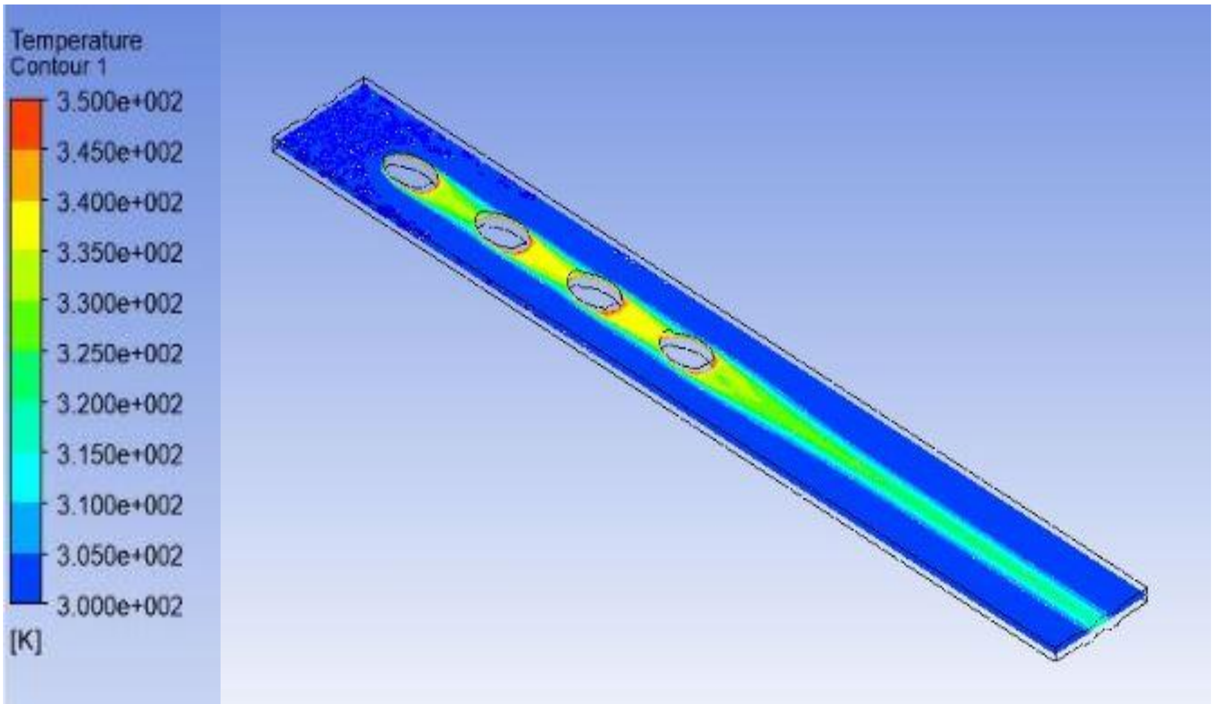


Fig 11: 2 corrugated Re 900 temperature distribution

5.2 Pressure Field

The variation of pressure distribution along the flow direction is displayed in Fig. 12,13,14 & 15. 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.

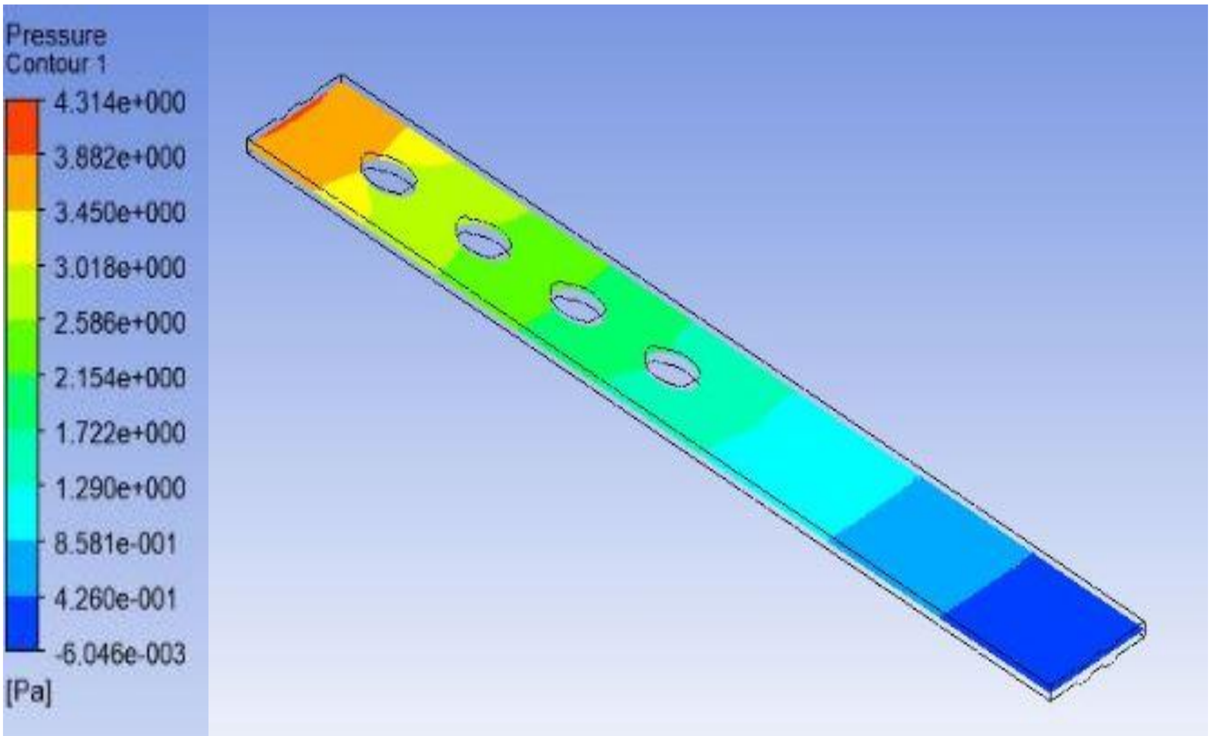


Fig 12: 2 corrugated Re 200 pressure distribution

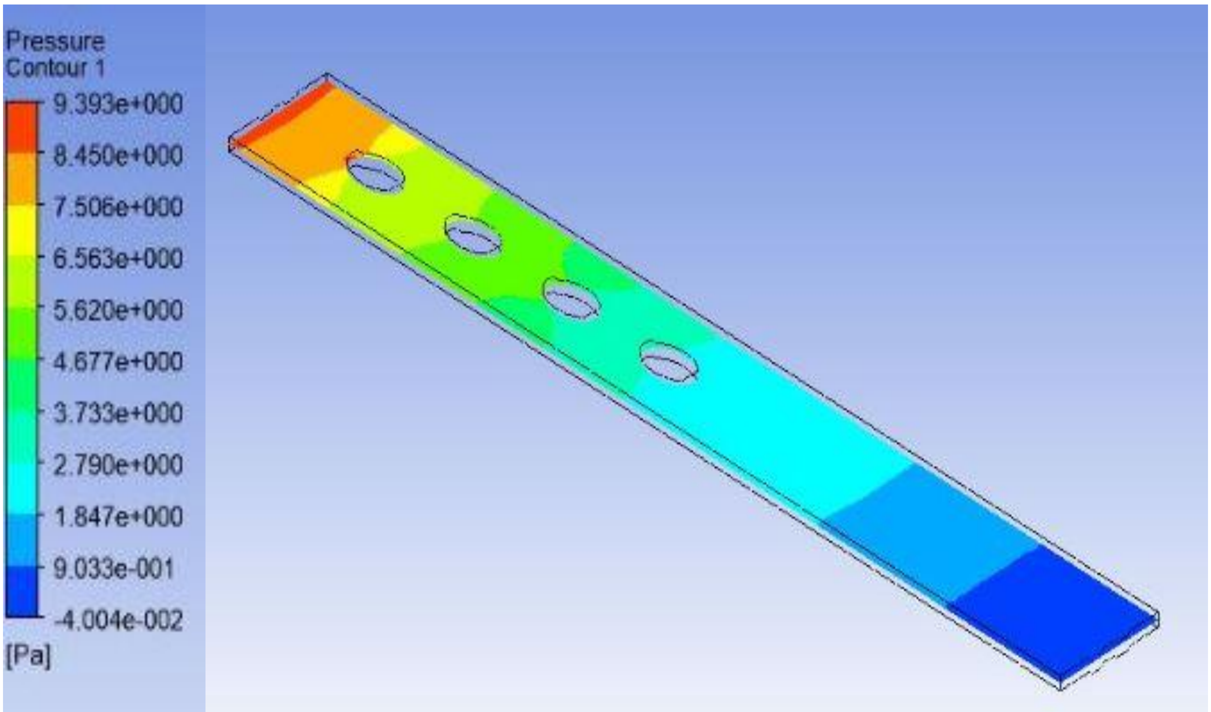


Fig 13: 2 corrugated Re 400 pressure distribution

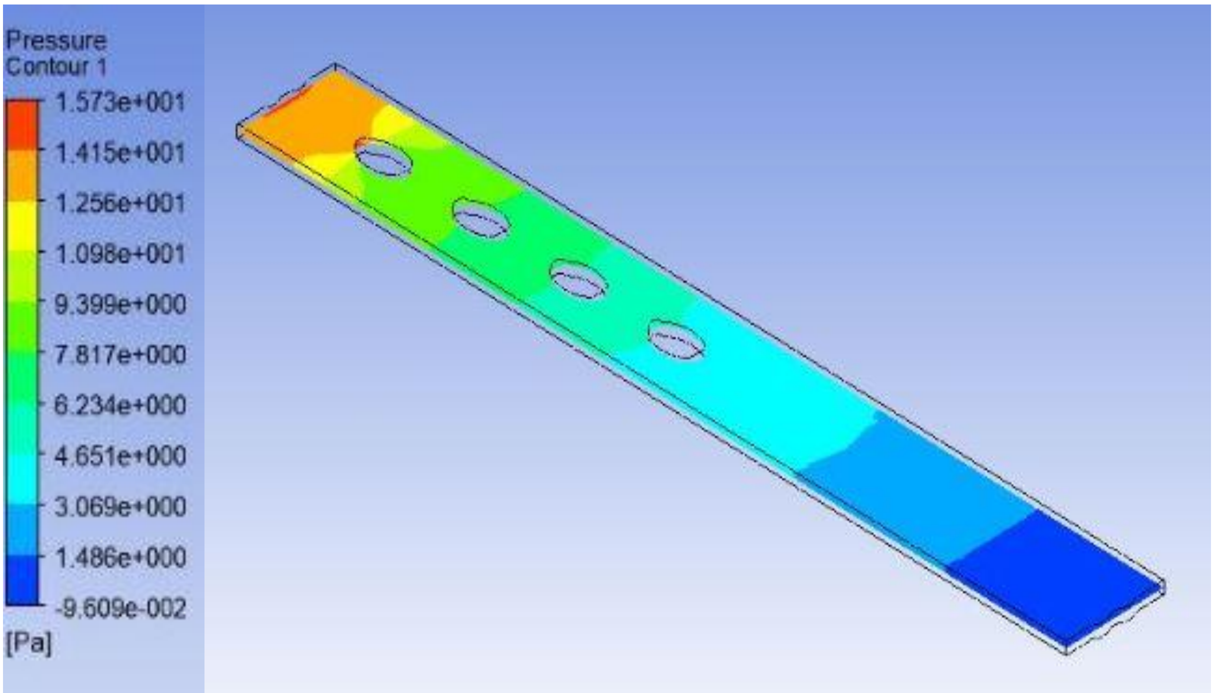


Fig 14: 2 corrugated Re 600 pressure distribution

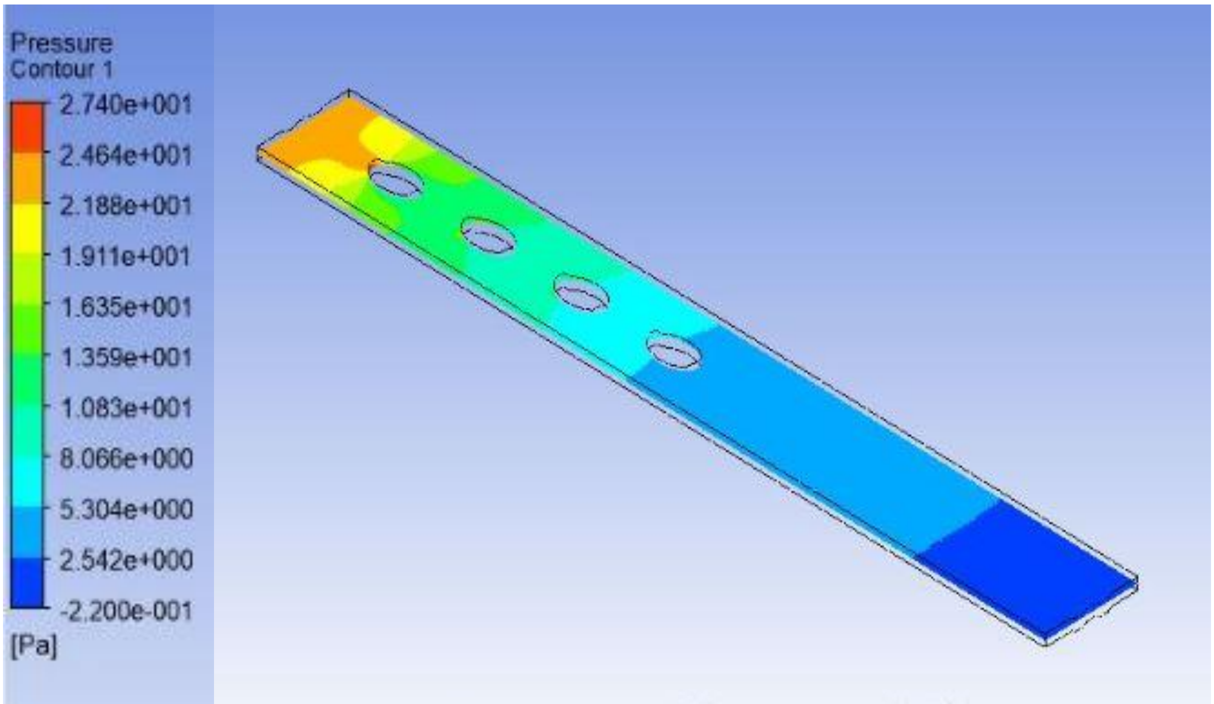


Fig 15: 2 corrugated Re 900 pressure distribution

5.3 Velocity streamline

Figs. 16,17,18 and 19 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.

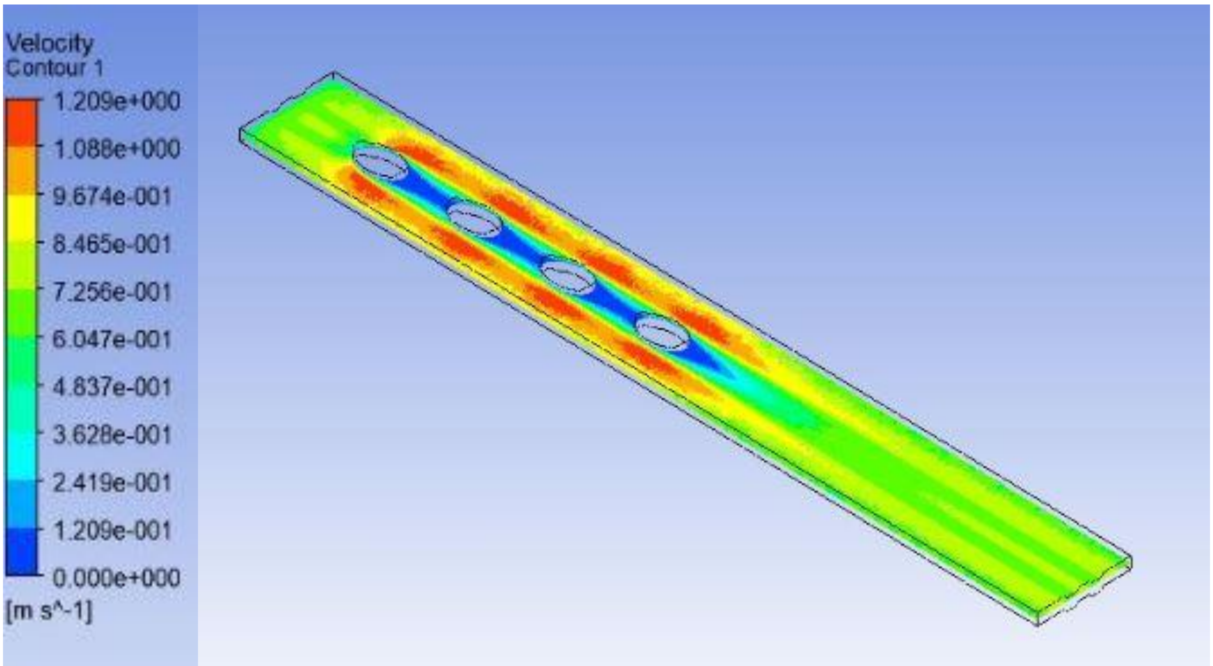


Fig 16: 2 corrugated Re 200 velocity distribution

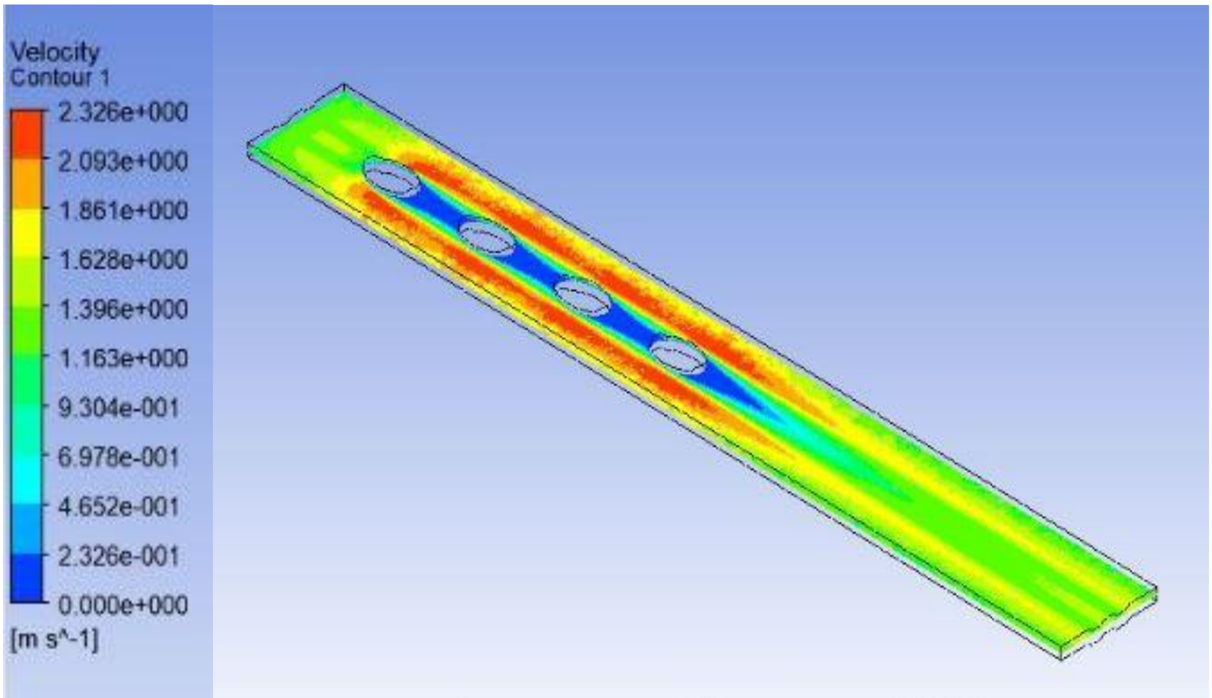


Fig 17: 2 corrugated Re 400 velocity distribution

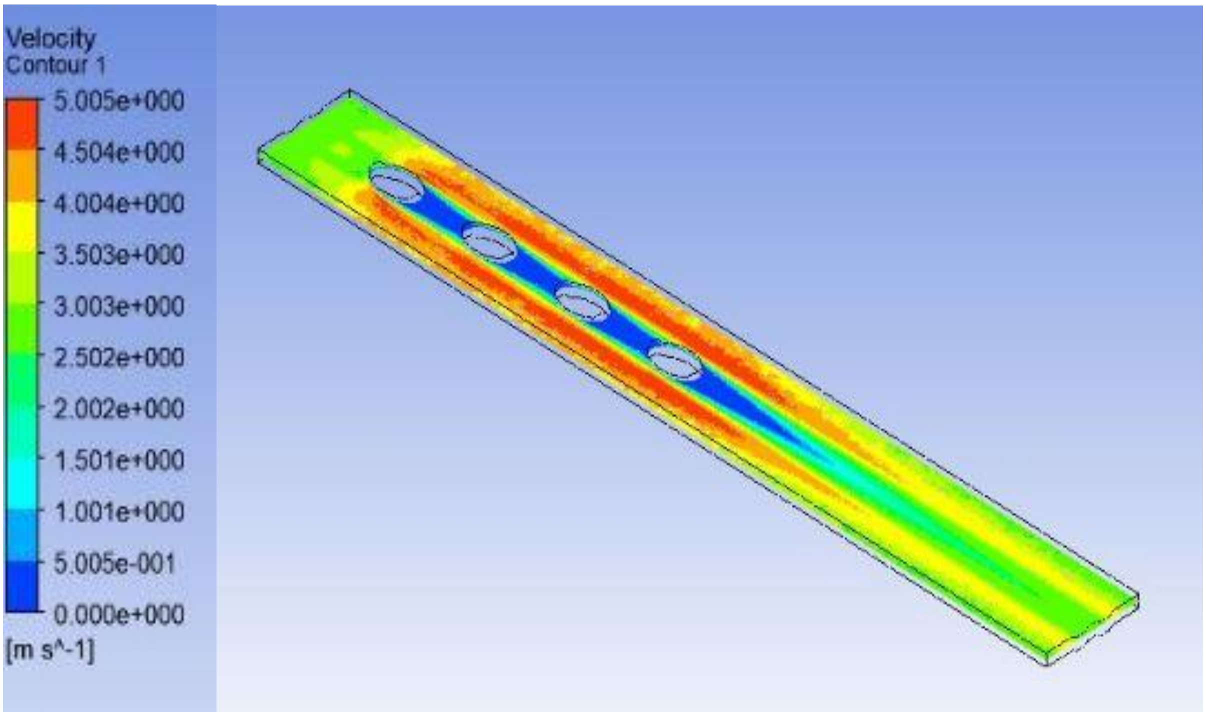


Fig 18: 2 corrugated Re 600 velocity distribution

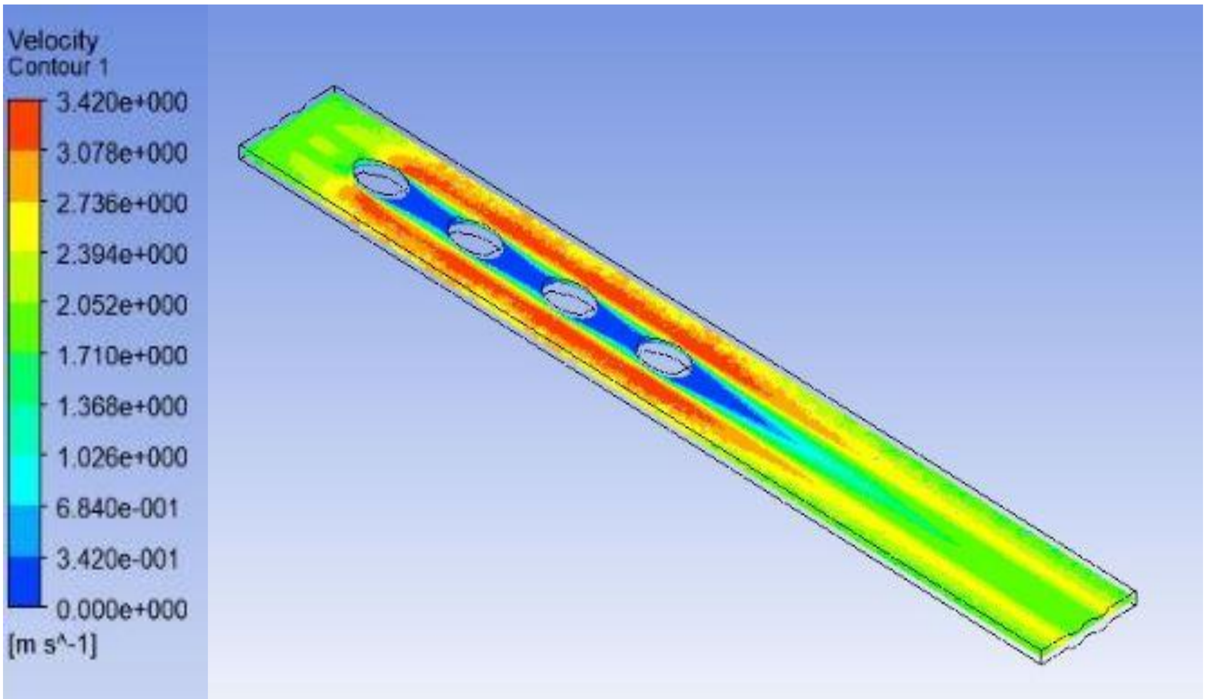


Fig 19: 2 corrugated Re 900 velocity distribution

5.4 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. We also compared the results with the literature referred here.

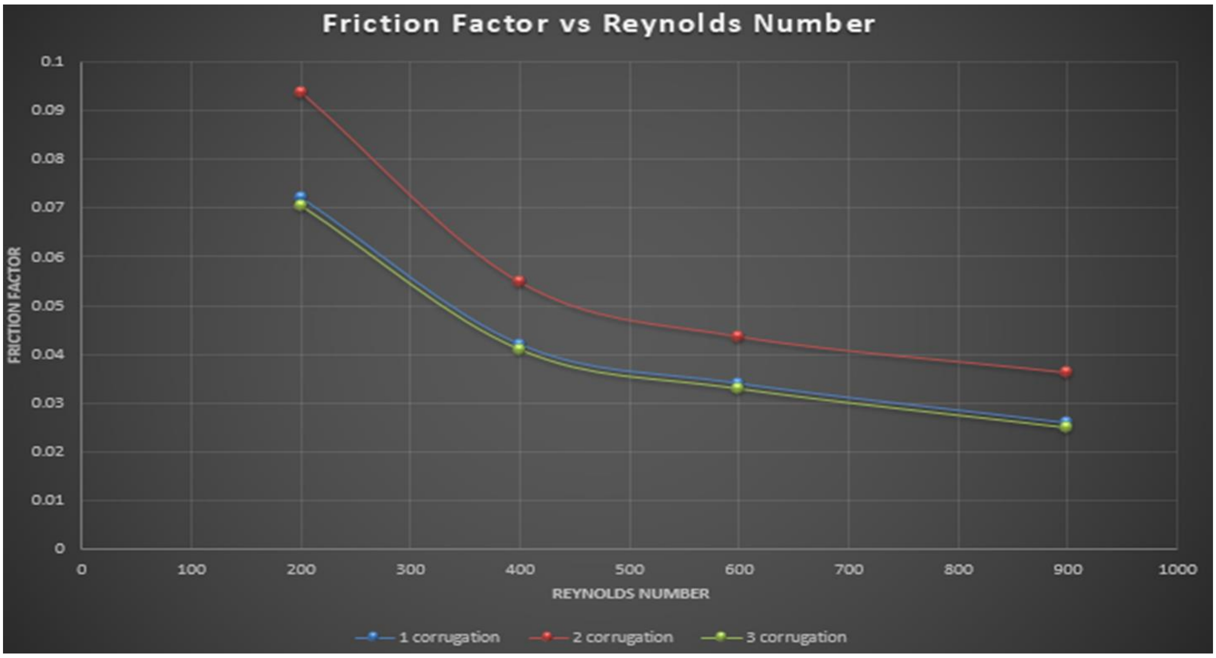


Fig 20: Friction Factor vs Reynold's Number

5.5 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.

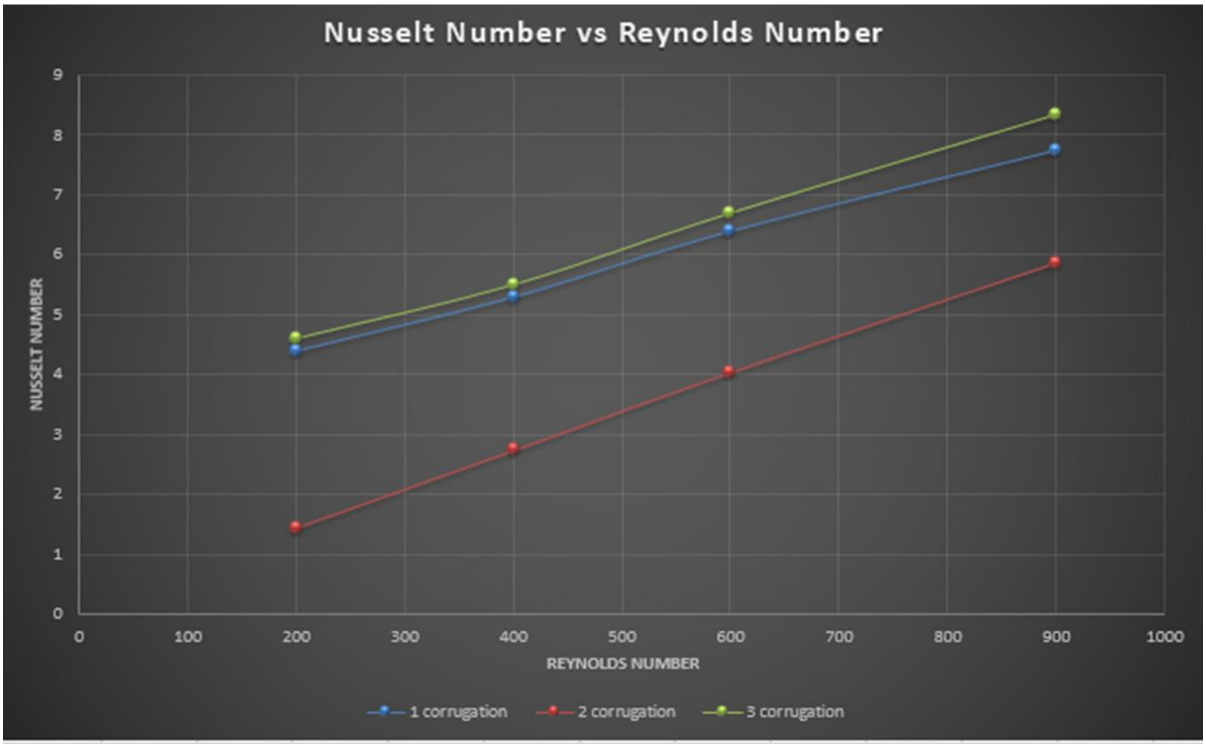


Fig 21: Nusselt Number vs Reynold's Number

5.6 J-colburn factor

We 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.22 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.

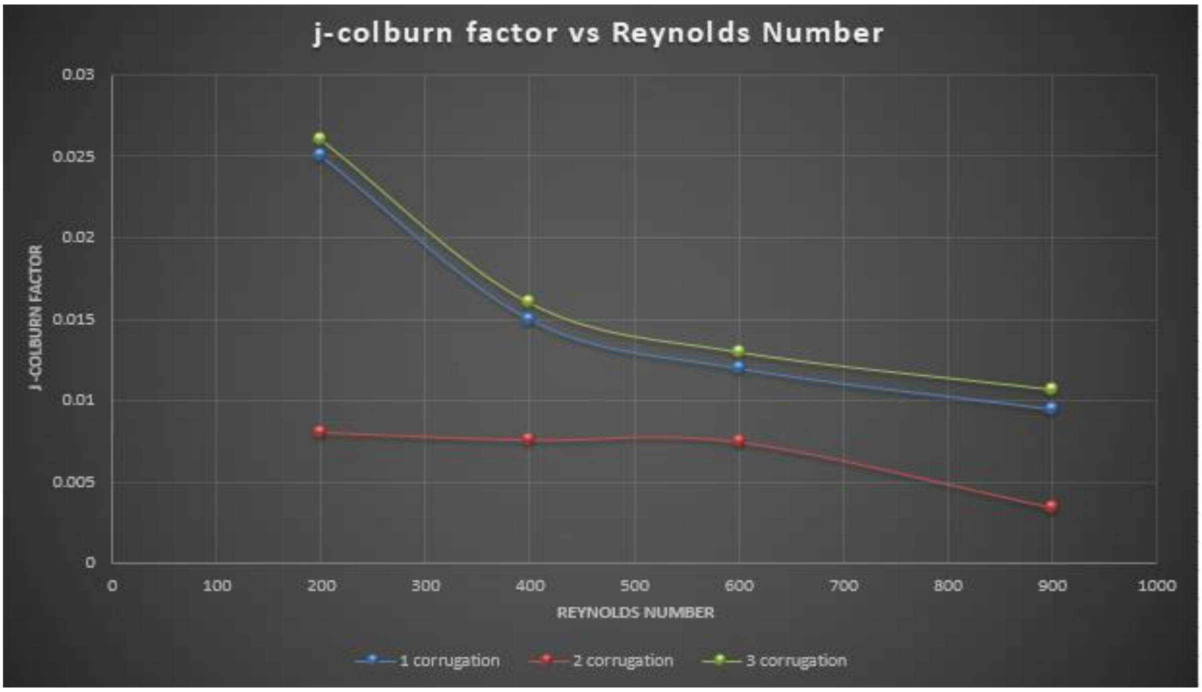


Fig 22: j-colburn factor vs Reynold's number

Chapter 6

Conclusion

This paper was presented 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, two corrugated and three-corrugated designs considering curvature regions of corrugated patterns in the tube directions. 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. Some remarks are as follows:

- 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.

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