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Numerical Evaluation of Thermal Hydraulic Performance in Compact Heat Exchangers with Vortex Generator

B.Sc. Engineering (Mechanical) Thesis

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CERTIFICATE OF RESEARCH

The thesis title "**Numerical Evaluation of Thermal Hydraulic Performance in Compact Heat Exchangers with Vortex Generator**"

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Candidate's Declaration

It is hereby declared that, their thesis or any part of it has not been submitted elsewhere for the award of any degree or diploma.

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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^2K]
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 (hD_h/k)
m_f	mass flow rate [kg/s]
P	pressure drop [pa]
P_l	longitudinal pitch [m]
P_r	Prandtl number

P_t transverse pitch [m]

Q heat transfer rate [w]

X_l Tube position from inlet [m]

Re Reynolds number

Re_{Dc} Reynolds number based on D_c

Re_{Dh} Reynolds number based on D_h

S volumetric source term

T temperature [K]

U_m mean velocity at the minimum flow cross-sectional [m s^{-1}]

δ_f fin thickness [m]

μ dynamic viscosity [kg/ms]

ρ density [kg/m3]

г diffusion coefficient

φ transport property

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ABSTRACT

The improvements in heat exchanger performance have long attracted many researchers as they are of great technical, economic, and ecological significance. The performance of the fin-and-tube heat exchanger (FTHE) can be enhanced with the change of design of compact heat exchanger. The main focus of changing the design is to increase or decrease the temperature and enhance heat transfer from one fluid to another. The key incentive of current research is to investigate the influence of various vortex generator in the FTHE and the parameters that enhance the thermal and hydraulic performance in the FTHE such as the geometry and arrangement. Three different geometry arrangement of corrugated profiles in FTHE such as ONCF (corrugated fin with one fluted domain), TWCF (corrugated fin with two fluted domains), and THCF (corrugated fin with three fluted domains) shapes are investigated by a parametric design exploration technique. The main objective of this research is to conduct the numerical analysis in compact fin and tube heat exchanger and to study thermal–hydraulic performance characteristics in FTHE with introducing new design of fins (ONCF,TWCF,THCF) and tube, determine the effect of different size and shape of vortex generator (VGs) by using computational fluid dynamics approach. The main purpose of introducing such design is to increased thermal efficiency and performance criteria of FTHE. The introduction of vortex generators behind tubes resulted in heat transfer augmentation but come together with higher pressure drop penalty which enhance the performance. The investigation of thermal–hydraulic performance criteria is conducted for Reynolds number in the range of 200–900. The outcomes of study indicated that the average Nusselt number for the FTCHE with corrugated fin can be increased up to 23%. The newly designed oval-tube fin with 3mm square shaped VGs shows a potential increase in the efficiency of the heating transfer and a mild pressure loss on the FTCHE compared to the other model.

1. INTRODUCTION

Heat exchanger (HX) is a widely used device in the field of HVAC&R that allows heat exchange between hot and cold fluids. In the heat exchanger device, the fluids in different temperature are separated by solid conducting wall. This solid conducting wall between the heat exchanging fluids is known as heat transfer surface. The phase of the fluids can be identical or contrasting depending on the nature of the device. The major parameters for classifying a high performance heat exchanger are higher thermal efficiency, lower pressure drop, economical fabrication cost and light weight characteristics. The motivation of increasing heat transfer rate has driven the researchers to design various kinds of heat exchanging devices for the past several decades. The primary and most concerning problem that has been faced by the investigators and innovators while designing the variations is the increment of pressure drop. Significant pressure drop is responsible for decreasing the thermal efficiency of these thermal devices despite of having higher heat transfer rate. Different studies and investigations are being performed to optimize the performance of the HXs.

HXs are categorized according to their variations in construction, heat transfer method, phase of fluids, flow arrangement, pass arrangement and heat transfer surface compactness. Among different kinds of HXs, compact heat exchangers (CHEs) are now stimulating prodigious heed to the researchers for their better thermal efficiency and less pressure loss characteristics. The main feature of a compact thermal device is higher area density which means large heat transfer surface area per unit of volume. Area density of CHEs is conditional on phase of liquids. The threshold value of area per unit of volume for gas to gas and gas to liquid CHEs are greater than or equal to 500 m^2/m^3 or 700 m^2/m^3. On the other hand, the ratio is greater than or equal to 200 m^2/m^3 or 400 m^2/m^3 for the liquid to liquid type CHEs.[1, 2]When one or both the fluid phase is gas, the enhancement of the performance of CHEs becomes indispensable. Area density on the gas side needs to be higher as the thermal resistance of gases become 10-15 times higher than liquids [3].

The flow propensity becomes laminar through the channels and causes ascending pressure loss due to the compact flow passages. The efficiency must be improved because laminar flow is related to small heat transfer coefficients. This is the reason, researchers are developing different heat transfer method which has created diversity in construction of CHEs [4]. Fin-and-tube heat exchanger is one form of compact heat exchangers. In this type of heat exchanger, a heat exchanging medium such as water, oil or refrigerant is forced to flow through the equally spaced parallel tubes of in-lined or staggered arrangements. On the other hand, a gas such as air is directed across the tubes in a lump of alternating layers of fins. Different types of tube shape, fin geometry are being designed to optimize the thermal-hydraulic performance of fin-and-tube compact heat exchangers (FTCHEs).

Researchers and designers have introduced various active, passive and compound techniques for enhancing thermal-hydraulic performance of HXs in past few decades. Both experimental and numerical investigation have been performed for achieving higher efficiency of heat exchanging devices by changing different designs and techniques [5-8].

Active techniques are basically applied by external competences to improve heat transfer performance. Electromagnetic field, fluid shake, surface quake, instalment of different mechanical systems, boundary layer suction are some widely used active techniques in industrial application of heat transfer. Electrostatic fields are combined with magnetic field which create force convection [9]. Fluid shaking is applied for single phase flow which is the most common application of active techniques. Surface vibration is also widely used to exploit the boundary layer for having better heat exchange performance. When these techniques are harmoniously used that is called compound technique [1]. In a study of [3], a detailed overview has been shown how these techniques can bring out the optimum and cost effective performance of CHEs.

Passive techniques are not dependable on any external competences. This results in less cost of heat transfer device with higher efficiency over active techniques. In CHEs different types of fin construction such as plain, wavy, corrugated; tube formation such as in-lined, staggered; various tube shapes such as oval, circular, rectangular; vortex generators, winglet angle modification and many more competitive modifications are introduced as passive techniques to increase performance [10, 11]. Sthhlik et al [12]presented a passive technique in a longitudinal finned tubes where performance was improved by enhancing heat transfer area and coefficient. The study shows that modifying the geometries can bring better result in fin side performance and reducing the area is required on gas side.

Numerous experimental and numerical simulation have been performed for examine the effect of design modifications for constructing efficient CHEs. Beecher et al [13] first performed a detailed study on heat transfer characteristics of wavy fin with circular tube in 1987 where correlations were developed to describe the colburn factor and friction factor. After that different experimental investigations on convective heat transfer were arranged for 2D channels documented in the literature of [14] and [15]. Experimental study on 3D wavy fin patterns were also performed for developing correlations of Colburn factor and friction factor to predict heat transfer and flow characteristics which are mentioned in (Wang, 1997), (Wang, 1995), (Wang, 1998), (Wang, 1999), (Yan, 2000) [16-20]. Experimental study set up by Guo and Tafti, 2003 [21] found that the influence of inlet flow angle of wavy fin pattern has significant influence on correlation between heat transfer coefficient and flow efficiency. Better heat transfer performance was found in a good offset strip fin than the perforated fin in an experimental study done by Shah [22]. Colburn factor against Reynolds number was presented for plain and wavy fin patterns in the experimental investigations of Wang et. al. [16, 23]

A review on both experimental and numerical analysis was demonstrated to increase thermal-hydraulic performance of HXs by using various types of vortex generator as a passive technique [24]. Bhutta et al [25] conducted an assessment that showed CFD method as an cost effective way to design and optimize the performance of HXs. The variations between experimental investigations and numerical simulations were 2-10% though it was up to 36% in some typical cases. This study was concluded with a validation of reliability of using numerical modeling for speedy solution and recommended stamping out the building of prototype. Jang et al, 1997 [26] showed in a numerical set up that different geometric properties has influence on the efficiency of wavy finned HXs. A comparative review of CHEs showed significant increment of nusselt number and overall heat transfer coefficient by changing the flow channel design from flat to wavy pattern with a penance of large amount of pressure drop. Boundary layer cannot be formed because of this changed fin geometry. The unstable boundary layer produce Görtler vortices and transported downstream. Creation of reverse flow and eddies were also found in [27] investigation. Possibilities of enhancement of heat transfer rate by introducing small corrugated 2D channel in HX was investigated by Rutledge and Sleicher in a numerical study [28].

Numerical visualizations were performed by Bhuiyan [29] where the influence of tube configuration, geometric pattern such as wavy angle, pitch, inlet flow angle was quiet evident in heat transfer performance of wavy FTHEs. Higher pressure drop and heat transfer were found in staggered formation than in line formation of tubes. Bhuiyan [30] also investigated the effect of different geometrical parameters of plate fin and tube heat exchanger (PTCHE) in turbulent flow regime by using CFD. The investigation showed reduction of heat transfer and pressure drop performance with the increment of both longitudinal and transverse pitch of fin. But opposite result was found for these cases by increasing fin pitch. Panse, 2005 [31] arranged an investigation in transitional flow regime for staggered arrangement in wavy fin and found the influence of wavy height and angle on heat transfer performance. A numerical study on multi row FTHE was performed by Wang Zhang keeping the flow laminar and three-dimensional. The study showed that colburn factor was 63-71% higher and friction factor was 75-102% higher in wavy fin pattern than the plain one. The investigation also demonstrated that in a four row wavy fin pattern, nusselt number decreased gradually from first tube to last tube [26].

A numerical study investigated the longitudinal tube pitch as one of the controlling factor of designing efficient heat transfer devices like CHEs. The numerical investigations showed irregular change in heat transfer coefficient due to uneven pitch ratio. Influence of transverse pitch was also found by maximizing Reynolds number at a minimum cross section of the fins [32].

Different types of fin geometry and modifications have been introduced in all these experimental and numerical investigations for better heat transfer performance. Variation of fin designs such as plane fins [33], louver fins [34-38], offset strip fins [39], perforated fins, micro fin tubes [40], sinusoidal fins [41, 42], wavy fins [16], herringbone wavy fins [43] etc have shown different competitive performances in both numerical and experimental study.

Bhuiyan presented a review on different fin patterns such as louver fin, plain and wavy fin, offset strip and perforated fin. The review showed that comparatively higher heat transfer rate was found giving higher pressure loss as penalty by making the flow path lengthy in wavy or corrugated fin patterns [44]. A numerical prediction in laminar flow regime arranged by Bhuiyan showed significant influence of flow distinction between plain and wavy fin configuration. Larger recirculation zone in plain fin pattern than wavy one is the differentiating factor [33].

Modification of different construction parameters such as fin pitch, tube shape, tube pitch, tube arrangements, number of tube rows are also responsible for bringing variation in the performance of CHEs [33, 45-47]. In the studies of [48-50] in depth investigation was performed for plain and wavy fin and tube CHEs in different flow regime. Among few researchers Mcnob [51] performed 3D simulations where side wall effect was considered to investigate thermal-hydraulic characteristics such as tube banks with fin. In some research fin spacing has also been found as one of the reasons of difference in performance of CHEs. Fin spacing can be defined as the distance between two fin. There are some contradictory results about the effect of fin spacing on heat transfer performance in CHEs [52]. It was found in a study of Rich [53] that though there was ramification of fin spacing on heat transfer performance for the Reynolds number below 1000, the impact started to decline when the Reynolds number went above 1000. A numerical study and flow visualization [54] have been performed to investigate the effect of fin spacing on heat transfer performance in a plate fin and tube heat exchanger keeping all other parameters constant. The flow becomes Hele-Shaw flow for small fin spacing. There is no creation of vortices in upstream as the fins neutralize the vortex generation.

The condition of the wake region behind the tubes becomes poor which is responsible for little nusselt number. With the increment of fin spacing, the heat transfer rate in wake region has been enhanced. Better nusselt number and less pressure drop has been found because of increasing fin spacing. An experimental study [55] shows that nusselt numbers increase up to a peak value by reducing fin spacing but it starts to drop by further decrease of fin spacing. Maximum heat transfer was found when the height of fin was taken within 2.5 mm to 6 mm for corrugated fin type in [56]

In a study of performance evaluation criterion (PEC) [57] was performed to optimize the heat-transfer performance of CHEs where heat exchanger length was evaluated according to different fin geometry. The evaluation showed the necessity of variation of number of tube rows and the longitudinal tube pitch with the change of fin length. Both Reynolds numbers and the longitudinal tube pitch have influence on friction factor and j colburn factor. This study also showed incorrect performance evaluation while neglecting longitudinal tube pitch for limited number of tube rows. Types of tube arrangement also play a vital role for the performance evaluation of fin and tube compact heat exchangers. The influence of in line and staggered formation of tubes in turbulent flow regime was compared with transitional and laminar flow in a numerical analysis of [48] and better performance was found in staggered configuration. Bouris et al. [58] suggested an configuration of six row tube bundle where drop-shaped cross-sectional arrangements were designed. The investigation showed large amount of decrease in fouling factor and pressure loss and improvement of heat transfer rate. A comparative numerical analysis of HX was performed for the tubes of different diameters in tandem arrangements maintaining cross flow. The result of the analysis showed around 30% reduction of fouling rate for unequal tubes in diameter which were placed in largest transverse spacing. The pressure loss and heat transfer rate of per unit volume was in acceptable range [59]. 3-D numerical simulation [60]presented optimized performance for three different types of tube configuration in HX with rectangular fin and elliptical tube.

This study investigated both overall performance and local thermal behavior of three variations mentioned earlier. The optimization result of two way interactions was close and influential for the in-line and staggered row arrangements while it was found marginal for staggered column arrangement. Dimensional parameters such as fin thickness, fin pitch and transverse tube pitch were also optimized for these three designs. Hovart and Markov [61] found lower mean values of drag coefficient by using wing shaped tubes in a staggered configuration instead of cylindrical tubes. In the study of [62, 63], it is evident that the relative distance between a pair of tubes largely dependent on Reynolds number, turbulence intensity, tube surface roughness etc. A critical spacing between two tubes has been determined in these study for better performance.

Various regression method has been applied to develop correlation between friction and heat transfer coefficient in plain fin and tube heat exchangers with small tube diameter where Reynolds number and different geometric parameters are the variables [64].Influence of tube shape while designing a FTCHE is significant for getting better heat transfer performance in the wake region behind the tubes. In an investigation of [65], 80% increment of thermal hydraulic performance has been found by modifying iso-sectional tube compared to circular shaped tube by performing Unsteady-RANS simulations. The study has also showed that significant improvement of wake region behind the tubes could be possible by reducing ellipticity of the tubes. In a study [66], a combination of elliptical and circular tubes is used and compared with the circular or elliptical tubes used alone. The heat transfer is more enhanced at low velocity for the elliptical tubes followed by circular tubes compared to circular tube alone. At high velocity, combination of elliptical and circular tube shows better performance than the elliptical tubes alone.

Circular tubes are easier to manufacture but the problem is the high pressure drop driven by large wake regions and flow separation. Non circular tubes have become a major interest in cross-flow heat exchangers as they have lower pressure drops compared to circular ones. In a study of [67], three geometries have been considered: circular, elliptic and wind shaped. The effect of these geometries has been studied with an average nusselt number and also considering the total generation of entropy. For each geometry, different correlation was found for large Reynold's Number. There is an optimum value of Reynolds Number for which the study have found a heat transfer between the fluid and solid surface with an effective heat flux. When Reynolds number is greater than $1.5x10^4$, elliptic and wing shaped geometries are far better than the circular ones and if the Reynolds number is greater than $2.3x10^4$, elliptic shapes give better overall efficiency compared to the other two. In an investigation of [68], oval tube wavy fin has 0.3-0.8% and 9.3-10.1% better efficiency than the big and small circular tubes employed in the heat exchangers respectively. In case of louvered fin, this reaches to 3.2-6.6% and 26-28.4% which is much higher than the wavy one.

An experiment [69] has been performed where different parameters like Reynolds number, diameter ratio and longitudinal pitch have been considered with values 3900, 0.5 and among 1.5, 2.0 and 2.5 respectively. The wake region always has a variation in velocity. This variation reduces when a pair of cylinder is used. This tandem arrangement gives an efficiency raise up to 40% compared to a conventional arrangement. In another study [70], a mixed tube bundle has been considered with circular and elliptical tubes. The tubes are arranged in inline and staggered configuration. A numerical simulation has been done to find out the thermal performance and a comparison is also done among mixed, circular and elliptical bundle of tubes. The convection heat transfer coefficient is lower, and the pressure drop is higher for elliptical tubes compared to the other two geometries. Though the heat transfer is higher for staggered arrangement, the pressure drop is also higher compared to inline tube bundle. A literature review [71] has been performed emphasizing on the curved tubes namely helical coil, spiral coil and other types of tubes. There are very less studies done on spirally coiled tubes compared to the other two.

A numerical simulation [72] has been done in this study to get the effect of elliptical finned tubes on the heat exchanger. Response surface methodology has been used to get the results. There are numerous factors considered namely axis ratio, longitudinal tube pitch, water volumetric flow rate, pitch of the fin, velocity of air etc. Rising of air velocity as well as the axis ratio causes the increment of the overall performance of the finned tube heat exchanger and at a lower velocity the opposite occurs. In [73], condenser has been considered as a heat exchanger and the study shows that elliptical finned tube condensers have higher heat transfer rate and lower pressure drop compared to circular ones and the amount of variation between them is 37% and 27% respectively.

A CFD simulation [74] has been performed where an elliptical fin tube heat exchanger has been considered with different elliptical ratios for finding out the effects on overall performance. This ellipticity ratio is between 0.6 and 0.8. The elliptical tubes are used at different inclinations to see more effects. J colburn factor increases up to a maximum point with the increment of inclination. Keeping the elliptical ratio 0.6 as minimum and maintaining inclination angle 20^0 maximum efficiency has been found. In a numerical investigation [75], that oval tubes compared to flat and circular tubes have the optimal configuration with 4.99% and 39.94% pressure drop at Reynolds number 400 and 900 respectively. The convection heat transfer coefficient raises to 13.99.

By reviewing all these investigations, it has been found that heat transfer rate and thermal characteristics can be improved with moderate pressure loss by designing different wavy fin geometries and shape of tubes. Gholami [76] showed significant improvement of thermal-hydraulic performance by designing one and three corrugated fins and oval shape tubes. Performance of CHX with plane fin and circular shape tube has been compared with newly designed CHXs and better results have been experienced.

In this study, corrugation of fin has been designed by changing some design parameters which is three corrugated fin. A numerical investigation has been performed for this newly designed CHX and compared with other corrugated designs. A gradual improvement of the thermal-hydraulic performance has been found for Reynolds number 200, 400, 600 and 900. Better results in wake region behind the oval tubes, improved nusselt number, j-colburn factor has been investigated with the increase of corrugation in same fin. Then taking three corrugated fin as the best option we added vortex generator for further enhancements .We designed vortex generatorof different size and shapes like square ,circle star .We also changed the shape of vortex generator and the number of vortex generators in three corrugated fin.We simulated each and every case for different Reynolds number starting from 200 to 900 .Then we analyzed heat transfer characteristics and thermal performance of each case and came to conclusion of suitable vortex generator design with higher heat transfer enhancement.

2. MODEL DESCRIPTIONS

2.1. PHYSICAL & COMPUTATIONAL MODEL

The geometry of FTCHE with one, two & three corrugated fin patterns & oval tube arrays are shown in the Fig 1,2,& 2. Here one different design of FTCHE with two corrugation fin pattern has been introduced based on the researches [76-80]. The corrugation pattern of fins have impact on both side of each fin to fluid flow characteristics. Corrugated patterns are designed as wavy forms having a series of curves with radii 0.68 mm and 0.50 mm which ae set at an angle of 45^o for the up and down side of the fin patterns respectively [76]. All the corrugated fin designs consists of 4 oval shape tubes with uniform spacing. The full detailed geometrical properties are mentioned in Table 1 for performing CFD simulations by the ANSYS Fluent 2012. The real computational field was stretched by $F_w/2$ ($F_w/2 > D_c$) to maintain more uniform and consistent inlet velocity. The exit domain was extended by F_1 for having a flow pattern to avoid recirculation flow and completely developed boundary conditions in outlet according to the research done by P.Cehuerl[81].

The core region of fin-and-oval tube compact heat exchangers having one, two and three-corrugated fin patterns and oval tube arrays is demonstrated in Fig. 1. FTCHE with two corrugated fin design has been introduced in this research [76-80].The pictorial representation of top view and front view of the corrugated fin shapes are shown. These kinds of fin patterns influence on both sides of each fin to fluid flow behavior. Corrugated patterns look like waves with a series of curves with radii 0.68 mm and 0.50 mm and included angles 45^0 for the up and down sides of fins profile, respectively. These corrugated fin patterns are oriented in tube arrangement along the direction of the flow. These fin patterns are introduced for augmentation of the fluid acceleration into the curvy regions of corrugated fin and to create a destabilized flow, modification of boundary layer and proper mixing of the bulk fluid. Created by the corrugated fin the destabilized flow not only reduces the thermal boundary layer thickness near the surface of the fin but also reduces the thermal resistance in airside. The design is introduced with two-corrugated fin patterns and four inline arrangement of oval shape of tubes One third of the regions employed in Fig 1 (a,b,c) are used as the main computational domain due to symmetry of the FTCHE models. The main geometrical parameters for performing the numerical investigation in two-corrugated fin design are shown in Table 1. More of the FTCHE geometry details could be found in the literatures [81-85] and the physical properties of air and aluminum are presented in Table 2. [86].

Three subdomains namely upstream side extended region, main region and downstream side extended region are consisted in the computational domain. The boundaries of the computational solid domains in upside and downside considered two neighbor corrugated fins, having an airflow channel with four inline oval holes in the main domain. The oval tubes' orientation is indicated from the entrance of flow and exterior sides of the fins in Fig. 1. To provide uniformity of the inlet velocity the boundary in the upstream region can be established at a distance of one tube diameter (D_c) in front of the leading edge of the fin [87, 88]. The real computational field was increased by $F_w /2$ ($F_w /2 > D_c$) in this study with small modification compared with D_c to maintain the inlet velocity to be consistent and uniform to a more extent. The exit domain was increased by F_1 to ensure a flow pattern to avoid recirculation of the flow and ensure for applying the completely developed boundary conditions at the outlet region according to Ref. [89]. The joins between fin flange and tube is assumed to be in perfect contact and the thermal resistance in the tube and flange is neglected by considering isothermal tube walls of outside diameter D_c. The corrugated fin thickness and the base fin thickness δ_f are same according to [77-80].

TABLE 1 COMPREHENSIVE GEOMETRIC PARAMETERS OF FTCHE [76]

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)	3.0
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 $D_c=D_{tube}+2\delta_f$	N	4

2.2. Mathematical Modeling and Governing Equations

Applying the conventional laws of mechanics to a Newtonian fluid gives the transport principal equations for a fluid. The direct solution of the transport governing equations is not possible. That is why the assumptions which are given below have been used in numerical modeling of fluid flow in the FTCHE according to some literature reviews [90-94].

2.2.1. ASSUMPTIONS

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

2.2.2. GEOMETRY OF CORRUGATED FINS

The range of Reynolds number was chosen according to the research done by [95-97]. The numerical simulations have been performed for Reynolds number 200, 400, 600 and 900 based on the hydraulic diameter and the mean velocity at the minimum flow cross-sectional. The one corrugated, two corrugated and three corrugated fin design have been presented in Fig 1 according to the research done by [76]. Top view and front view of two corrugation design with geometric characteristics is shown in Fig 2

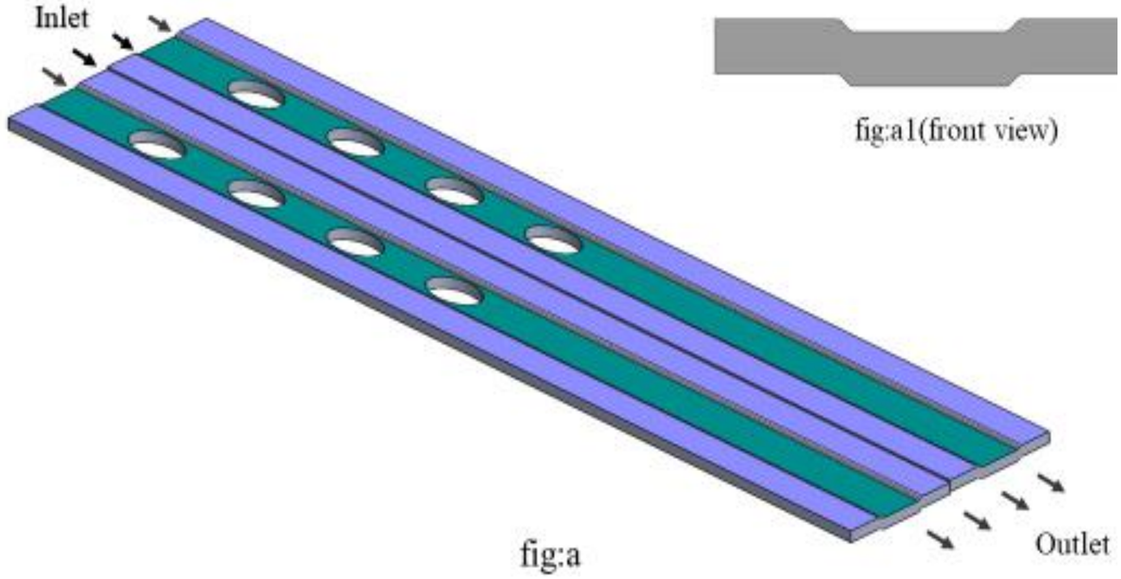


fig:a1(front view)

fig:a

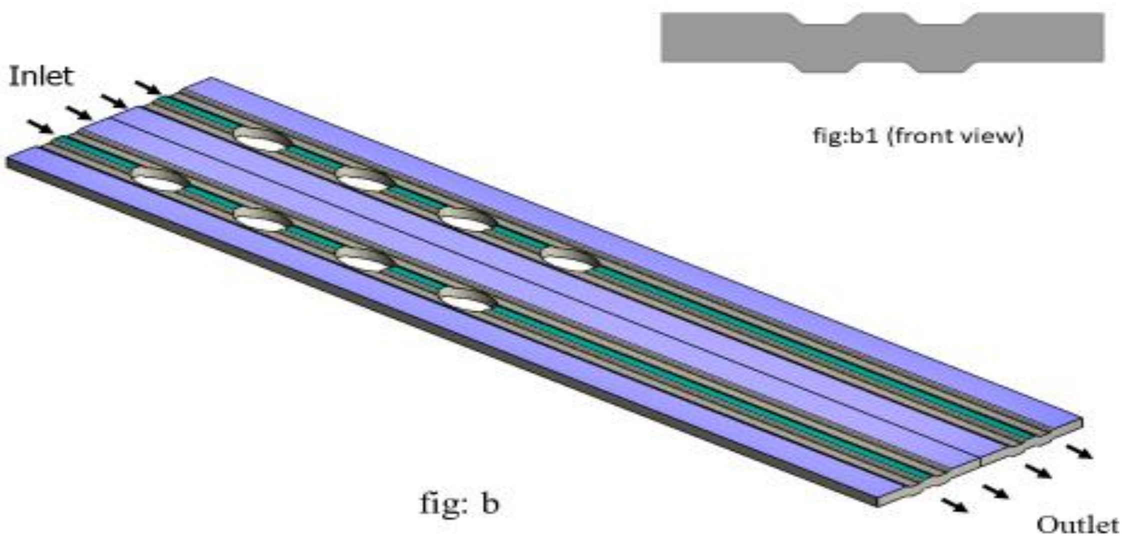


fig:b1 (front view)

fig: b

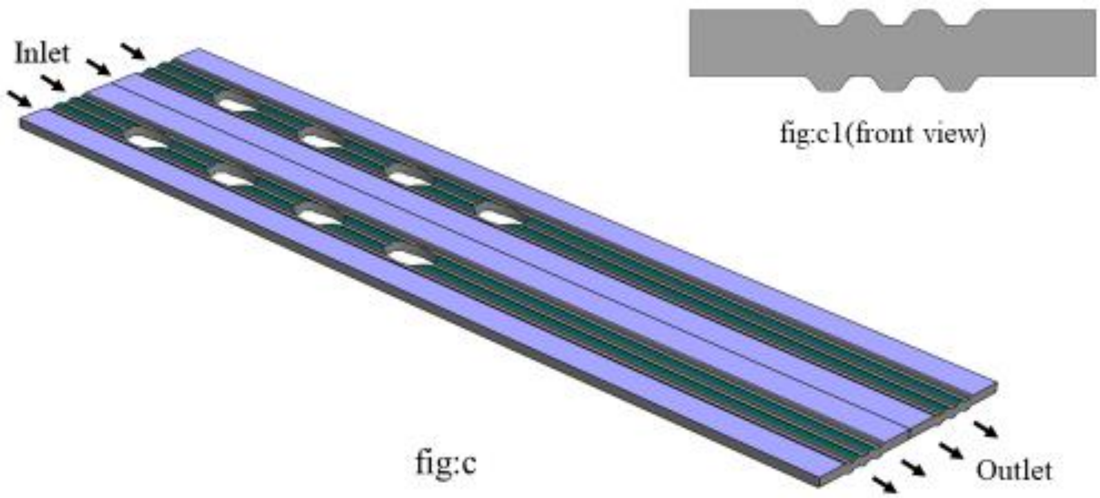
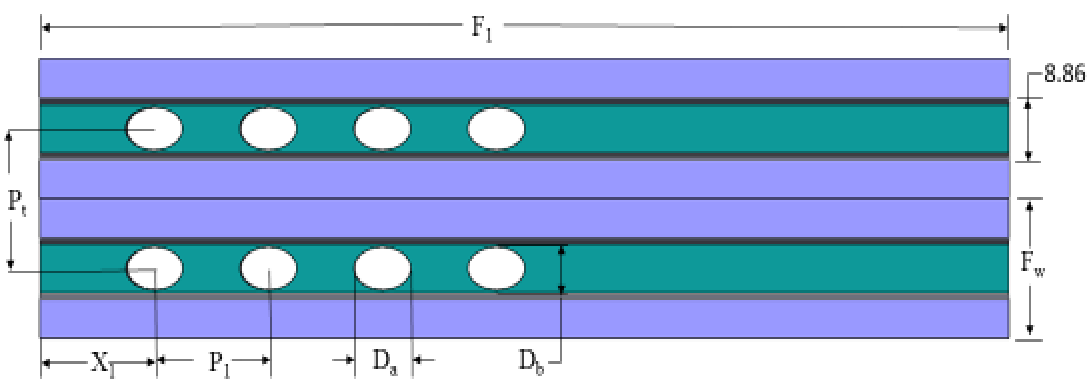
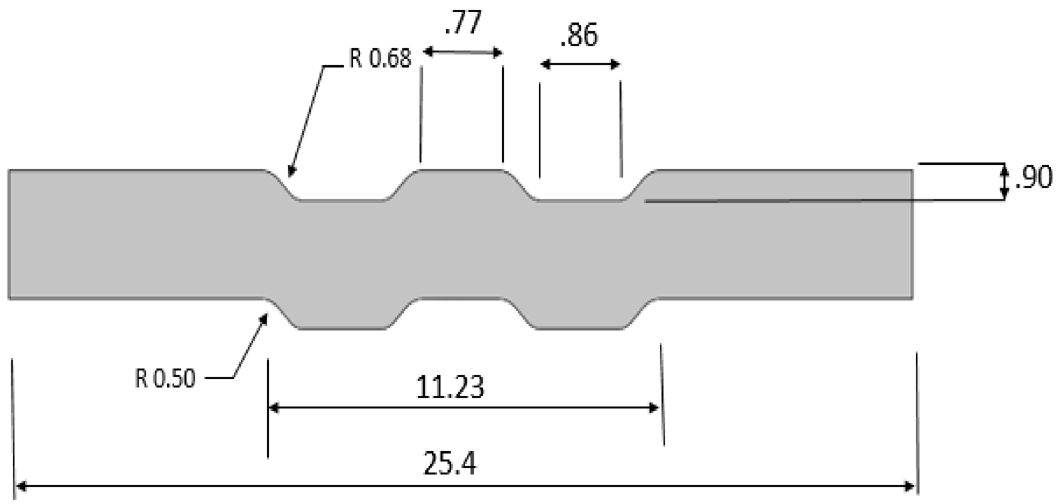


FIGURE 1: A)ONE CORRUGATED DESIGN B)TWO CORRUGATED DESIGN C)THREE CORRUGATED DESIGN



(a)



(b)

FIGURE 2 GEOMETRIC CHARACTERISTICS OF TWO CORRUGATED FIN PATTERNS: (A) TOP VIEW; (B) FRONT VIEW [76]

2.2.3. GEOMETRY OF VORTEX GENETORS (VGS)

With the same hydraulic diameter, the different shapes (square, star and circle) of vortex generator are found in the three corrugated fin shown in Fig 3. Numerical simulations have been performed of different Reynolds number in three corrugated fin with one and two-line vortex generators for different shapes, such as square, star and circle. Fig 4 describes three corrugated fin with one line and two line vortex generators of square shape.

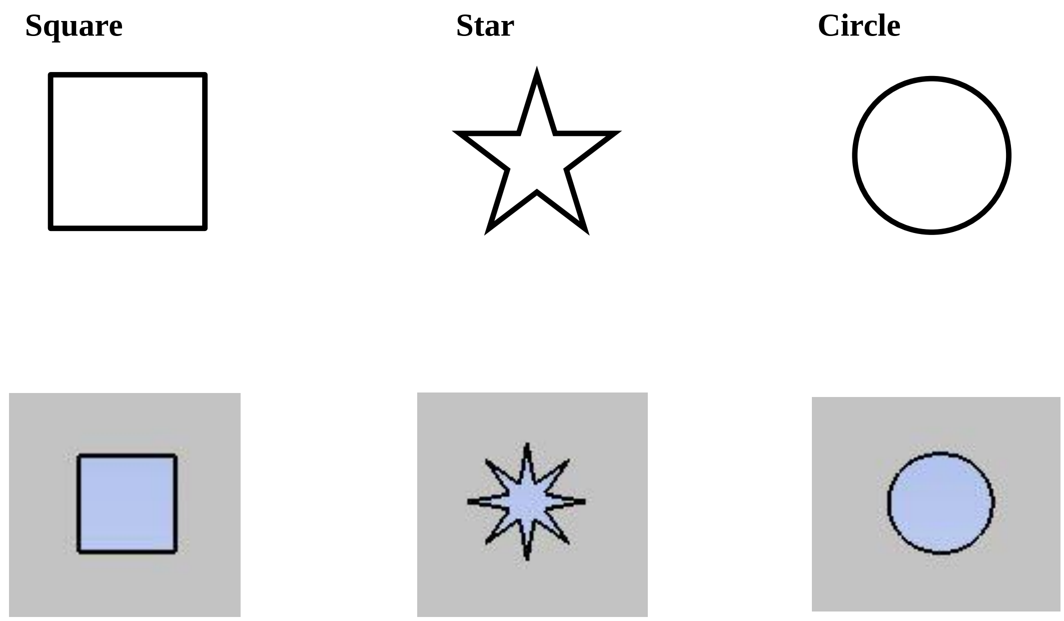


FIGURE 3: DIFFERENT SHAPES OF VORTEX GENERATOR

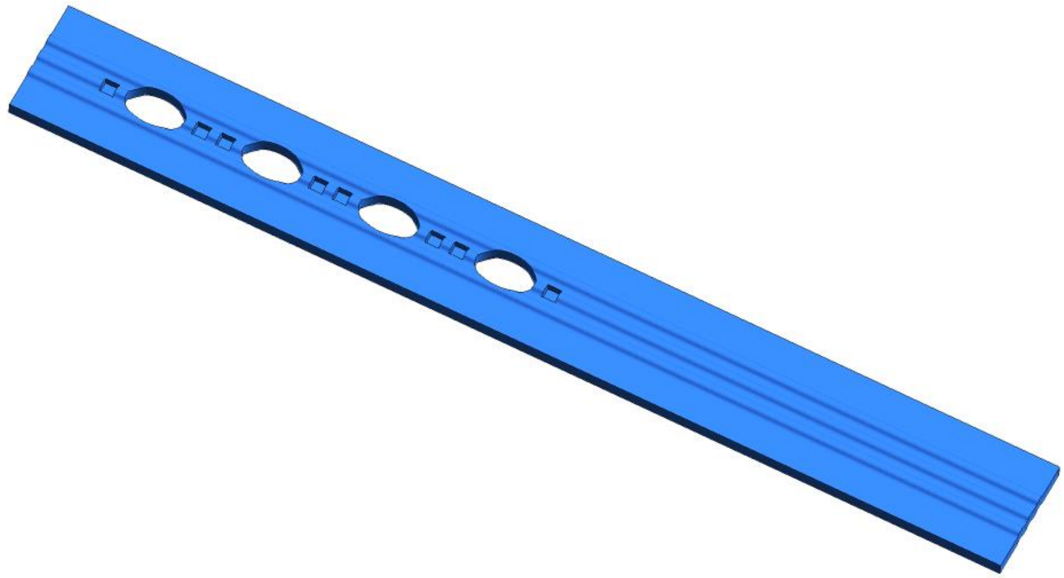


FIGURE 4: ONE LINE SQUARE SHAPED VORTEX GENERATOR IN THREE CORRUGATED FIN

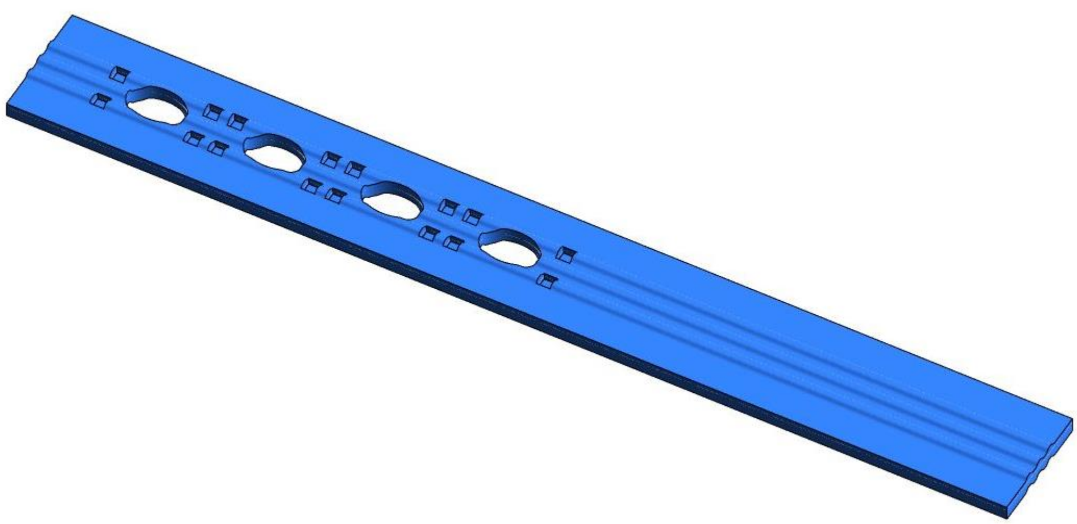


FIGURE 5 :TWO LINE SQUARE SHAPED VORTEX GENERATOR IN THREE CORRUGATED FIN

2.3. Boundary Conditions

The boundary conditions at all surfaces are needed to identify by the fluid dynamic transport equations which are in the form of elliptic partial differential equations used in computational domain. It can be found in some selected researches [57, 77, 78, 90, 92, 94, 98]. The air volume that passes through the gap between a pair of fins is extended downstream from the outlet of the last tube to gain better applications of boundary conditions which ensures a\the flow in the computational domain of the actual FTCHE. Numerical oscillations have been reduced by following this technique [99]

The boundary conditions are presented in table 2.

TABLE 2: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	For the outlet region ,Neuman boundary condition (one way)
$U=U_m=constant$ $U_x=U_y=0$ $T=T_{in}=350K$	$\frac{\partial U_x}{\partial y}=0$ $U_y=0$ $\frac{\partial U_z}{\partial y}=0$ $\frac{\partial T}{\partial y}=0$	In the upside and bottom side of boundaries $U_x=U_y=U_z=0$ $\frac{\partial P}{\partial n}=0$	$\frac{\partial U_x}{\partial z}=0$ $U_y=0$ $\frac{\partial U_z}{\partial z}=0$ $\frac{\partial T}{\partial z}=0$ $\frac{\partial P}{\partial z}=0$

$\frac{\partial U_x}{\partial y}=0$ $U_y=0$ $\frac{\partial U_z}{\partial y}=0$ $\frac{\partial T}{\partial y}=0$ $\frac{\partial P}{\partial y}=0$	$\frac{\partial U_y}{\partial x}=0$ $U_x=0$ $\frac{\partial U_z}{\partial x}=0$ $\frac{\partial T}{\partial x}=0$	$\frac{\partial U_y}{\partial x}=0$ $U_x=0$ $\frac{\partial U_z}{\partial x}=0$ $\frac{\partial T}{\partial x}=0$	
$\frac{\partial U_x}{\partial x}=0$ $U_x=0$ $\frac{\partial U_z}{\partial x}=0$ $\frac{\partial T}{\partial x}=0$ $\frac{\partial P}{\partial x}=0$	For the surface on the 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 $U_x=U_y=U_z=0$ $\frac{\partial T}{\partial x}=0$	

2.4. PARAMETER DEFINITIONS

In this study, the flow condition was characterized by Reynolds number, bulk temperature, heat transfer rate and overall heat transfer coefficient. The Colburn factor was used to assess the performance, Nusselt number and Friction factor are used to determine the thermal-hydraulic characteristics of FTCHE. These parameters are defined as follows:

Reynolds number is a dimensionless quantity and it helps to predict the flow patterns in different situations .It measures the overall significance of two sorts of forces for given stream conditions, and is a manual for when fierce stream will happen in a specific circumstance[100].

$$\mathbf{Re}= \rho U_m D_h/\mu$$

The *j* factor or Colburn *j* factor or the Colburn-Chilton *j* factor is a dimensionless factor for heat transfer that was first proposed by Prof. Colburn. [101] It has been broadly utilized for calculating the heat transfer coefficient in the design & performance prediction of heat exchangers, particularly compact heat exchangers.

$$j= hPr^{2/3}/\rho U_m C_p$$

Fanning friction factor is a dimensionless number and ratio between the local shear stress and the local flow kinetic energy density. It is one-fourth of the Darcy friction factor and is an component in the calculation of pressure loss due to friction in a pipe. It is a function of the roughness of the pipe and the level of turbulence within the liquid flow.

$$f=2\Delta P A_c/\rho U_m^2 A_o$$

Heat transfer performance denotes the increase of heat transfer in a definite a body , channel or heat exchanger .It is mainly function of colburn factor and Reynolds number .

$$F=f\,(Re_{Dh.})^3$$

Pressure drop mainly characterized as the difference in total pressure between two points of a fluid carrying network. It happens when frictional forces, evolved by the resistance in flow, act on a fluid as it flows through the tube.

$$\Delta P=P_{inlet}-P_{outlet}$$

Logarithmic mean temperature difference (LMTD) is used to determine the temperature driving force for heat transfer in flow systems, most notably in heat exchangers. For heat exchanger with constant area and heat transfer coefficient, the larger the LMTD, the more heat is transferred .It is mainly used for the analysis of a heat exchanger with constant flow rate and fluid thermal properties..

$$\Delta T_m=[(T_{wall}-T_{inlet})-(T_{wall}-T_{outlet})]/\ln[(T_{wall}-T_{inlet})/(T_{wall}-T_{outlet})]$$

Heat transfer, also simply referred to as heat, is the movement of different temperature of thermal energy from one object to another.There are three different ways in which energy can be transferred: conduction (by direct contact), convection (by fluid movement) and radiation (by electromagnetic waves):

$$Q=\acute{m}C_p(T_{outlet}-T_{inlet})$$

The transfer rate of heat between a solid surface and a liquid per unit area per unit difference in temperature is convective heat transfer coefficient. The coefficient of convective heat transfer depends on the physical properties of the liquid and the physiological condition.

$$h=Q/A_o\Delta T_m$$

The Nusselt number is the ratio of convective to conductive heat transfer across a boundary. A dimensionless parameter used in calculations of heat transfer between a moving fluid and a solid body.Unity[102] value represents the pure conduction and value between 1-10 denotes laminar flow and value with 100-1000 represent active convective heat transfer and turbulent flow [103]

$$Nu=hD_h/k$$

3. Methodology

3.1. Numerical Approach

Numerical investigations for one, two and three corrugation fin design and oval tube arrays have been performed by CFD simulation software, Ansys 12.0. Initially discretization of transport equations in the computational domain has been executed by using finite volume method. The purpose of discretization is to simplify the governing partial differential equations in order to bring out the algebraic equations. The upwind and hybrid schemes are used to discretize. The partial differential equations are set upon every control volume which ensures conservation of relevant quantity like energy, mass, momentum for each control volume in a discrete way. In this case, local errors are removed swiftly and convergence is accelerated by using algebraic multi-grid scheme. The hydrodynamic equations for all three directions are solved by COUPLED scheme. This is a solution algorithm and this code is used because it takes less time to solve and the solution ensures reliability, it removes associated approximate conditions, eliminates nonlinearities in the physical domain and stretched and skewed meshes. The continuity and momentum equations are also solved in coupled method by pressure-based COUPLED solver. Dependent variables are solution vector in a single matrix equation which contain unknown velocities and pressure are also solved then [104]. The solution was converged when the residuals were less than 1×10^{-4} and for continuity and 1×10^{-8} for energy equation. For all these conditions, flow variable values became stable irrespective to any number of iterations.

3.2. Grid Generation and Grid Independency

Grid generation or meshing is the basis of many computational methods that solve complex partial differential equations in case research of fluids. If additional orthogonal grids can be generated, the number and complexity of the calculations will be reduced and the solutions will therefore be accurate and quicker. A appropriate grid generating number for the physical domain is the first phase which governs the fluid flow. The three primary variables favoring the grid generation technique are the simplicity of mesh, resolution promptness, convergence and precision of the outcomes acquired in the calculation. ANSYS Workbench was employed to fix the governing equations. ANSYS meshing produced computational domain meshes. For distinct grid numbers the validation of the grid independence for the numerical solution is done. To validate grid independence solutions, the selection and evaluation for one-corrugated fin designs at ReDh 1⁄4900 were performed at four distinct grid point numbers, including exactly 147264, 198967, 272745 and 281310 components. The average number of Nusselt and the Fanning frictional factor for the 2011-1831 million mesh elements are lower than 0.14 and 1.90 percent, as shown in Table , of the million mesh elements. As a consequence, the third amount of mesh components taken in the computer domain is around 2.0 million in order to ensure a balanced balance between computational time and solution precision. For other instances, similar validations are also carried out. Figure 6 shows grid generation information for the FTCHE model with three corrugated fins and oval pipes. Details of meshing for all subdomains can be shown in the top and front opinions of grid allocation. The computer domains are divided into three subdomains consisting of extended entry, main, and exit domains with very more structured grids with good quality and maximum element shapes of hexahedra. Table 6 with statement of element form lists further details of element configurations generated by the ANSYS Meshing systems of the ANSYS Workbench platform. The grid production process is designed so that hexahedron meshes can fill a given volume more effectively than other mesh shapes. The results are more accurate and decrease the computing time, two key features of this type of grid generation. 3.4. Verification and numerical simulation Validation of numerical results mainly performed as two key parameters for the assessment of thermo-hydraulic characteristics of FTCHEs on the heat transfer performance and friction factor in this study. According to the experimental conditions and test model described in Ref.[70], validation of model is organized. This evidence describes the use of numerical modeling to simulate FTCHE's airflow and heat transfer as a CFD approach.

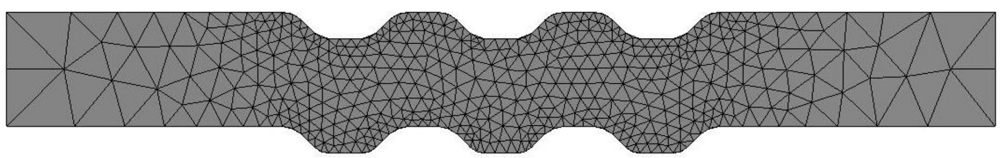
	Number of Mesh elements			
	147264	198967	272745	281310
Nu	7.12	6.8889	6.7181	6.7090
f	0.02064	0.02094	0.02207	0.02165

TABLE 3 :GRID INDEPENDENCY TEST

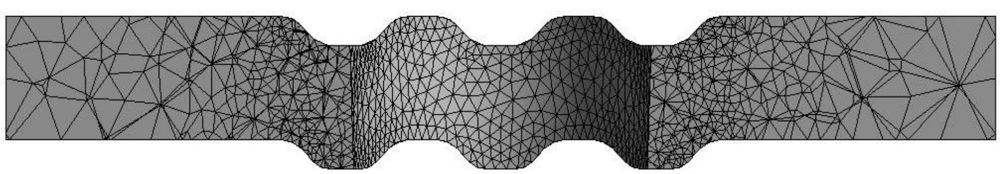
	One corrugation	Two corrugation	Three corrugation
Number of nodes	292771	289591	281310
Number of elements	1523022	1488711	1450506
Tetrahedral size	1523022	1488711	1450506

TABLE 4:MESH CHARACTERISTICS OF DIFFERENT GEOMETRIES OF FTCHE

a)Isometric view of mesh generation in three corrugated fin

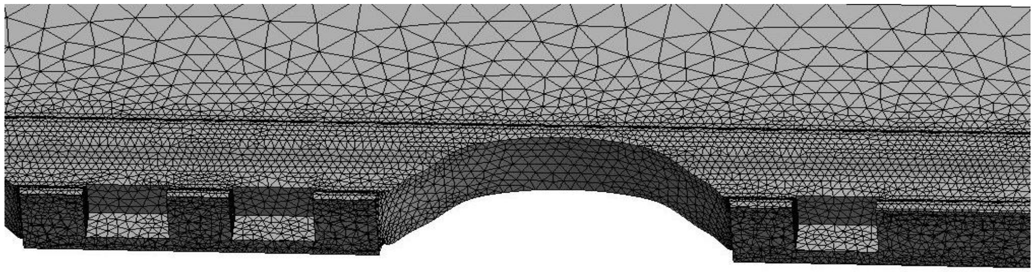


b) Front view of mesh grid from inlet section in three corrugated fin

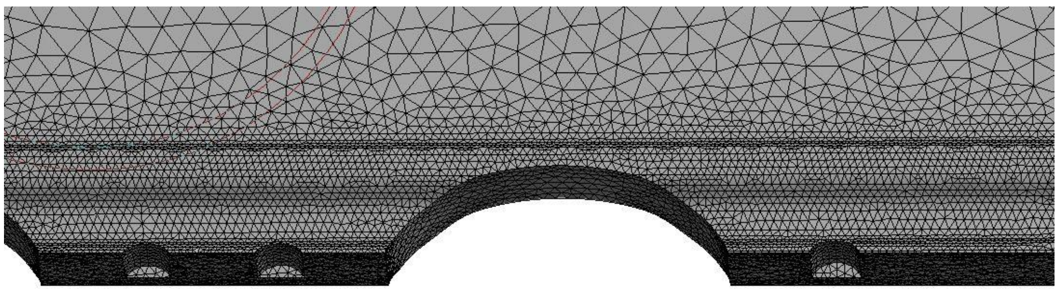


c) Cut section from middle region and mesh grid of oval tube

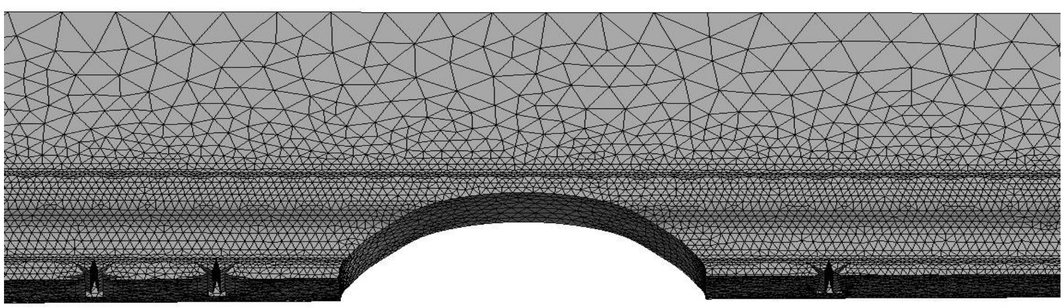
FIGURE 6 MESH GRID GENERATION OF THREE CORRUGATED FTCHE



a) Cut section from the mid region and mesh grid of square shaped vortex



b) Cut section from the mid region and mesh grid of circular shaped vortex



c) Cut section from the mid region and mesh grid of star shaped vortex

FIGURE 7:MESH GRID GENERATION OF DIFFERENT VORTEX SHAPES (SQUARE ,CIRCLE ,STAR)

TABLE 5 MESH CHARACTERISTICS OF DIFFERENT GEOMETRIES OF FTCHE

	Square		Star		Circle	
	2mm	3mm	2mm	3mm	2mm	3mm
Number of nodes	260800	278436	264420	267103	263874	262486
Number of elements	1429263	1429263	1345328	1357273	1345171	1336825
Tetrahedral size	1429263	1429263	1345328	1357273	1345171	1336825
Max. face angle	127.623°	127.245°	127.82°	129.583°	127.781	125.503
Min face angle	55.254°	54.37°	56.13°	129.005°	56.0688	56.1059

4. VALIDATION

The numerical results were mainly validated in this analysis as a two important parameter for evaluating the FTCHEs ' thermal-hydraulic efficiencies in terms of Colburn factor and friction factor. The validation of the model is arranged according to the test conditions and the model described in experimental analysis by Wang et al [23]

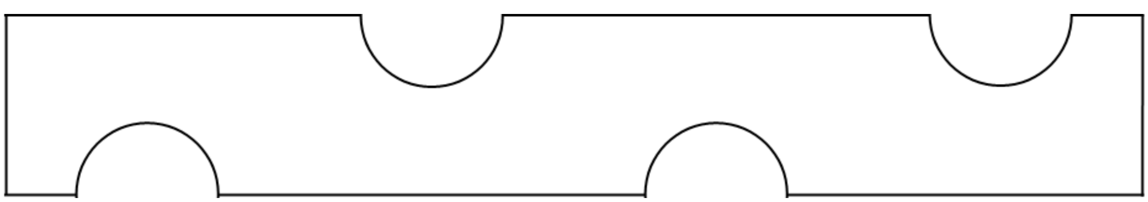
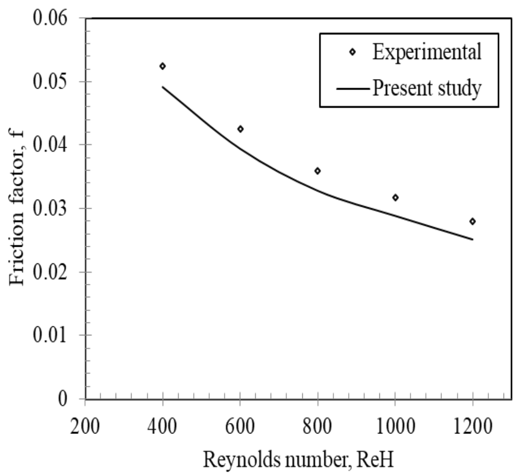
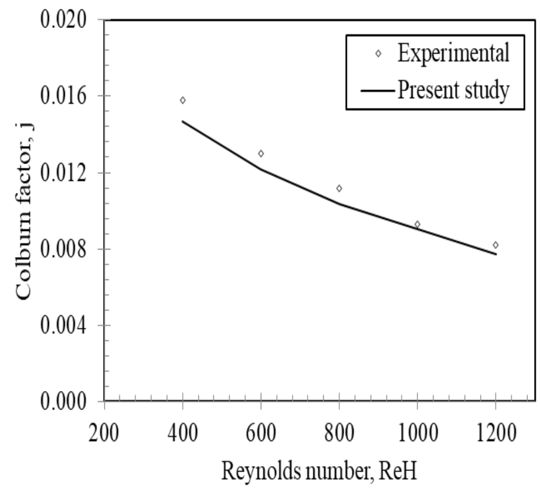


FIGURE 8 :COMPUTATIONAL DOMAIN OF STAGGERED CONFIGURATION FIN CONFIGURATION.

The present Validation study for code validation purpose are same and is given below:

Ll (mm) = 22, Lt(mm) = 25.4 Fp (mm) = 3.00, Ft(mm) = 0.13





The favorable consistency of numerical simulation between the numerical results and experimental data is revealed in this study and the numerical model therefore reliable for the prediction of thermal transfer characteristics and flow structures in compact heat exchangers. The largest discrepancy between the numerical outcomes obtained by the current modeling and the experimental data for the Colburn j factor and friction factor were about 10.33% and 9.81% respectively for the laminar flow range.

5. Result and Discussion

The enhancement of heat transfer is associated with the heat transfer coefficients and pressure losses generated by the device. The enhancement of heat transfer is the process of increasing the thermal capacity of heat exchangers. The overall performance of heat exchanger can be improved by increasing the heat transfer coefficient and reducing the thermal resistance of the heat exchangers. In order to increase the heat transfer, mixing of hot and cold fluids and poor heat transfer in the wake region should be improved. The corrugated fins reduce the thickness of the boundary layer near the fin surface and increase the convective heat transfer coefficient.

5.1. PRESSURE FIELD

The variation of pressure distribution along the flow direction is displayed in fig. 9,10,11 shows the pressure contours in three configurations of FTCHE with different Reynolds number. From the pressure contours, the effect of corrugated fins on the dynamic flow characteristics along the physical domain is clearly displayed. As the fluid flows across the FTCHE, the pressure varies along the physical domains.

At the entrance region higher values of pressure distributions are observed. In the entrance region, with the increase of Reynolds number pressure distribution is increasing. As the fluid flow across the FTCHE, the pressure drop is increasing along the tube. Generally, flow across the FTCHE experience drag force which include pressure drag and the frictional drag.

In the figure, four tubes between the fin surfaces experiences these drag force. Pressure drag is associated with the pressure difference between the upstream and downstream direction of flow which is caused by the periodic separation of flow over surface. The eddy motion in the fluid resulted by the body passage caused pressure drag. Frictional drag is caused by the friction of a fluid against the surface of an object that is moving through it.

The drag is linked to the wake area formation which is easily seen behind the pipe of the FTCHE. Additionally, pressure at the weak region is gradually low compared to the other section of FTCHE. Adverse pressure gradient between the entrance and exit region of the FCTHE is led to flow separation.

The separation of the stream affects the area of the wake and the pressure losses due to formation of the eddy. This causes the fliud surface's pressure drag.

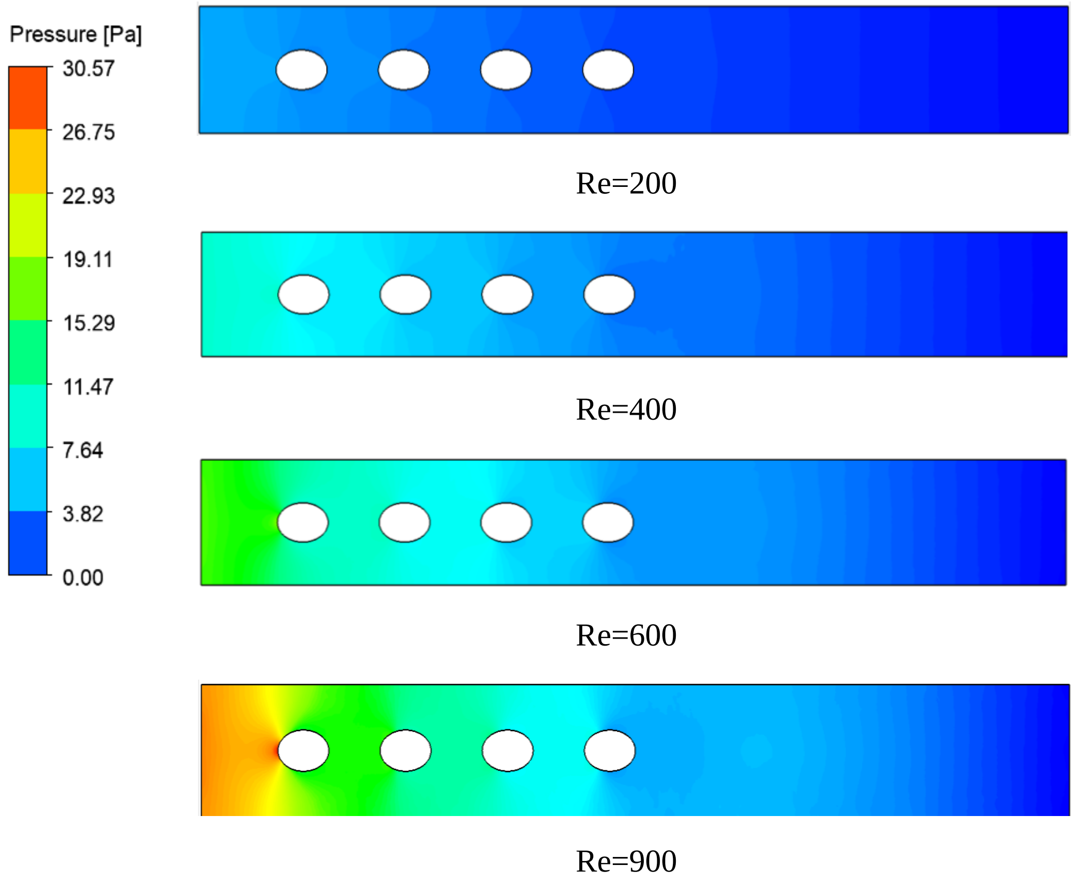


FIGURE 9 :PRESSURE PROFILE FOR ONE CORRUGATED FIN

From the contour figure it is visible that with the increase of Reynolds number the increase of pressure occurs and also pressure drop increase respectively. Along with the increase of pressure friction factor decreases exponentially. For two corrugated fin the increase of pressure with respect to base case is comparatively higher For three corrugated fin the change of pressure is highest

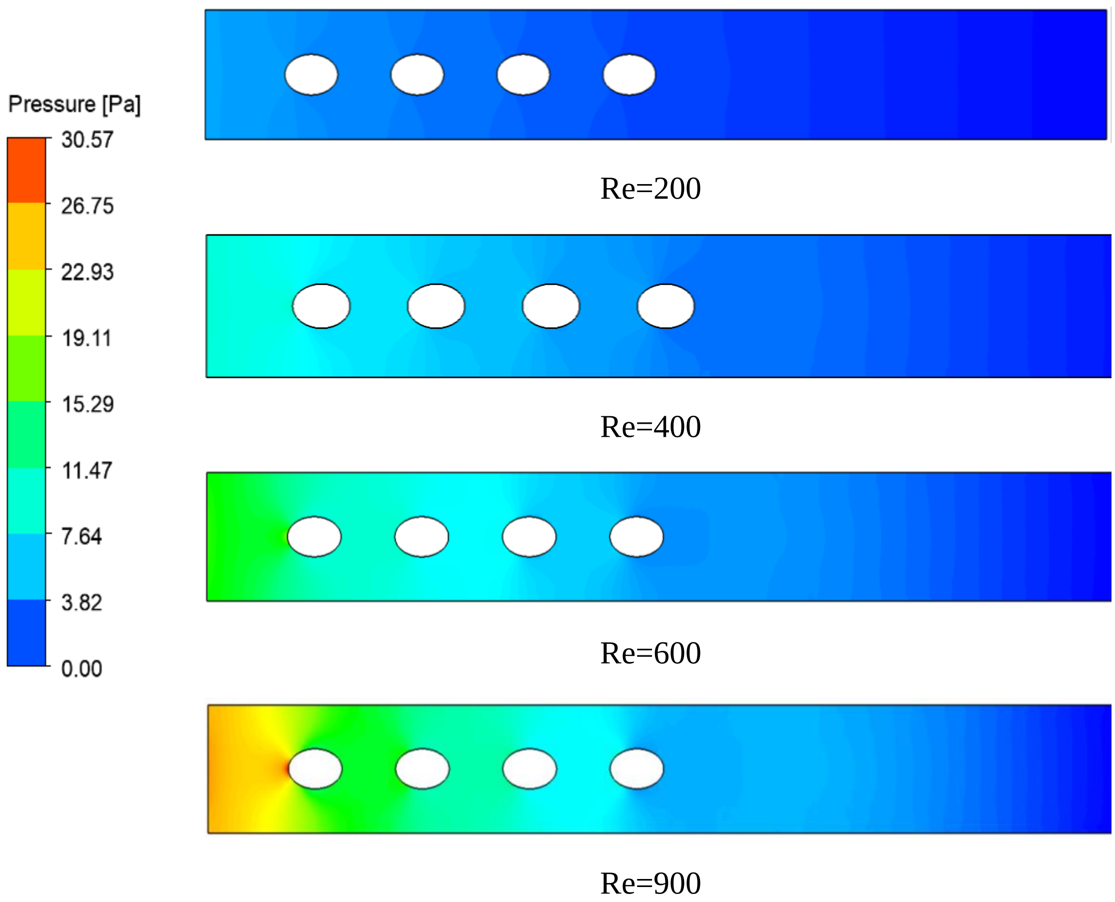


FIGURE 10PRESSURE PROFILE FOR TWO CORRUGATED FIN

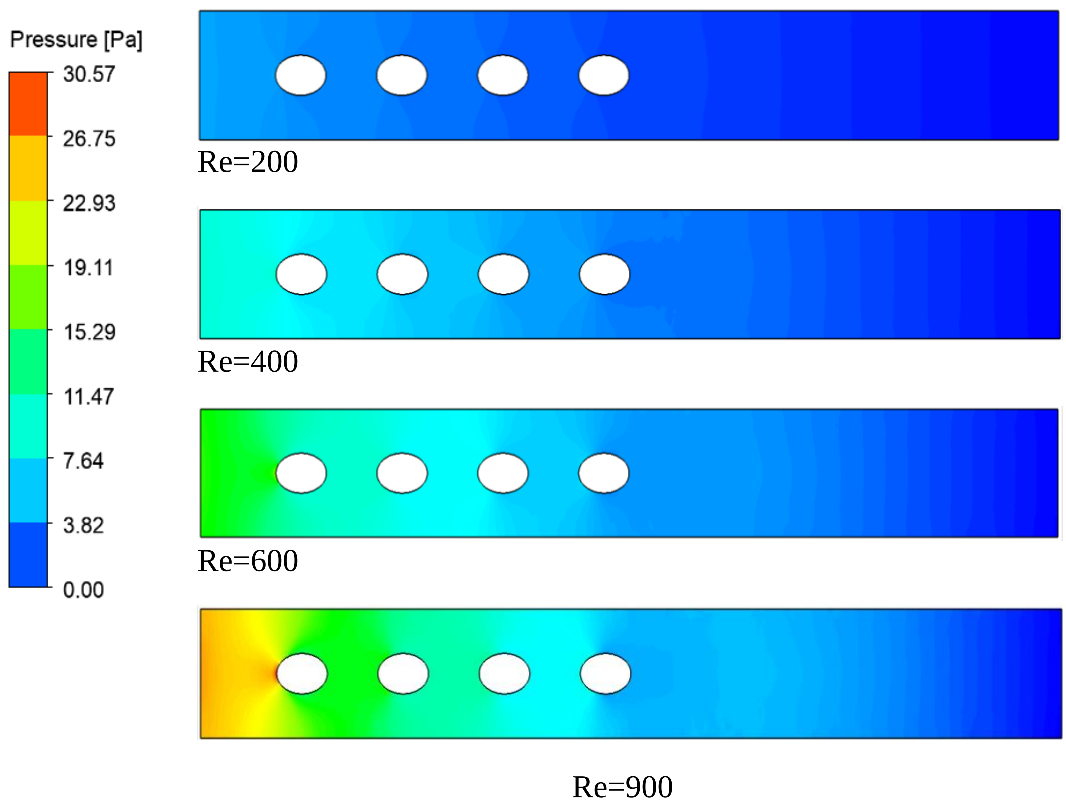


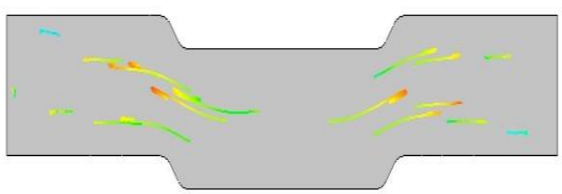
FIGURE 11 :PRESSURE PROFILE FOR THREE CORRUGATED FIN

5.2. FLOW PATTERN AND STREAMLINE

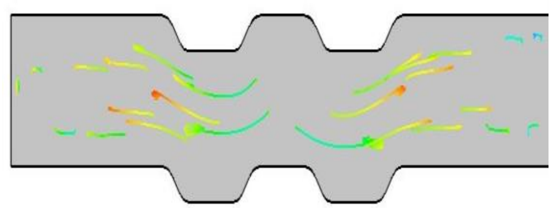
Predominantly flow behavior will be focused over physical domains for the presentation of the flow pattern study and the main focus of the flow behavior will be the wake region behind the tubes belonging to the Reynolds number 900. The effect of conventional form of the FTCHE on the flow pattern of oval tubes is showed in wake region of fig 12 . . In comparison with Fig. 12, the decrease in wake region in the rear of tube is clearly shown in Fig. 12.

The flow pattern in this model significantly enhanced the fluid flow mixing method from top to bottom of the pipes and the cold fluid mixing of the wake areas. From the streamlined it is found that three corrugations have less wake region than both one and two corrugation

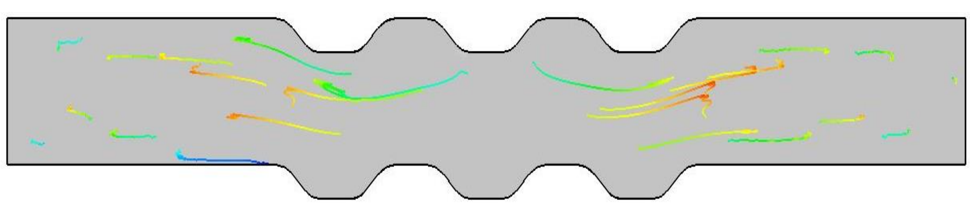
Now for a fixed Reynolds number three corrugation FTCHE exhibits lower resistance than FTCHEs with two corrugations. Two corrugation has a lower resistance than one corrugation FTCHE. the wake regions shows the effect of using one ,two ,three corrugations in FTCHE design .When the size of the wake decreases ,it indicates that the separation of the boundary layer on the bluff bodies occurs further along surface than before . after the comparison of the geometries present in this study ,it is found that the clear change in flow structure behind the tube and also the significant decrease in the wake region of three corrugation in fig 12 is clearly visible .corrugated design in FTCHE is a very effective process to increase the mixing proportions between upstream flow and the flow that stays in the wake region .The results shows that the most effective is three corrugated fin and the least effective is one corrugated fin .another reason is because of more momentum near the tube surface three corrugation is most effective .



a) One corrugated fin



b) two corrugated fin



c)Three corrugated fin

FIGURE 12: FRONT VIEW OF DIFFERENT CORRUGATED FINS WITH VELOCITY STREAMLINE .

5.3. TEMPERATURE DISTRIBUTION:

The temperature contours of fluid flow for one selected surfaces in the physical domains at h=.72 are demonstrated in fig 13,14,15. The results show the temperature contours at Reynolds number starting from 200 to 900 . The effects of corrugated fin and oval tubes is clearly shown on the temperature distribution in cross flow between of the one ,two and three-corrugated fin surfaces. The variation of temperature contours in these two FTCHEs can be explained with improving vortex core region and decreasing of the wake region involved with more fluid mixing process in the back of the tubes .The temperature distribution increases with the change of corrugated designs and it is highest in three corrugated fin .

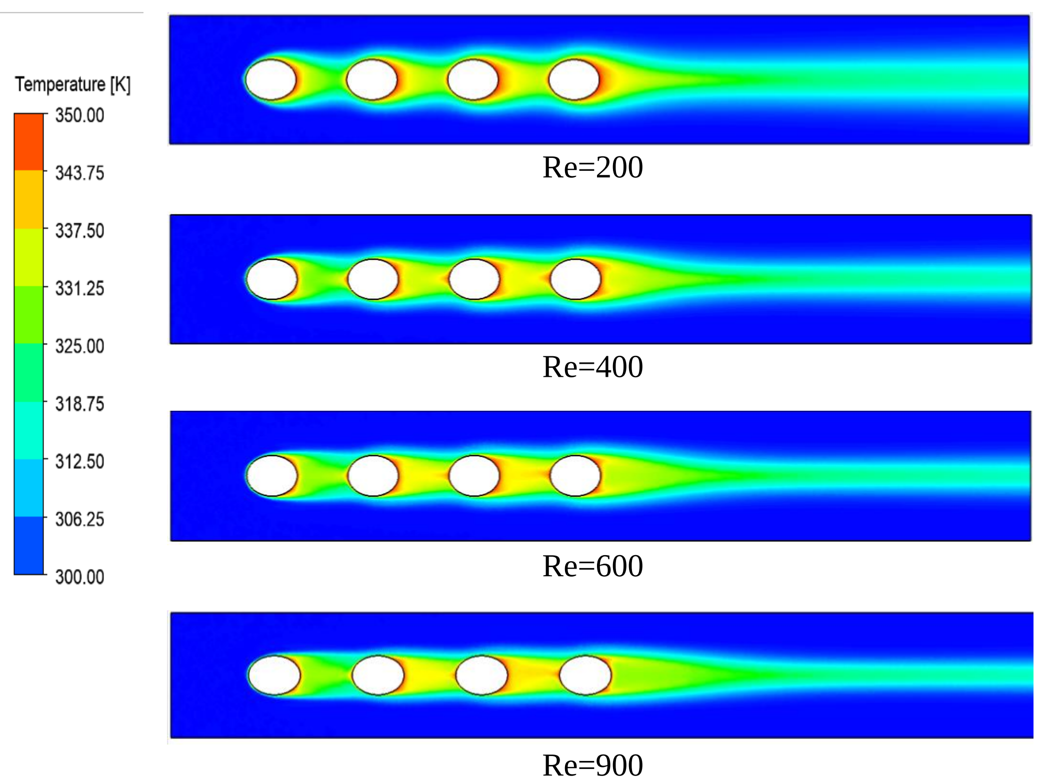


FIGURE 13TEMPERATURE PROFILE FOR ONE CORRUGATION

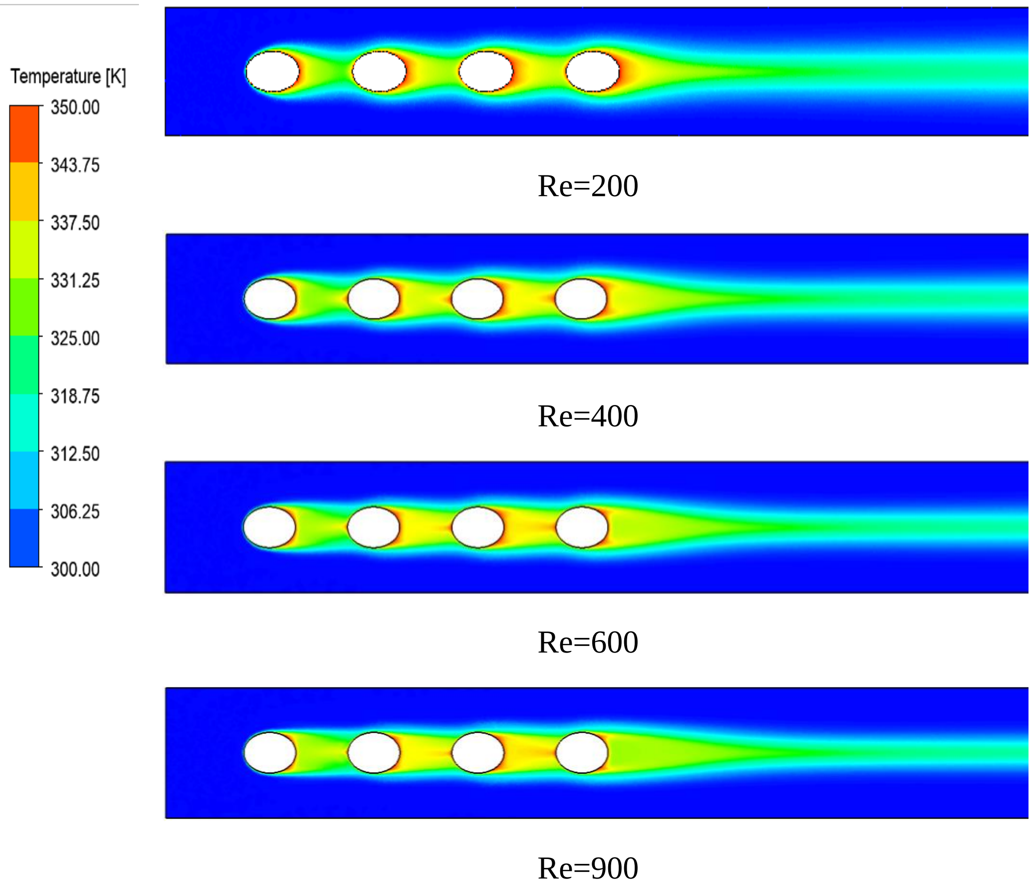


FIGURE 14TEMPERATURE PROFILE FOR TWO CORRUGATION

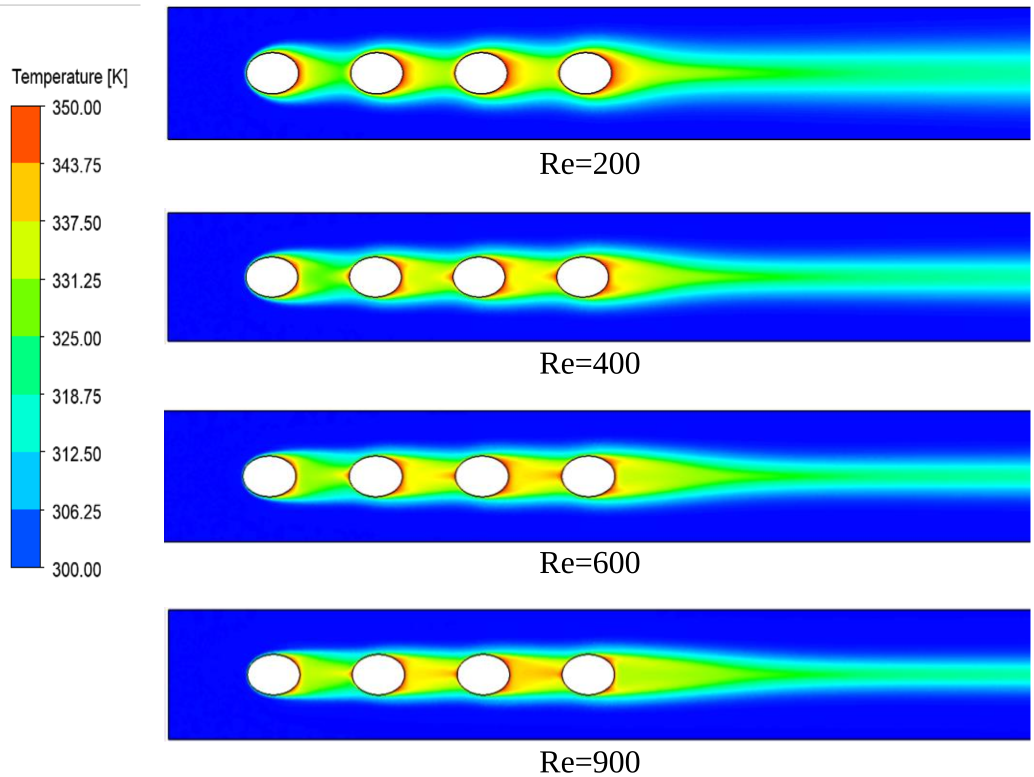


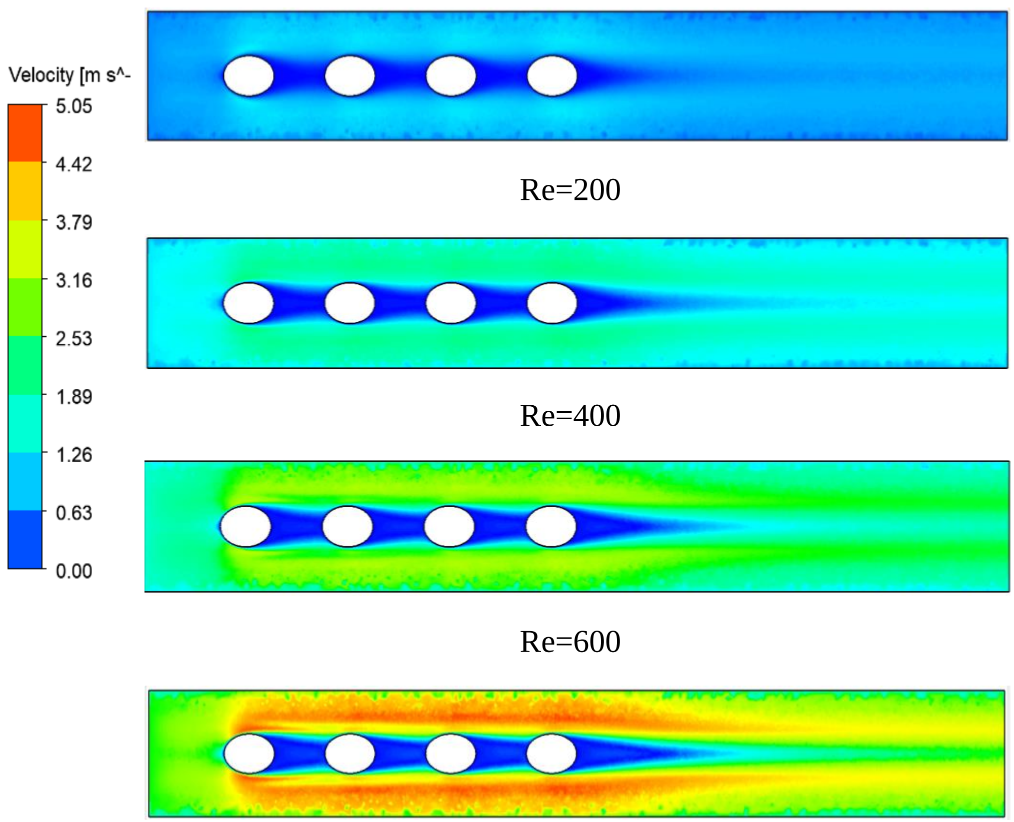
FIGURE 15 TEMPERATURE PROFILE FOR THREE CORRUGATION

5.4. Velocity field:

Fig 16,17,18 demonstrates the physical domain velocity contour of one,, two and three corrugated fins on the plane. It was discovered that, with the shift in the amount of Reynolds number, the impacts of corrugated fins and pipes influenced the velocity contour of the fin field and the heat exchanger pipe. Despite of geometries and structures differences, application of the corrugated fins with different Reynolds number may, in particular on tuber surfaces, modify flow patterns through the FTCHE and accelerate flow speed rate towards the wake zone. The inlet velocity of FTCHE increases with the increase of Reynolds number. It was observed from the fig that the velocity range improves with the amount of Reynolds number. At the smaller Reynolds number there is a lack of mixing method in the wake region which leads to a reduced convective heat transfer. Higher

Reynolds number improves the fluid mixing method and accelerates the flow rate across the tuber surface. From the fig it was observed that corrugated fins had great impact on the velocity contour. It delays the segregation of the flow, improves the flow rate, reduces the pipe drag and drives the primary flow into the wake zone.

wake region and stream mixing are linked to the convective heat transfer fundamental mechanism. It has also been noted that by adding corrugated fins to the FTHE, it causes bulk fluid mixing, destabilizes the flow and alters the boundary layer thickness across the FTHE. Greater enhancement of the mixing method and alteration of the boundary layer shown in the figure showed the effect of three corrugated fin designs with a greater Reynolds number.



Re=900

FIGURE 16 :VELOCITY PROFILE FOR ONE CORRUGATION

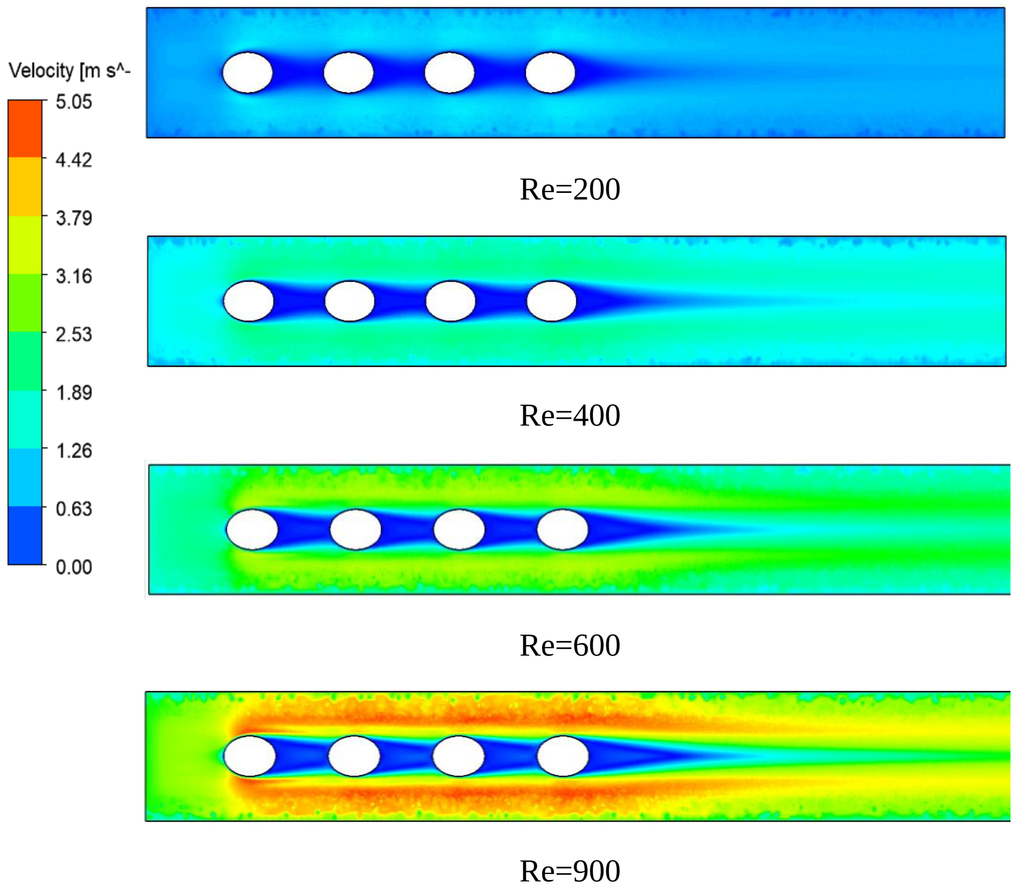


FIGURE 17VELOCITY PROFILE FOR TWO CORRUGATION

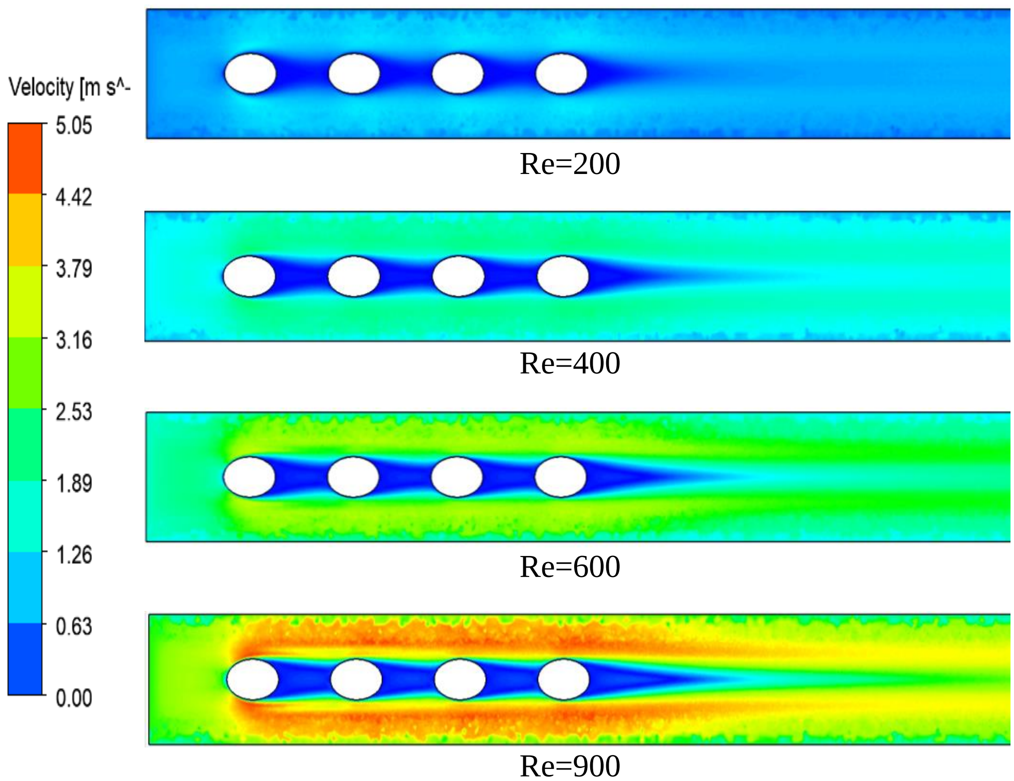


FIGURE 18: VELOCITY PROFILE FOR THREE CORRUGATION

6. GRAPH

6.1. HEAT TRANSFER CHARACTERISTICS

Pressure drop and friction factors are directly related to each other.With the increase of pressure drop friction factor decrease drastically . The range of Reynold number that are considered are from 200 to 900 . Base case considered as the FTCHE with one corrugation . The result shown in the that smaller values of friction factor in the physical domain is present in the base case that's one corrugation FTCHES compared to other design .In addition with the increase in Reynolds number the variation of drag coefficient decreases ,over a large range in Reynolds number it seems to be constant .

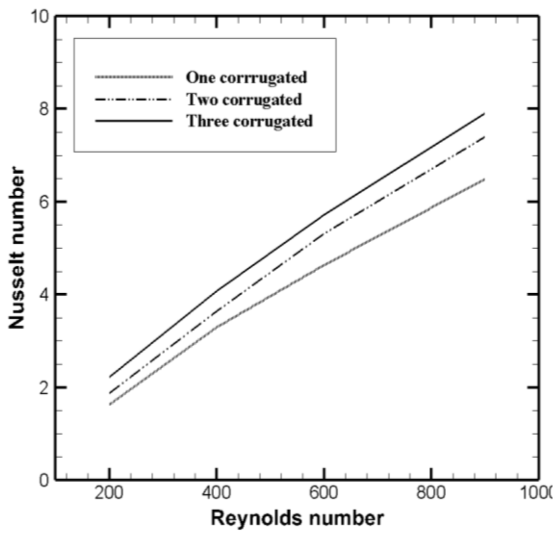
The average of nusselt number of various shapes of FTCHE is demonstrated in fig.this results agreed with the result displayed in the section with temperature contour . it is clearly visible that significant increase of nusselt number in three corrugated fins. It is also seen that in all three cases nusselt number increases with the increase of Reynolds number .If we study the graph properly we will find that the in Reynold number 200 from 1st corrugation increase in the nusselt number is 15.2% in second and in the three corrugation FTCHE the increase is 36.18% which is quite quite significant amount of improvement .If we look at Reynold number 400 the increase in nusselt number is 10.61% in two corrugation and 23.93% in three corrugation .In 600 reynold number the increase in nusselt number is 14.65% and 23.45% for three corrugation .In Reynold number number 900 the increase in the nusselt number is 14.21% and the increase in three corrugation is 21.847 % .

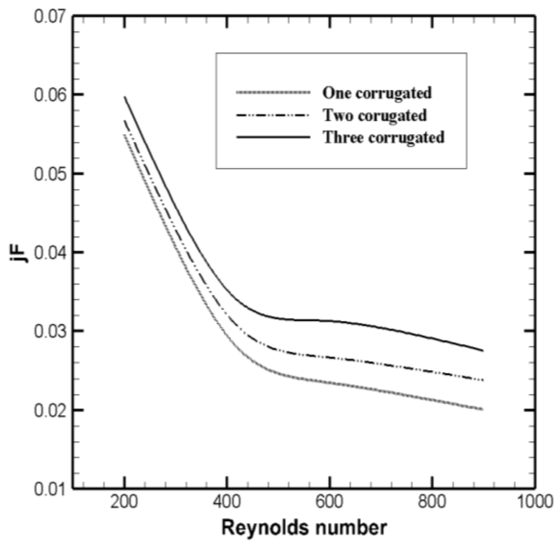
The j-colburn factor reffered to a non dimensional quantity and shows a relation between the geometry ,fluid properties ,heat transfer and flow conditions .For different Reynolds number the result of simulation is displayed in fig as the the variation of the j colburn factor as a criterion analogy between heat and momentum . The numerical findings show that the fin configurations influence thermal and hydrodynamic characteristics significantly.With more curvature areas in corrugated fines, the j factor is growing. In the graph we see the inverse exponential of J with respect of Reynold number.But in every

Reynold there is difference in J with respect of different corrugation design .In Reynolds number 200 the increase in the value of j is 10.9079% for two corrugation and 24.9783 % for three corrugation design .For Reynolds number 400 the increase in j in two corrugated design is 11.656 % and for three corrugated design is 23.7713% .For 600 the change is more higher with respect to lower Reynolds number that is 16.10% for the two corrugated fin and 26.68 % increase for three corrugated FTCHE .For 900 the increase with respect to baseline is highest .the increase in The JF factor represents an index to assess the general output which depicts better heat transfer efficiency in the pressure drop shown in the figure. 11.

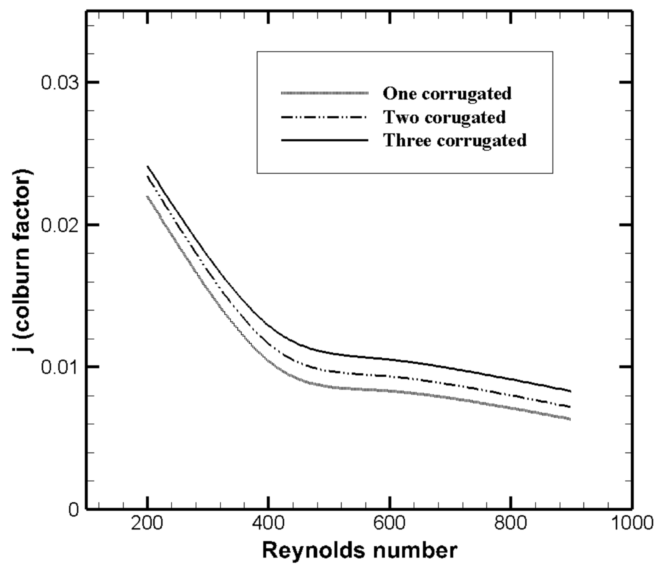
The graph shows enhanced performance of three corrugated fins compared with the other corrugated design of FTCHEs .The variation of JF factor versus Reynolds number for different case studies related to FTCHEs with the increase of Reynolds number the JF factor is decreasing and the trend of decreasing is more with the one corruted fin that means baseline .

It is found that three corrugated fin has a better overall thermal hydraulic performance and also better heat transfer than other corrugated fins .If we analyze the data closely with the base line that means with the one corrugated we found 3.26 -18.67 % increase in JF value with the increase of Reynolds number .and for three corrugation the increase is 8.82 -37.4%





(a) (b)



(c)

FIGURE 19(A) NUSSELT NUMBER VS REYNOLDS NUMBER.

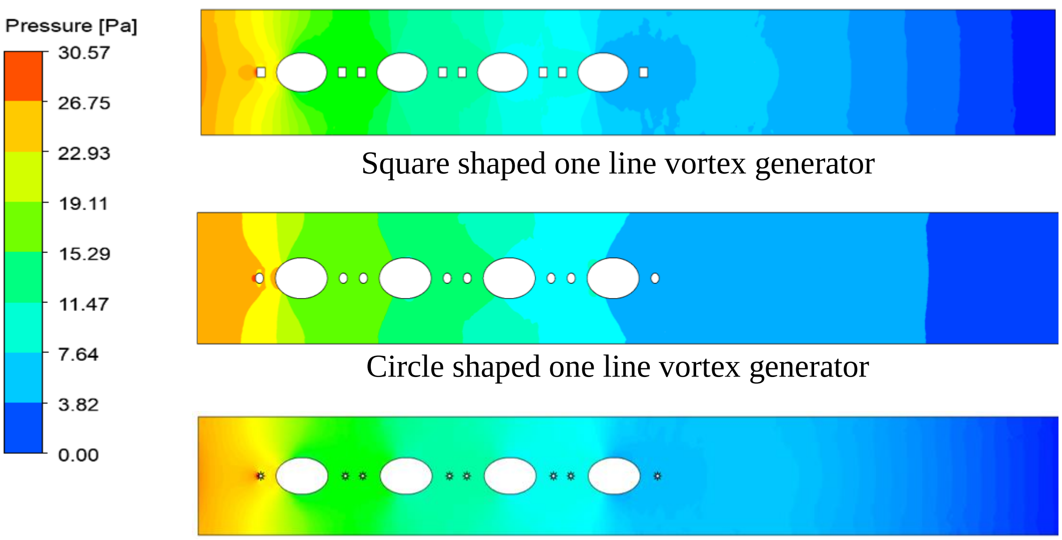
FIGURE 20 (B) OVERALL THERMAL HYDRAULIC PERFORMANCE VS REYNOLD'S NUMBER

FIGURE 21 (C)J-COLBURN FACTOR VS REYNOLDS NUMBER.

7. Vortex generator

7.1. Pressure field

Figure 22 illustrates the pressure distribution and variability throughout the FTCHE reference case and the case with a vortex generator behind the tubes at various geometries and arrangements. The effect of applying various geometries and configurations of vortex generators across the FTCHE channel behind the tubes is clearly shown from the pressure contours. Higher pressure distribution is typically observed in all cases at FTCHE's entry area. The pressure varies and changes for each case as the fluid flows across the FTCHE. Of Reynolds numbers from 200 to 900, different structure of the stress field is also obtained. The pressure distribution one line and two line with different shapes are demonstrated below. With the increase of vortex size pressure drops decrease slightly .



Square shaped one line vortex generator

Circle shaped one line vortex generator

Star shaped one line vortex generator

FIGURE 22:PRESSURE DISTRIBUTION OF ONE LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE AT RE 900

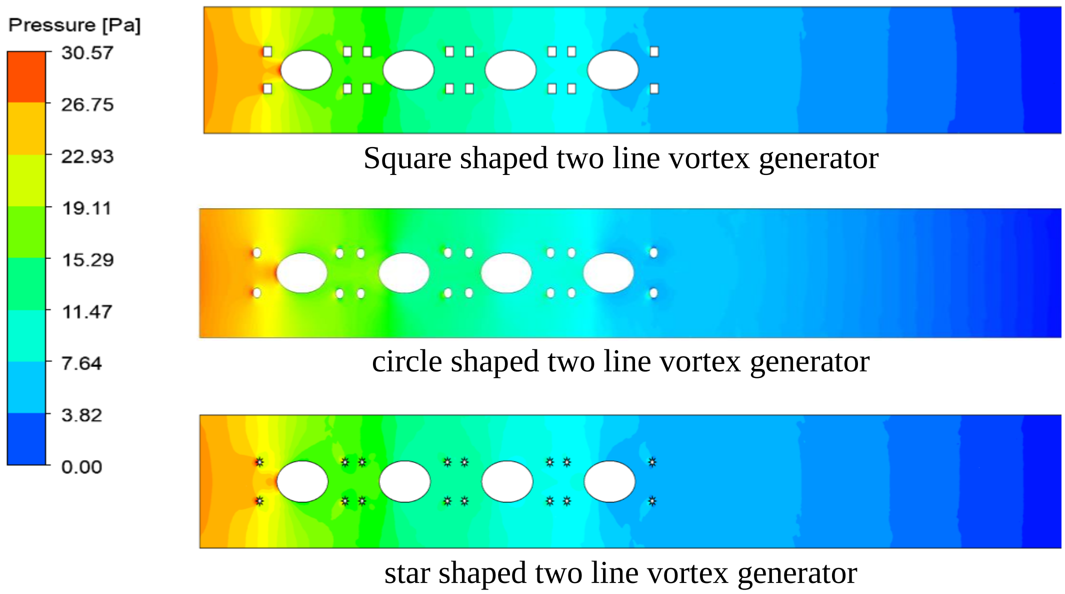


FIGURE 23: PRESSURE DISTRIBUTION OF TWO LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

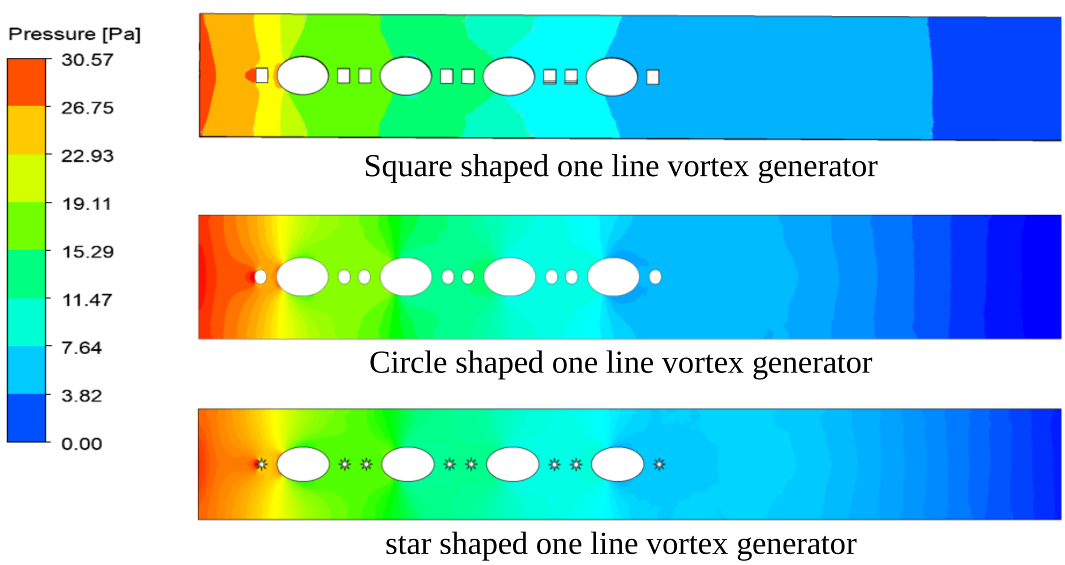


FIGURE 24: PRESSURE DISTRIBUTION OF ONE LINE 3 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

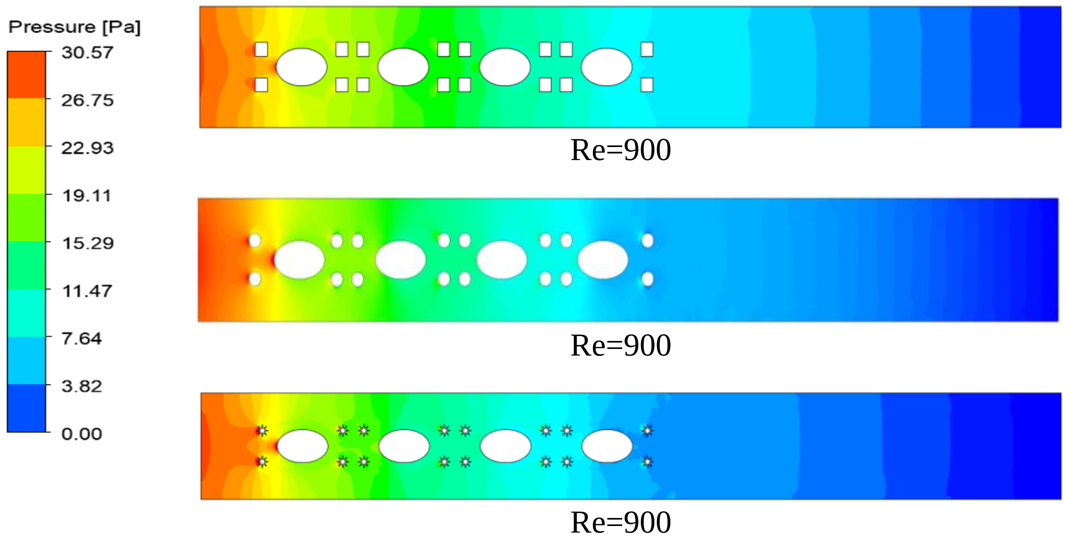
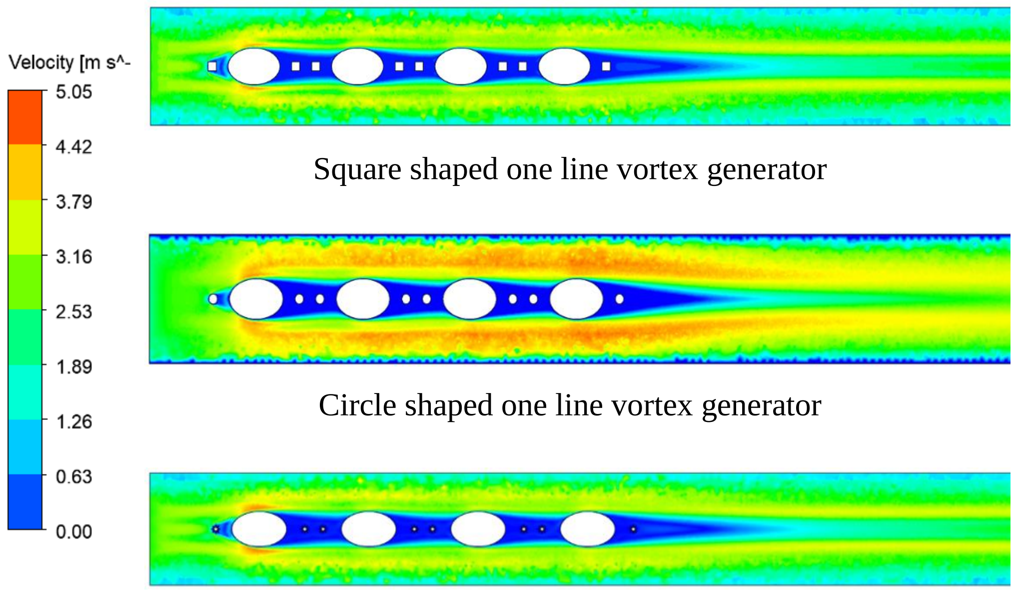


FIGURE 25:PRESSURE DISTRIBUTION OF ONE LINE 3 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

7.2. VELOCITY FIELD:

Fig 26,27,28 demonstrates the physical domain velocity contour of vortex generators of different shapes in three corrugated fins on the plane. The effect of vortex in three corrugated fins and pipes was observed to have effects on the velocity contour of the fin and heat exchanger pipes with the shift in the sum of Reynolds. In terms of variations in morphology and design, use of the vortex with various Reynolds can shift the flow patterns through FTCHE and speed up flow rate towards wake areas, particularly on tuber surfaces.The inlet velocity of FTHE increases with the increase of Reynolds number. From the fig it was found that with Reynolds number the velocity range increases. There is a lack of a mixing system at the smaller Reynolds number in the wake area which results in a decreased convective heat transfer. The greater number of Reynolds increases the liquid mixing process and speeds up the flow rate across the tuber layer. The fig was shown to have great impact on the contour of the vortex generators design It slows the stream isolation, increases the flow rate, decreases the tube flow and leads the principal flow into the wake area.

Wake and stream mixing are correlated with the underlying principle of convective thermal transfer. It was n, too. The application of differnet vortex generators to the FTCHE has contributed to the mixing of the bulk material, destabilized flow and changed the thickness of



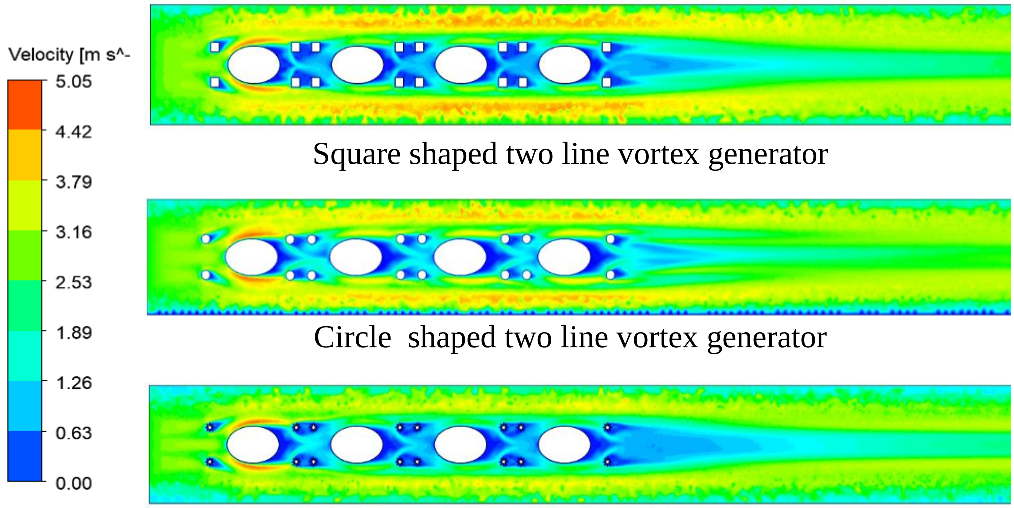
Square shaped one line vortex generator

Circle shaped one line vortex generator

Star shaped one line vortex generator

FIGURE 26VELOCITY DISTRIBUTION OF ONE LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

the boundary layer across the FTCHE. The result of square shaped vortex generator models with a greater number of Reynolds was shown in the picture with an improvement in the mixing process as well as improvements in the boundary layer.



Square shaped two line vortex generator

Circle shaped two line vortex generator

Star shaped two line vortex generator

FIGURE 27: VELOCITY DISTRIBUTION OF TWO LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

7.3. TEMPERATURE DISTRIBUTION:

Figure 28,29,30 indicates the fluid flow temperature contours in one specific chosen surfaces in the physical areas at h=.72. The results show the Reynolds numbers from 200 to 900 for the temperature contours. The influence on the distribution of temperature in the crossflow between different shapes of vortex generator three-corrugated fin areas of the corrugated fin and oval tubes can be clearly seen. In these two FTCHEs, the increase for temperature contour can be clarified by raising the vortex core area and reducing the deficiency region in which the mixing system is more liquid in the back of the pipes.The increase in average temperatue is highest in square shaped vortex generator .It is clearly seen in the courtour .the temperature change is comparatively low in the circle shaped vortex generator .

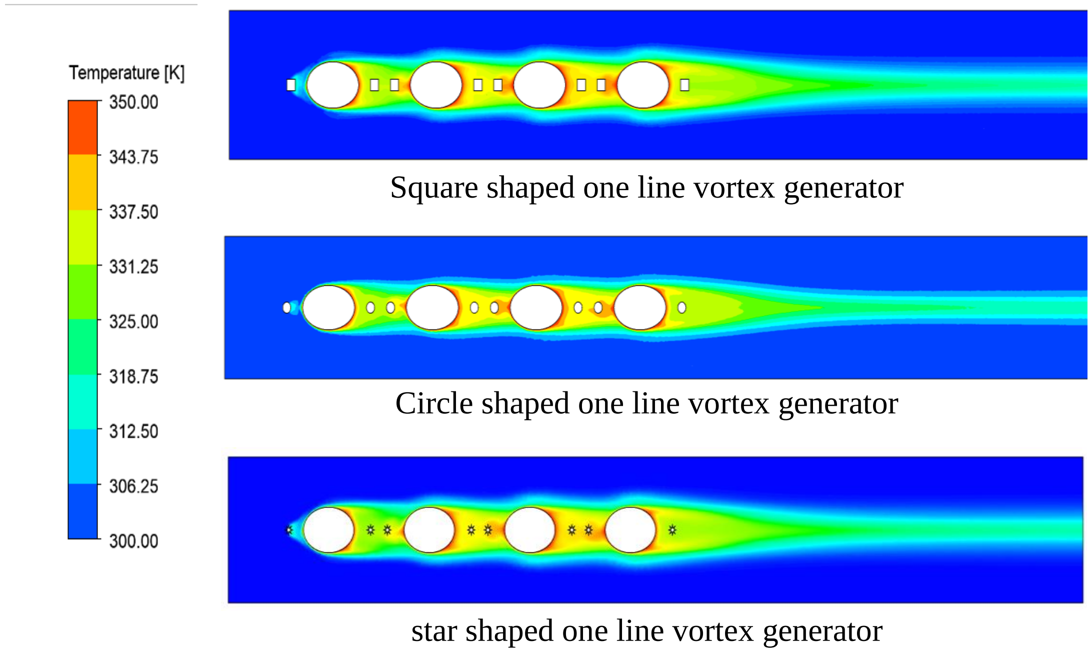


FIGURE 28:TEMPERATURE DISTRIBUTION OF ONE LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAP

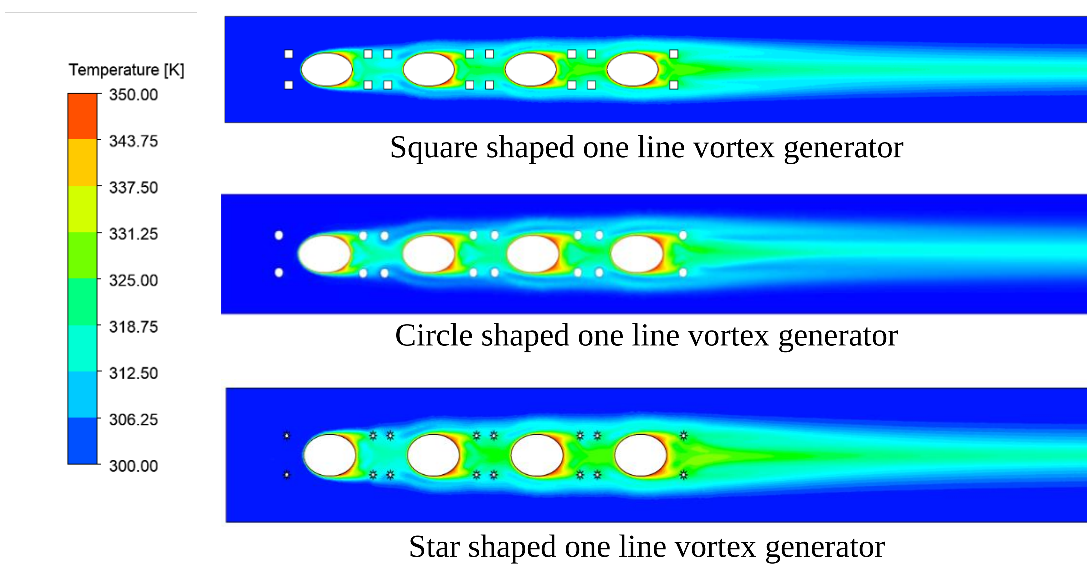
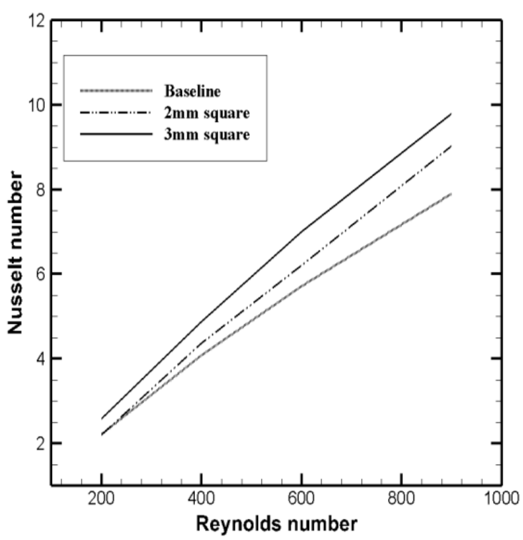
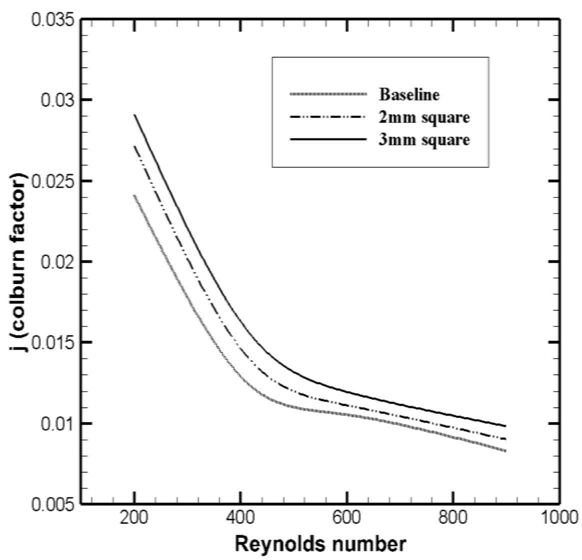


FIGURE 29 :TEMPERATURE DISTRIBUTION OF ONE LINE 2 MM VORTEX GENERATOR WITH (SQUARE ,CIRCLE ,STAR) SHAPE

7.4. HEAT TRANSFER CHARACTERISTICS

Figure 30 shows the average Nusselt number of different FTCHE cases with and without a vortex generator. The findings agree well with the effects of the temperature distribution as shown in section. The average number of Nusselt increases with the number of Reynolds in every case. The average number of Nusselt cases varies for each number of Reynolds.The variation of nusselt number happens due to the change of shape of the vortex .From the graph it is evident that the vortex with square shape has highest nusselt number and the shape with star has second highest and circle becomes the lowest in increment in nusselt number .The increase of nusselt number of square shaped vortex generator is 16-23% with respect to base line.





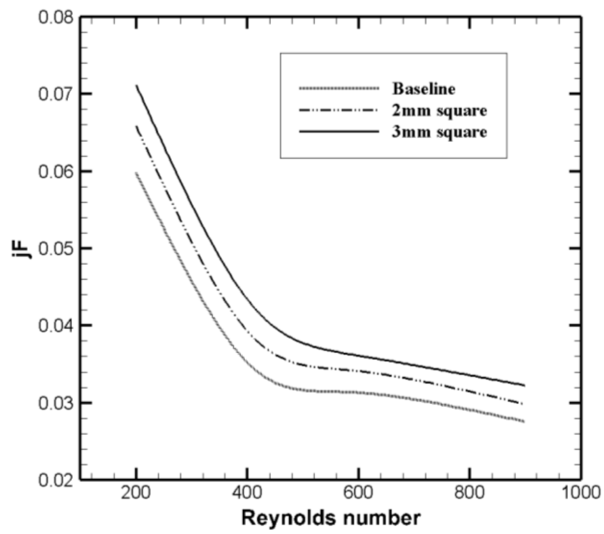


FIGURE 30: COMPARISON OF HEAT TRANSFER CHARACTERISTICS OF SQUARE SHAPED VORTEX GENERATOR WITH BASELINE AND DIFFERENT SIZES .

Nusselt number for 3mm square VGs fin is increased by 16.65-23.8% & 2mm square fins 1%-14.1% over the baseline(three corrugated fins)Colburn factor for 3mm square VGs fin is increased by 13.4-26.1% & 2mm square fins 8.4%-13.2% over the baseline(three corrugated fins)The overall thermal performance for 3mm square VGs fin is increased by 15.4-23.1% & 2mm square fin is 8.1%-11.5% over the baseline(three corrugated fins)

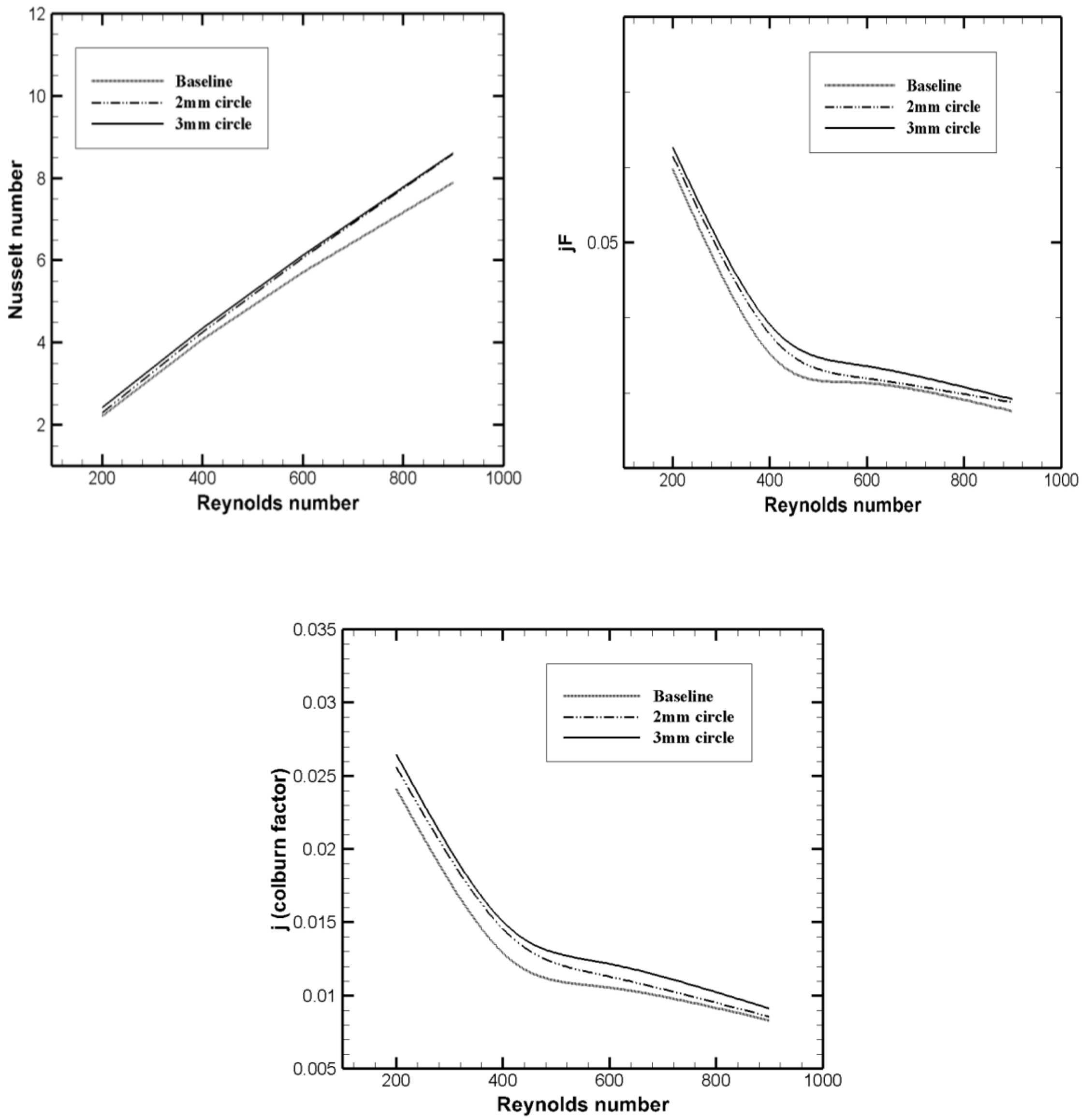


FIGURE 31 : COMPARISON OF HEAT TRANSFER CHARACTERISTICS OF CIRCLE SHAPED VORTEX GENERATOR WITH BASELINE AND DIFFERENT SIZES .

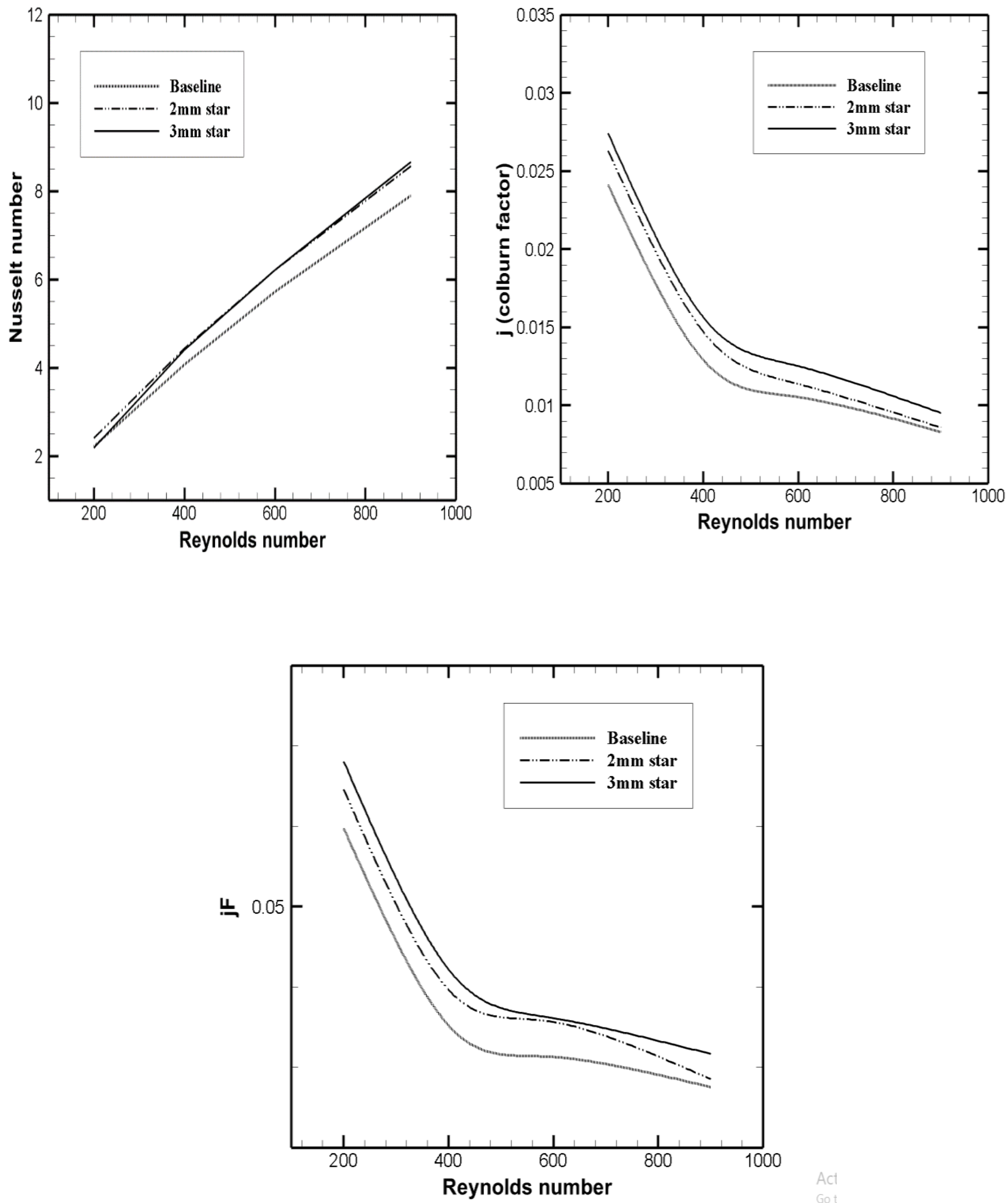
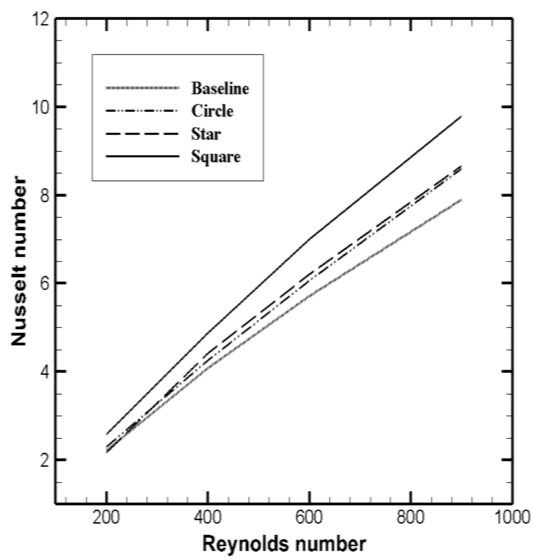


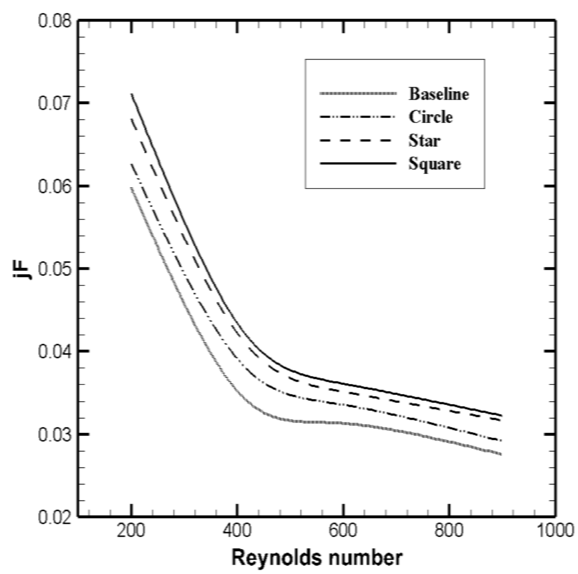
FIGURE 32: COMPARISON OF HEAT TRANSFER CHARACTERISTICS OF CIRCLE SHAPED VORTEX GENERATOR WITH BASELINE AND DIFFERENT SIZES

From baseline the nusselt number increases in square shaped VGs is 16.65-23.8% in star shaped VGs is 1.85-9.6% in circle shaped VGs is 6.7-9.02% .

The colburn factor is increased from baseline In square shaped VGs is 13.4%-26.1% In star shaped VGs is 13.7%-21.1% In circle shaped VGs is 9.6%-16.4% .

The overall thermal performance is increased from baseline In square shaped VGs is 16.1%- 23.19% In star shaped VGs is 13.6%- 19.83% In circle shaped VGs is 7.3%- 11% .





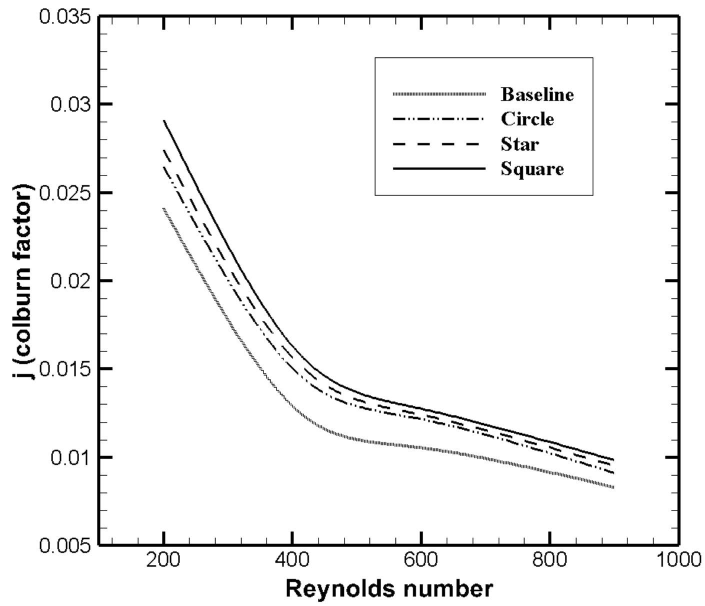


FIGURE 33: COMPARISON OF HEAT TRANSFER CHARACTERISTICS OF DIFFERENT SHAPED VORTEX GENERATOR WITH BASELINE .

8. CONCLUSION

Three-dimensional computational fluid dynamic simulations have been performed to investigate the thermal and hydraulic performance across the FTCHE with and without vortex generator in various geometries and arrangements. In this study three different corrugation geometries are investigated and three different vortex (square ,circle ,star) shapes are implemented to enhance the heat transfer enhancement .The vortex generators are placed behind the tubes . The results obtained has been presented in this paper. Some of the conclusions for the present study are as follows:

- From three configurations of FTCHE ,the Three corrugated fin gives the best heattransfer enhancement . Nusselt number for the FTCHE with three corrugated fins can be increased by 21.847-36.18% over the baseline case and the corresponding pressure difference decreased up to 23% varying with different

Reynolds number . three corrugated fin has a better overall thermal hydraulic performance and also better heat transfer than other corrugated fins and it is enhanced upto 8.82 -37.4%

- Along with the introduction of vortex in three corrugated fin the heat transfer is enhanced and the enhancement continues with the increase of vortex size .camparing between 2 mm and 3 mm square shaped vortex , the nusselt number increases 8.4-17.6% in 3 mm vortex from 2 mm .the overall hydraulic performance of 3 mm vortex is also 5.8-10.4% higher than 2 mm vortex .So, 3 mm vortex gives better heat transfer enhancement in every shapes .

- The heat transfer enhancement varies with the change of vortex shape .From baseline the nusselt number increases in star shaped vortex is 1.85-9.6% ,in circle shaped vortex is 6.7-9.02% and in square shaped vortex is 16.65-23.8%.The overall thermal performance is increased up to 23.19% in square shaped vortex ,up to 19.83% in star shaped vortex ,up to 11% in circle shaped vortex .From this calculation it is evident that square shaped gives the best thermal performance irrespective of other vortex geometries .

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