بسم الله الرحمن الرحيم

ISLAMIC UNIVERSITY OF TECHNOLOGY

DHAKA, BANGLADHESH

ORGANIZATION OF ISLAMIC COOPERATION

A thesis submitted to the Department of Mechanical and Chemical Engineering (MCE), Islamic University of Technology (IUT), in the partial fulfillment of the requirement for degree of Bachelor of Science in Mechanical Engineering.

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CERTIFICATE OF RESEARCH

The thesis title "***Heat Transfer Augmentation of Compact Heat Exchangers Using Modern Innovative Techniques, A review and Parametric study***" submitted by **Muhammad Awais (Student no: 131469)** has been accepted as satisfactory in partial fulfillment of the requirement for the Degree of BSc in Mechanical and Chemical Engineering on November, 2017.

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DECLARATION

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

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ACKNOWLEDGEMENTS

I express my heartiest gratefulness to Almighty Allah for His divine blessings, which made us able to complete this thesis successfully. The project was carried out by the author under the close supervision of ***Dr. Arafat Ahmed Bhuiyan*** ***Lecturer*** **of Mechanical and Chemical Engineering, Islamic University of Technology (IUT)**. The supervisor had dedicated his valuable time, even at odd hours, which had ensured the completion of this project. We express profound gratitude to the supervisor for his support, cooperation and guidance in accomplishing the project.

I wish to take this opportunity to express my sincerest gratitude and heartiest thanks to my dear parents and family members for being such delightful people and the inspiration for our effort. Without their prayers, continued encouragement, moral and financial support, it would not have been easy for us to accomplish this work, indeed their reward is with the Almighty Allah. I also thank to all my friends, colleagues and whoever helped us in the course of our work and or write-up. We seek excuse for any errors that might occur in this report despite of our best efforts.

ABSTRACT

This project presents a comprehensive review and numerical study on different ways of enhancing heat transfer rate and pressure loss reduction in compact heat exchangers (CHXs). The sole objective of this study is to gather major thermodynamic features of CHXs presented by researchers through both experimental and numerical investigation for innovative designing purpose of heat exchangers. The influence of fins and tubes spacing, geometry and shape on heat transfer performance is widely discussed. The effectuality of different fins types and their pattern, height, pitch/spacing on heat transfer augmentation and pressure drop reduction were considered. It is seen that convex louver fin yields better heat transfer performance than plane and louver fins. The influence of fin spacing on heat transfer performance varied with Reynolds number but their impact on pressure drop is negligible. The impact of elliptical and circular tubes with inline and staggered alignment on heat transfer enhancement and pressure loss is also widely discussed. Staggered alignment as compared to inline alignment of tubes in HXs leads to the higher heat transfer performance at the expense of larger pressure drop and also the optimal position for tubes is at downstream region instead of upstream region due to the significant enhancement of heat transfer performance because of horseshoe vortex formation.

A numerical study was performed to acknowledge the influence of vortex generators on heat transfer augmentation and pressure drop reduction. The impact of VGs row numbers with different configuration is widely discussed. The effect of tubes arrangement i.e. inline and staggered on heat transfer performance and pressure drop of compact heat exchanger is also elucidated. It was found that CHXs with vortex generators yield better heat transfer rate as compare to compact heat exchanger without VGs. However, this arrangement yields higher pressure drop. Moreover, it was found that staggered tube arrangement tends to give higher heat transfer performance than inline arrangement of tubes.

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CHAPTER 1: INTRODUCTION

1.1: Motivation for heat transfer enhancement:

For well over a century, efforts have been made to produce more efficient heat exchangers by employing various methods of heat transfer enhancement. The study of enhanced heat transfer has gained serious momentum during recent years, however, due to increased demands by industry for heat exchange equipment that is less expensive to build and operate than standard heat exchange devices. Savings in materials and energy use also provide strong motivation for the development of improved methods of enhancement. When designing cooling systems for automobiles and spacecraft, it is imperative that the heat exchangers are especially compact and lightweight. Also, enhancement devices are necessary for the high heat duty exchangers found in power plants (i. e. air-cooled condensers, nuclear fuel rods). These applications, as well as numerous others, have led to the development of various enhanced heat transfer surfaces.

In general, enhanced heat transfer surfaces can be used for three purposes: (1) to make heat exchangers more compact in order to reduce their overall volume, and possibly their cost, (2) to reduce the pumping power required for a given heat transfer process, or (3) to increase the overall UA value of the heat exchanger. A higher UA value can be exploited in either of two ways: (1) to obtain an increased heat exchange rate for fixed fluid inlet temperatures, or (2) to reduce the mean temperature difference for the heat exchange; this increases the thermodynamic process efficiency, which can result in a saving of operating costs.

Enhancement techniques can be separated into two categories: passive and active. Passive methods require no direct application of external power. Instead, passive techniques employ special surface geometries or fluid additives which cause heat transfer enhancement. On the other hand, active schemes such as electromagnetic fields and surface vibration do require external power for operation.

1. Increase the effective heat transfer surface area (A) per unit volume without appreciably changing the heat transfer coefficient (h). Plain fin surfaces enhance heat transfer in this manner.

2. Increase h without appreciably changing A. This is accomplished by using a special channel shape, such as a wavy or corrugated channel, which provides mixing due to secondary flows and boundary-layer separation within the channel. Vortex generators also increase h without a significant area increase by creating longitudinally spiraling vortices exchange fluid between the wall and core regions of the flow, resulting in increased heat transfer.

3. Increase both h and A. Interrupted fins (i. e. offset strip and louvered fins) act in this way. These surfaces increase the effective surface area, and enhance heat transfer through repeated growth and destruction of the boundary layers.

1. *Influence of interrupted surfaces on heat transfer augmentation and pressure drop:*

Heat exchangers with fins and round tubes are widely used in industrial, air-conditioning and refrigeration applications to meet the demand for saving energy and resources. To reduce size and weight of heat exchangers, various fin patterns have been developed to improve the air-side heat transfer performance. Typical fin geometries are plate, wavy and louver fin surfaces. Generally, the complexity of the air flow pattern across the fin-and-tube exchangers makes the numerical simulation very difficult.

Extended or finned surfaces are widely used in compact heat exchanger to enhance the heat transfer and reduce the size. Common among these are automobile radiators, charge air coolers, automobile air-conditioning evaporators and condensers to meet the demand for saving energy and resources. In these applications, the heat transfer is normally limited by the thermal resistance on the air side of the heat exchangers. Therefore, various augmented surfaces have been developed to improve air side heat transfer performance. Typical fin geometries are plain fins, wavy fins, offset strip fins, perforated fins and multi-louvered fins, which, besides increasing the surface area density of the exchanger, also improve the convection heat transfer coefficients. Of these, wavy fins are particularly attractive for their simplicity of manufacture and potentials for enhanced thermal-hydraulic performance. The air side thermal hydraulic performance of wavy fin and round tube heat exchangers have been studied by many researchers.

The heat transfer performance of an air-cooled heat exchanger is highly dependent on the pattern of fins. To meet the demand for saving energy and resources, heat exchanger manufacturers are trying to increase performance, and to reduce size and weight. For a typical fin-and-tube heat exchanger, the thermal resistance is generally on the air side. As a result, complicated fin patterns have been developed to obtain better air-side performance. Convex-louver fins were developed to improve air-side heat transfer performance.

The plate fin and tube geometry is a common configuration in heat exchangers. These heat exchangers are commonly operated with liquid inside the tubes and air on the outside, because of which the external thermal resistance is usually the most critical. Some of the characteristics of the hydrodynamics of plate-fin and tube geometries are: boundary layers developing from the leading edges of the fins and over the tube, roll-up of these boundary layers into horseshoe vortices ahead and around the tube, separation of the boundary layer on the tube and recirculation bubbles or vortex shedding in the wake. At small fin spacings the streamlines are analogous to those for potential flow over a cylinder.

In plate-fin or fin-tube heat exchangers the flow between the plates can be considered as a channel flow. The fluid with poor heat transfer characteristics flows around the tubes through the channels formed by neighboring fins. The purpose of fin is to improve the rate of heat transfer on the gas side to a value comparable to liquid side. The flow structure is characterized by the fin-tube interaction. Near the fin leading edge developing channel flow occurs. The interaction of the channel flow and tube cross-flow leads to the formation of a strong horseshoe. vortices which extends far into the tube wake and increase heat transfer on the cylinder and fin in its region of influence. However, behind the tube the heat transfer from the fin is reduced particularly in the re-circulation zone. The degradation of the heat transfer in the wake of the tube can be avoided by introducing longitudinal vortices in the wake. A simple way to generate longitudinal vortices is to punch or attach small triangular winglets on the fin with an angle of attack to the main flow direction. The resulting longitudinal vortices in the wake of the vortex generator disturbs the boundary layer growth on the fin, and mixes the hot and cold fluids continuously, thereby increasing heat transfer.

CHAPTER 2: LITERATURE REVIEW

Asako and Faghri [6] numerically investigated two-dimensional steady laminar flow with Re = 100– 1000, and heat transfer in plate channels with triangular profiled wall corrugations that are maintained at a uniform temperature. Subsequently, triangular corrugations with round corners were considered [7]. Zhang et al. [5] numerical investigated the effects of wall-corrugation aspect ratios and fin space ratios on the vortex structure and enhanced heat transfer for low rate with Re = 100–1000. Metwally and Manglik [8] have investigated two-dimensional periodically developed laminar flow and heat transfer in sinusoidal wavy channel with different corrugation aspect ratios. More complex three-dimensional, cross-corrugated have also been computationally modeled in a few recent studies [9–11]. Goldstein and Sparrow [12] first studied the corrugated channels with triangular waves used the naphthalene sublimation method. Rush et al. [13] conducted flow visualization test for sinusoidal wavy passages to investigate the local heat transfer and flow behavior of the fluid in the laminar and transitional flow region. Using the visualization methods, they reported that the flow field is characterized as steady and unsteady and the location of the onset of mixing is found to depend on the Reynolds number and channel geometry. In all, it has generally been observed that wall corrugations induce a steady vortex or swirl flow in the trough region of the wavy wall in the low Reynolds number. This results in flow mixing and boundary layer disruption and thinning, thereby significantly enhancing the heat transfer. For wavy fin-and-tube heat exchangers, experimental data were reported several times in the literature. Giovannoni and Mattarlo [1] presented the heat transfer coefficient data for sample coils; however, friction data were not reported in their study. Beecher and Fagan [2] tested 27 fin-and-tube heat exchangers, with 21 of them having wavy fin patterns. All the test samples by Beecher and Fagan [2] were three-row staggered arrangements.

Fin-tube heat exchangers are employed in a wide variety of engineering application, for example, in a geothermal, fossil, process plant and air-conditioning, etc. It is well known that the horseshoe vortices formed around tubes on fins enhance the heat transfer of fins, causing a large pressure loss due to form-drag in such heat exchangers [3,4]. In order to enhance fin-side heat transfer between the fin and the gas flow, one way is to use vortex generators to produce longitudinal vortices inducing strong swirling motion that serves to bring about the enhancement of heat transfer at a modest expense of the additional pressure loss. Fiebig et al. [10] investigated experimentally the influence of delta winglet position on the fin surface having single tube, by punching a pair of vortex generators ahead and behind the tube, and found an increase of local heat transfer coefficient up to 100% and mean heat transfer coefficient up to 20% at the optimum winglet position immediately behind tube, for the Reynolds number ranging from 2000 to 5000 based on fin pitch, H. They also observed a reduction of 10% in the pressure loss, and explained as the delayed separation on the tube due to longitudinal vortices generated by the vortex generator which introduces high momentum fluid into the wake region behind the tube. **L**ongitudinal vortices are generated naturally in fin-tube heat exchanger passages by the interaction of the flow velocity profile with the heat exchanger tube. These naturally occurring vortices are called horseshoe vortices. Longitudinal vortices can also be created through the use of winglet vortex generators mounted or punched into the fin surfaces. Jacobi and Shah [14] provide an excellent review of heat transfer enhancement through the use of longitudinal vortices. Various winglet shapes have been studied. Fiebig et al. [15], using the unsteady liquid crystal thermography technique, found that delta winglets provided the highest local heat transfer enhancement.

CHAPTER 3: NUMERICAL METHOD

3.1: GOVERNING EQUATIONS

The present study was performed considering thermal transport with convective heat transfer. Air is used as working fluid assuming constant properties (k=0.0261W/mK, μ=1.831x10^{-05} Ns/m^2, Pr=0.736, ρ=1.185 kgm^{-3}). Assuming a steady three-dimensional incompressible flow with no viscous dissipation and viscous work, effects of body force and buoyancy are neglected, both laminar and transitional flow conditions are considered. For transitional flow solutions three turbulent models are used namely standard k-ε, k-ω and RNG k-ε model. The flow is described by the conservation laws for mass (continuity), momentum (Navier-Stokes) and by the energy equations are as follows:

$$\left[\frac{\partial u_i}{\partial x_i} = 0\right] \tag{1}$$

$$\rho\left(u_j \frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_T)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

$$\rho C_P\left(u_j \frac{\partial T}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\lambda + \frac{\mu_T C_P}{Pr_T}\right)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

In Equations (2) and (3) μ_T and Pr_T are turbulent viscosity and turbulent Prandtl number respectively. As suggested by Yuan [14], Pr_T = 0.9 was used in the current study. The value of μ_T is determined based on the specific turbulence model that is being used. In k-ω turbulent model the μ_T is linked to the turbulence kinetic energy (k) and turbulence frequency (ω) via the following relation:

$$\mu_T = \rho\frac{k}{\omega} \tag{4}$$

The transport equations for *k* and ω were first developed by Wilcox [15] and later it was modified by Menter [16], can be expressed as:

$$\rho\left(u_j \frac{\partial k}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{k3}}\right)\frac{\partial k}{\partial x_j}\right] + P_k - \beta'\rho k\omega \tag{5}$$

$$\rho\left(u_j \frac{\partial \omega}{\partial x_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{\omega 3}}\right)\frac{\partial \omega}{\partial x_j}\right] + 2(1-F_1)\rho\frac{1}{\sigma_{\omega 2}\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j} + \alpha_3\frac{\omega}{k}P_k - \beta_3\rho\omega^2 \tag{6}$$

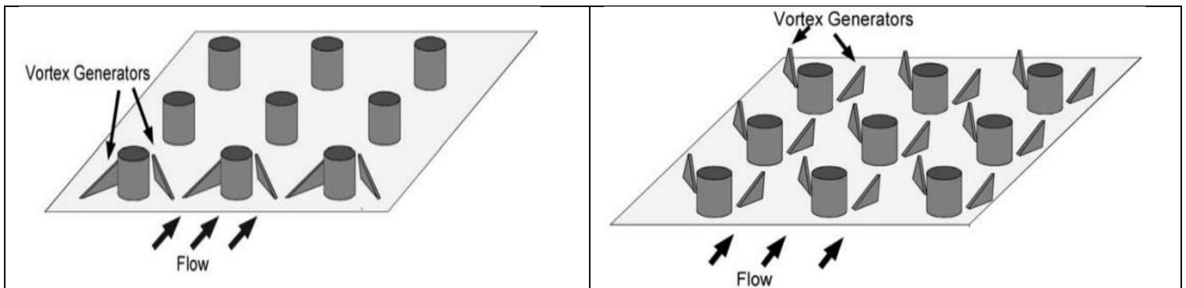
In equation (6), F_1 is a blending function and its value is a function of the wall distance. F_1 = 1 and 0 near the surface and inside the boundary layer respectively. The constants of this model (ϕ_3) are calculated from the constants ϕ_1 and ϕ_2 based on the following general equation.

$$\Phi_3 = F_1\Phi_1 + (1 - F_1)\Phi_2 \tag{7}$$

The model constants are given as $\alpha_1 = 5/9$, $\beta' = 0.09$, $\beta_1 = 0.075$, $\sigma_{\kappa 1} = 2$, $\sigma_{\omega 1} = 2$, $\alpha_2 = 0.44$, $\beta_2 = 0.0828$, $\sigma_{\kappa 2} = 1$, $\sigma_{\omega 2} = 1/0.856$. Details of these different turbulent models are documented in [13, 15-17]. In Eqns. (2) and (3) if we eliminate the terms containing μ_T then it becomes the equations for the laminar flow.

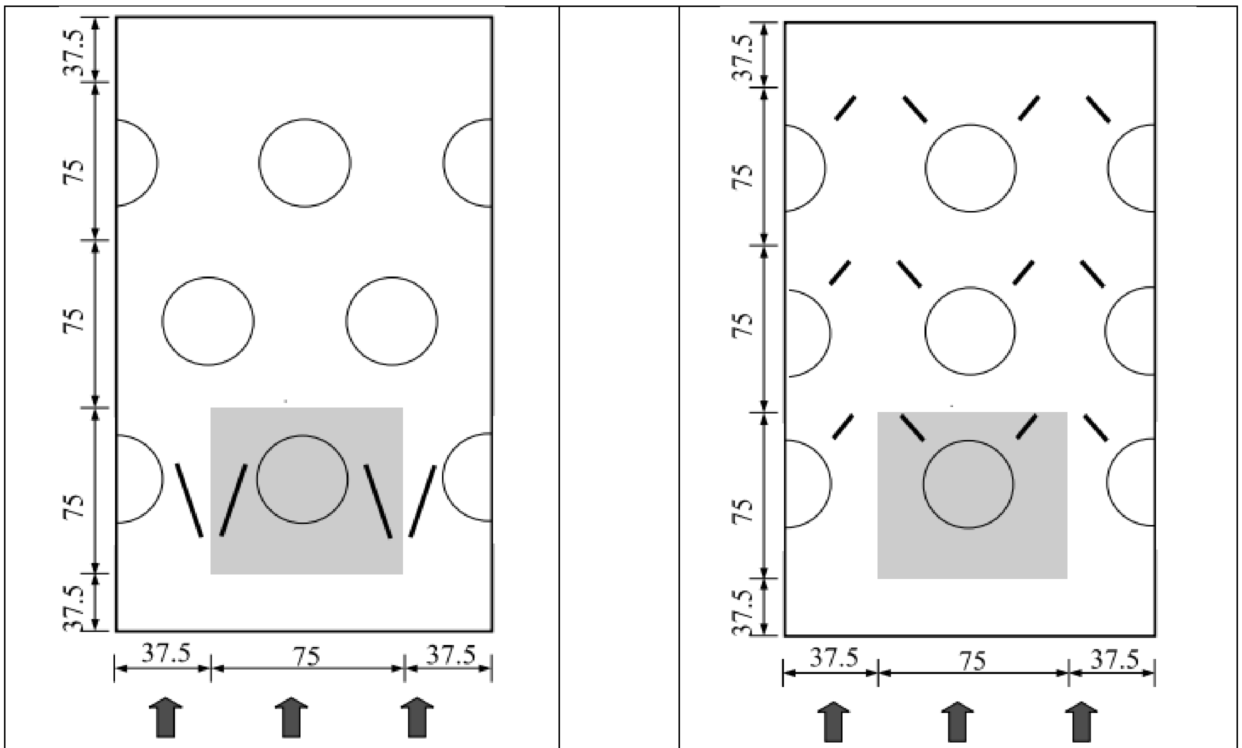
3.2: GEOMETRY AND COMPUTATIONAL DOMAIN

The geometric arrangements of the test-cores are shown in Fig. 1 and listed in Table 1. The fin pitch H is 5.6 mm. Both stream wise and spanwise pitches of tube banks are equally set to 75 mm. The vortex generators consist of delta winglet pairs made of 0.3 mm thick Bakelite. The present configuration of the delta winglet pair is called as common flow up configuration in which the spanwise distance between the leading edges of a winglet pair is wider than the one between the trailing edges. The base length l and height h of the winglet are 30 mm and 5 mm, respectively. In order to illustrate the excellent performance of the configuration proposed in the present study, the configuration with all three rows of winglet pairs shown in Fig. 2(b) and Table 1 was also examined experimentally in the present study. This common flow down configuration of winglet pairs with height/base length aspect ratio, h=l ¼ 1=2, was proposed by Fiebig et al. [2] that gave the best performance in their study.



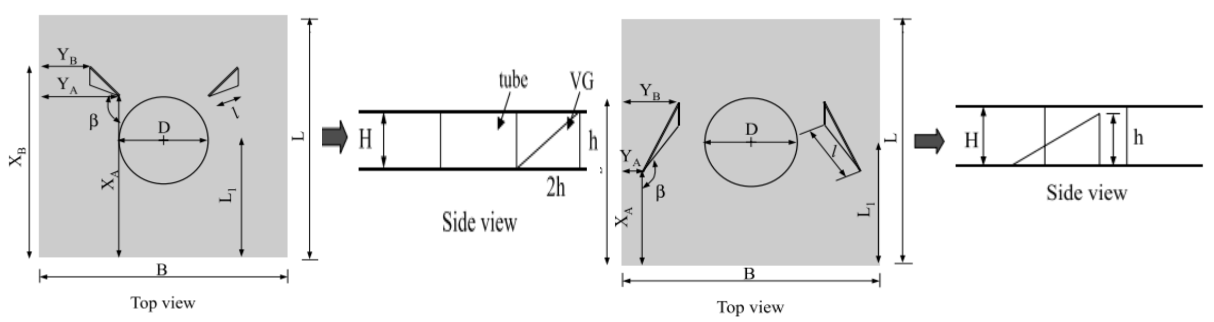
a b

Figure 1: Schematic diagram of different flow configuration, I). Common flow up, ii). Common flow down.



a b

Figure 2: Computational domain used for the modelling, a). Inline tube configuration, b). Staggered tube configuration [27]



(a) (b) (c) (d)

Figure 3 : Vortex position around tubes and across fins a-b).In common flow down configuration, c-d). In common flow up configuration [27]

CHAPTER 4: Cases considered

Followings are the important cases that we have considered in our numerical study to acknowledge their impact on heat transfer enhancement and pressure drop of compact heat exchanger. While varying the different velocities or Reynold number with tubes and Vortex Generators pattern and rows we've extracted various results. We have also varied the tube shape such as circular and oval types. We have considered two cases of VGs rows i.e. one and three rows aligned across the tubes. Their location around the tubes is very significant as perfect location yields suitable result.

<table>
<tr><td rowspan="2">Types of flow</td><td rowspan="3">No of winglets row</td><td rowspan="3">Tube Shape</td><td colspan="6">Tube arrangements</td></tr>
<tr><td colspan="3">In-lined</td><td colspan="3">Staggered</td></tr>
<tr><td rowspan="8">CFU
CFD</td><td>Re</td><td>u_{in}</td><td>θ</td><td>Re</td><td>u_{in}</td><td>θ</td></tr>
<tr><td rowspan="3">one row</td><td>Oval</td><td rowspan="7">200-2500</td><td rowspan="7">0.5-3.5</td><td rowspan="7">160°
165°
170°
175°</td><td rowspan="7">200-2000</td><td rowspan="7">0.5-3.5</td><td rowspan="7">160°
165°
170°
175°</td></tr>
<tr><td>Circular</td></tr>
<tr><td>Rectangular</td></tr>
<tr><td rowspan="4">Three rows</td><td>Oval</td></tr>
<tr><td>Circular</td></tr>
<tr><td>Rectangular</td></tr>
<tr><td>Circular</td></tr>
</table>

4.1: BOUNDARY CONDITIONS:

Velocity inlet is taken for inlet section of fins in Ansys fluent. Air is used as a working fluid here with following properties,

At inlet section, air is flowing with different velocities with changing Reynold numbers. For varying air velocities of 0.5 to 3 m/s corresponding Reynold number is 200 to 2500 with hydraulic diameter of cylindrical, elliptical and square tubes. Standard air properties are used at inlet section i.e. temperature and pressure is about 313k and 101kpa respectively. While air outlet is considered as pressure outlet with zero-gauge pressure. Top and bottom surfaces of channel is taken to be as a wall section with 370k temperature and no-slip condition. Vortices surfaces are also considered as a wall surfaces with same properties. Aluminum material is selected for whole channel with significant properties such as …

4.2: MESHING TECHNIQUES AND GRID TESTS:

Meshing is performed to divide the computational domain into several numbers of nodes or cells to obtain the result at throughout the boundary of computational domain. The outcomes of simulation depend on accuracy of meshing. As when computational domain is divided into more number of nodes it would provide better results than compared to the computational domain with less number of domains. We have started our meshing from dividing our geometry into 100000 elements then obtained results i.e. friction factor, pressure drop, Nusselt number and then went for next suitable highest number of elements i.e. 150000 and repeated the same process. To gain the better outcomes we went further and divided our geometry into more number of nodes i.e. 200000 and also found results for this grid elements. After comparing the result in form of pressure drop vs Reynold number as shown in fig below. It was found that computational domain with 150000 yields straight line or in other words provides good results as compared to the other grid elements.

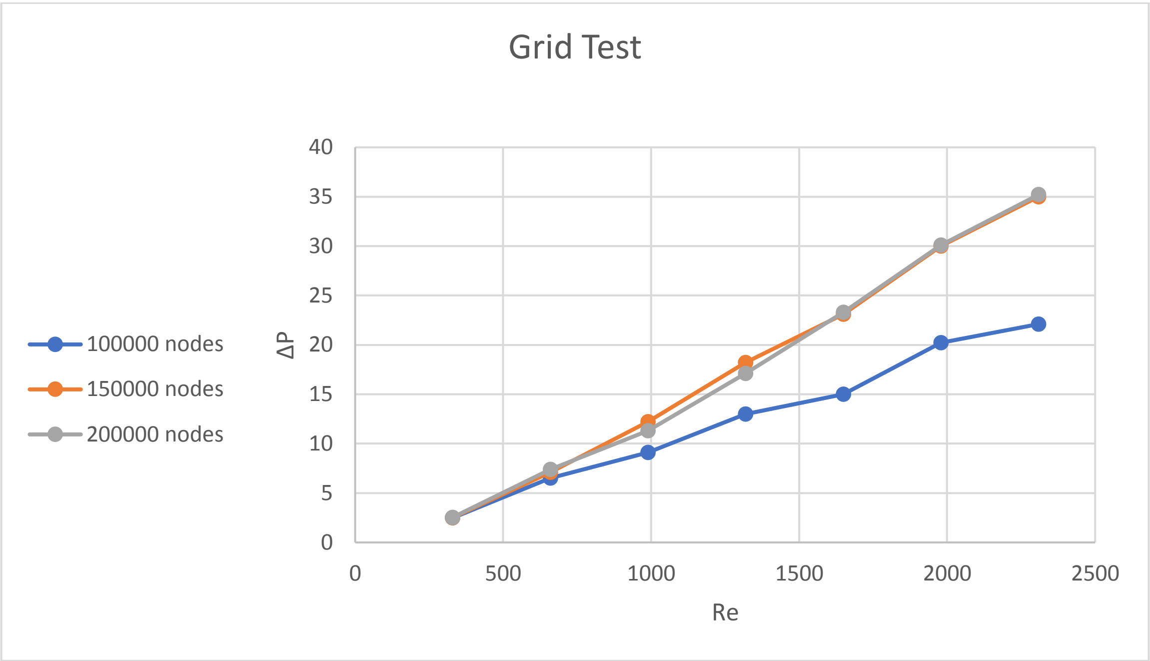


Figure 04: Grid independency test result

Figure 4: Grid Test

5: Results and Discussions:

5.1: Effects of number of winglets row (one, three):

Following figures elucidates the comparison of compact heat exchanger with and without the vortex generators. The influence of winglet row on temperature gradient, velocity distribution, and pressure drop is depicted in fig while keeping the Reynold number at 1320. It can be seen that heat exchangers without winglets possesses more poor heat transfer zone or wake region behind the tubes and yield low temp gradient at that region which in result reduces the overall temperature gradient. Horseshoe vortices are induced by tubes are only vortices produced in baseline HX which is the only reason of providing mixing of fluid and pressure drop.

While in case of fig and fig longitudinal vortices induced by winglets tends to yield different results. As these vortices enhance the flow mixing and provides better heat transfer performance. However, this heat transfer enhancement is penalized by higher pressure drop. As the number of winglet rows increases the pressure drop rises too with significant performance in terms of heat transfer rate.

PRESSURE CONTOUR:

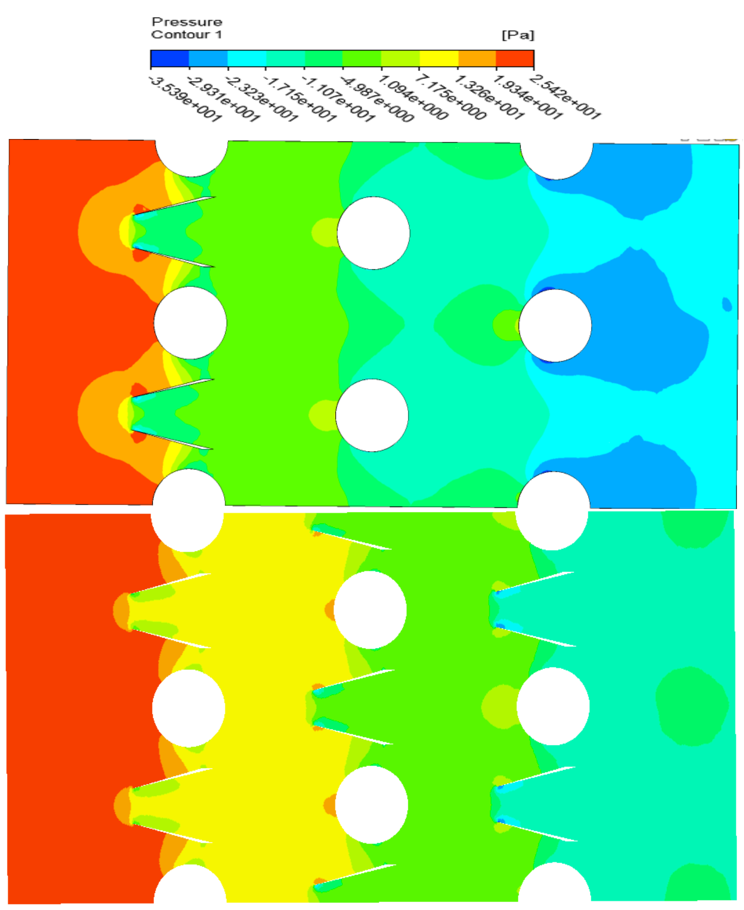


Figure 5: Effect of winglet row arrangement on Pressure drop with staggered tube arrangements and VGs attack angle 165^{0}

Velocity Contour:

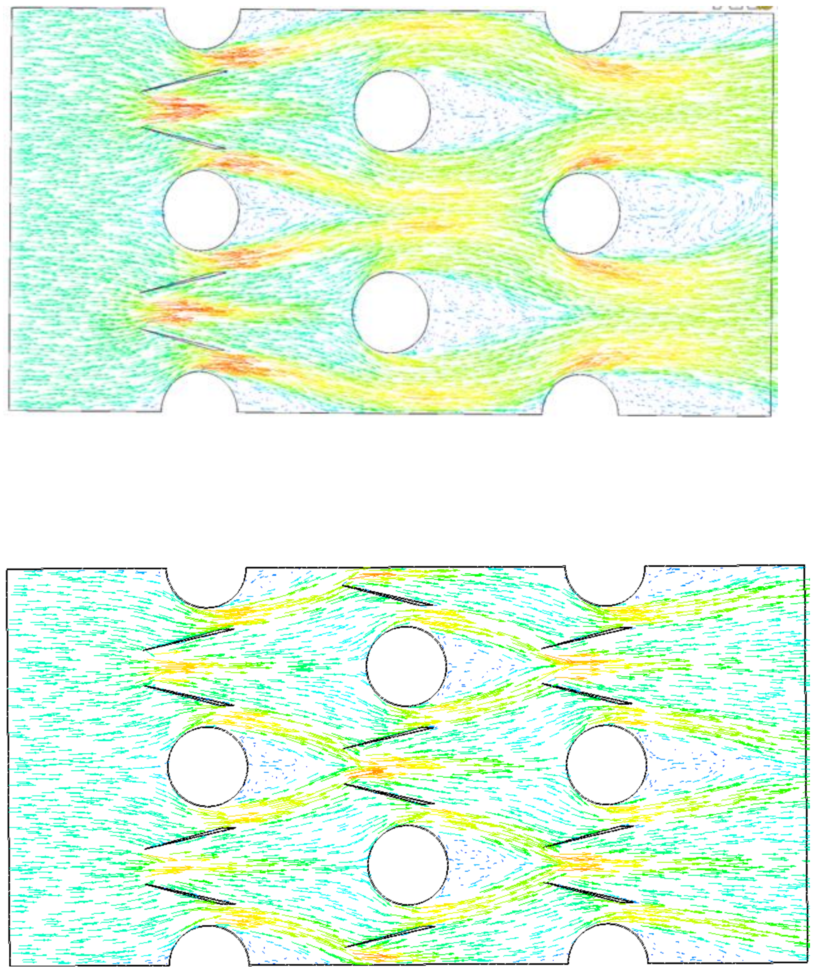


Figure 6: Effect of winglet row arrangement on Velocity contour staggered tube arrangements and VGs attack angle 165^{0}

5.2: Effects of inlet flow arrangements (up and down):

VGs can be incorporated in different configurations. The conventional configurations for the vortex generators are Common flow up (CFU) and Common flow down (CFD). Both configurations provide different results in terms of heat transfer and pressure drop. Common flow up configuration is the configuration in which the distance between leading edge of the vortex arranged around the first longitudinal row of tubes and leading edge of the front face of the channel is less. While on the other hand in the case of Common flow down configuration this distance increases as the vortex in this configuration arranged behind the tubes at suitable distance from the front faces of the tubes which makes this configuration a bit different from the formal configuration.

Both configuration produce longitudinal vortices which yield higher heat transfer performance at moderate pressure drop. However, LVGs induced CFD yields stable vortices and remains in the downstream region of the channels. Which makes it more preferable over CFU configuration. Moreover, it can be seen from the fig that pressure drop provided by CFD is higher than CFU configuration.

Temperature contour:

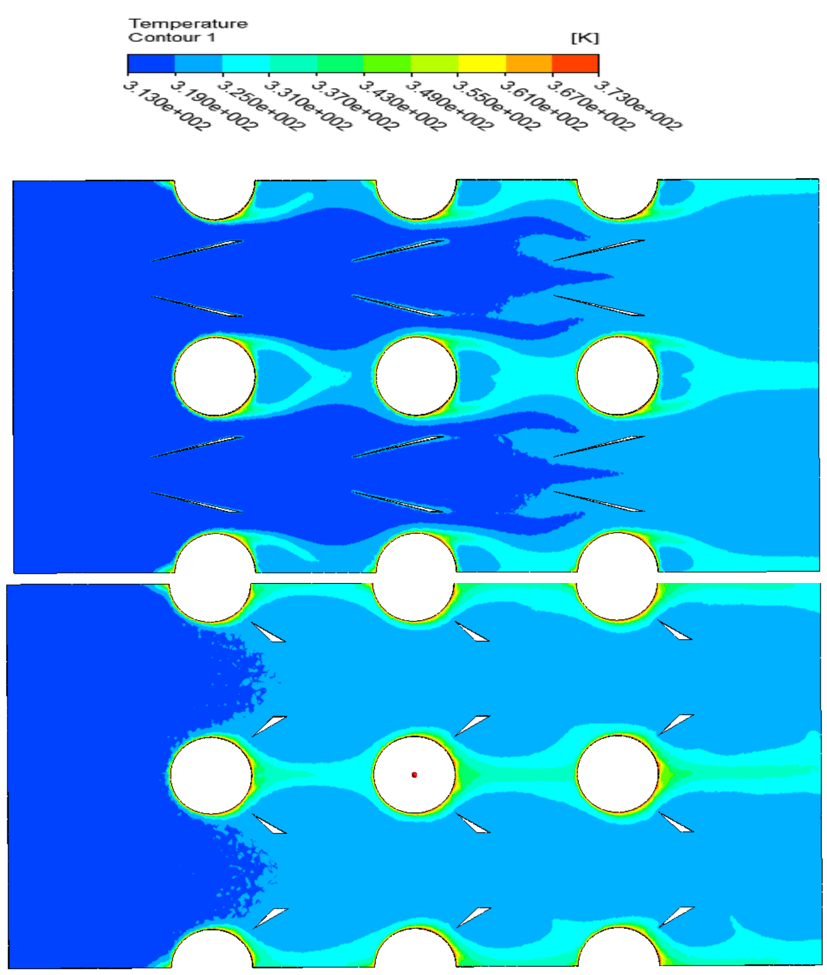


Figure 7: Effects of inlet flow arrangements on Temperature contour: Common flow up and common flow down configuration

Pressure contour:

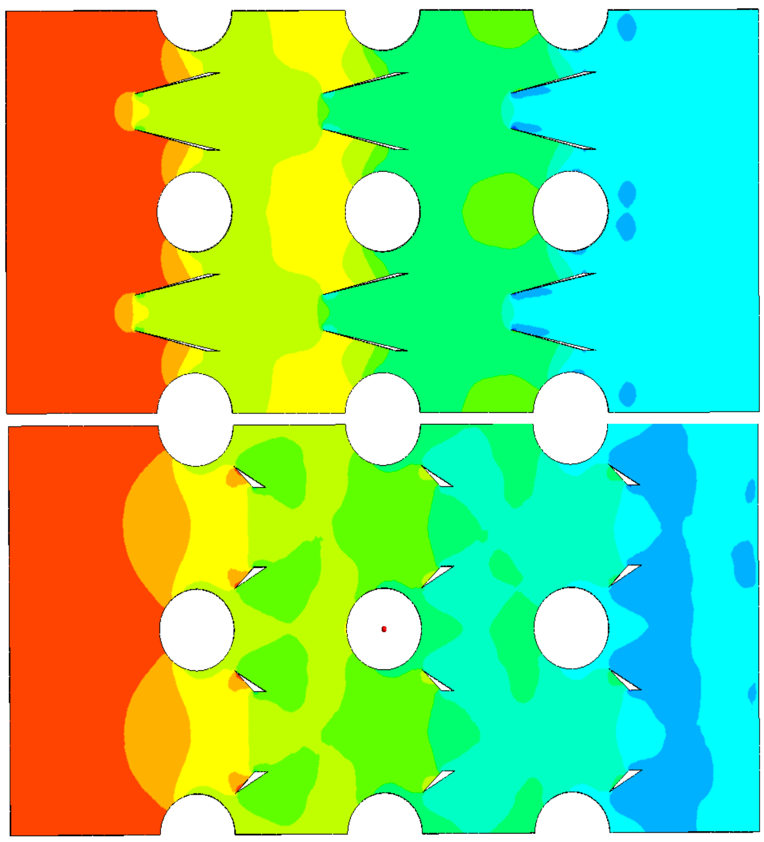


Figure 8: Effects of inlet flow arrangements on Pressure contour: Common flow up and common flow down configuration

5.3: Effects of tube arrangements (inline and staggered)

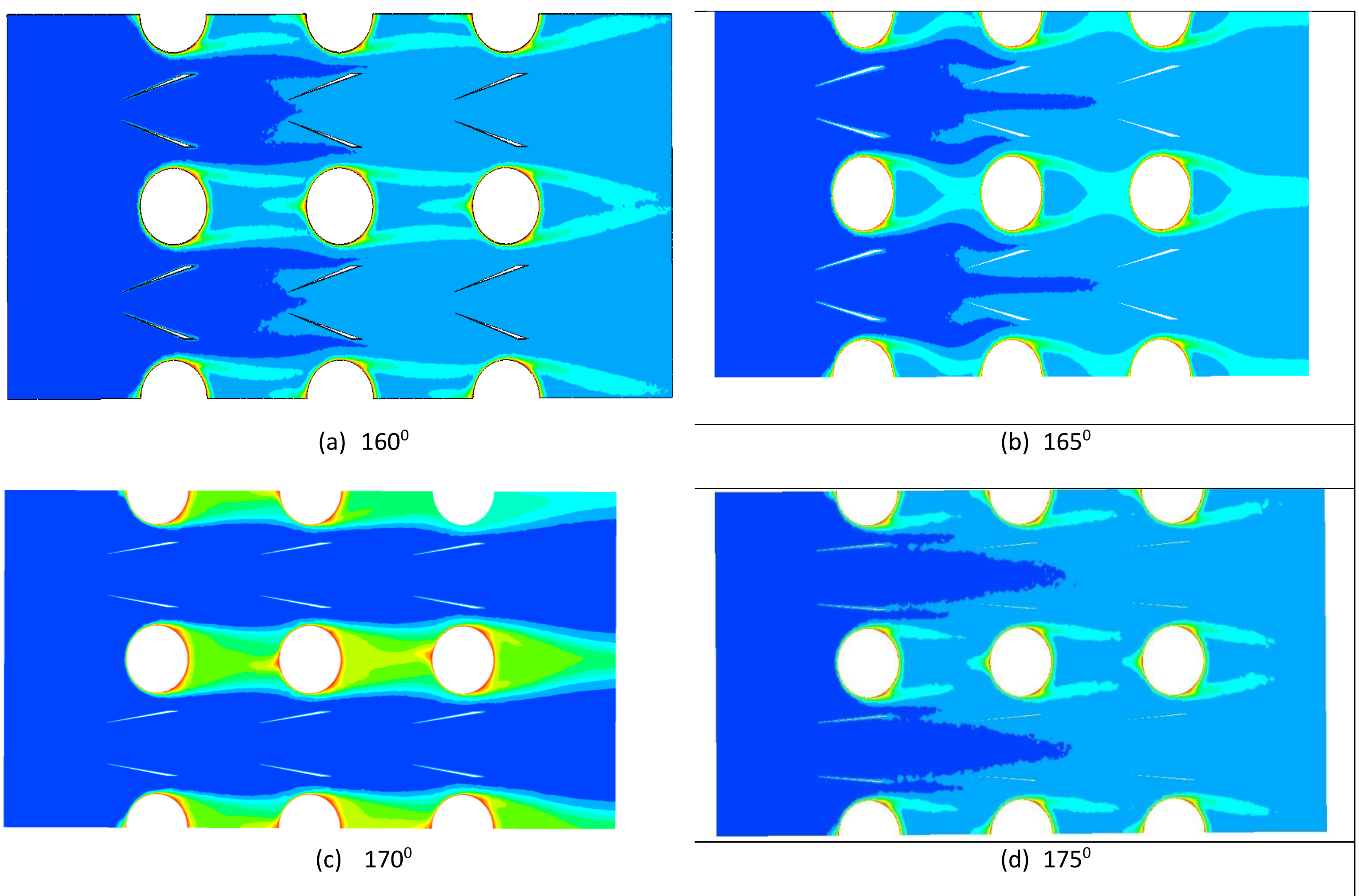
The inline and staggered arrangement of tubes is depicted in fig to fig with their influence on velocity, pressure and temperature of air flow in the channels. For inline arrangement, it was noticed that temperature gradient is a bit smaller as compare to staggered arrangement of tubes. This gradient difference yields higher heat transfer rate when channel is arranged with staggered arrangement of tubes. As higher heat transfer rate comes along with higher pressure drop.

It could be explained from the figures that huge glow circulation appears behind the tubes i.e. wake region is comparatively higher in result poor heat transfer zone behind the tubes.

5.4: Effects of angle of attack of vortex generators

The influence of attack angle of vortex generators also contributes significant influence on the performance of compact heat exchangers in terms of pressure drop and heat transfer augmentation. We have assumed three attack angles of VGs i.e. 165^0, 170^0, and 175$^{0.}$ As wake region reduction behind the tubes certainly dependent on the attack angles of VGs. As the attack angle rises the circulation zone behind the tubes increases with incorporating better flow mixing of hot and cold fluid. However, with the rise of the heat transfer performance pressure drop increases too. Higher attack angle tends to yield better performance. Moreover, it is mandatory to analyze the optimum attack angle at which desired heat transfer performance is achieved at moderate pressure drop. In our case we conclude attack angle of 30 and 45 yield desired outcomes. As very high attack angles lead to the reduction of LVGs formation and finally heat transfer performance.

Temperature contour:

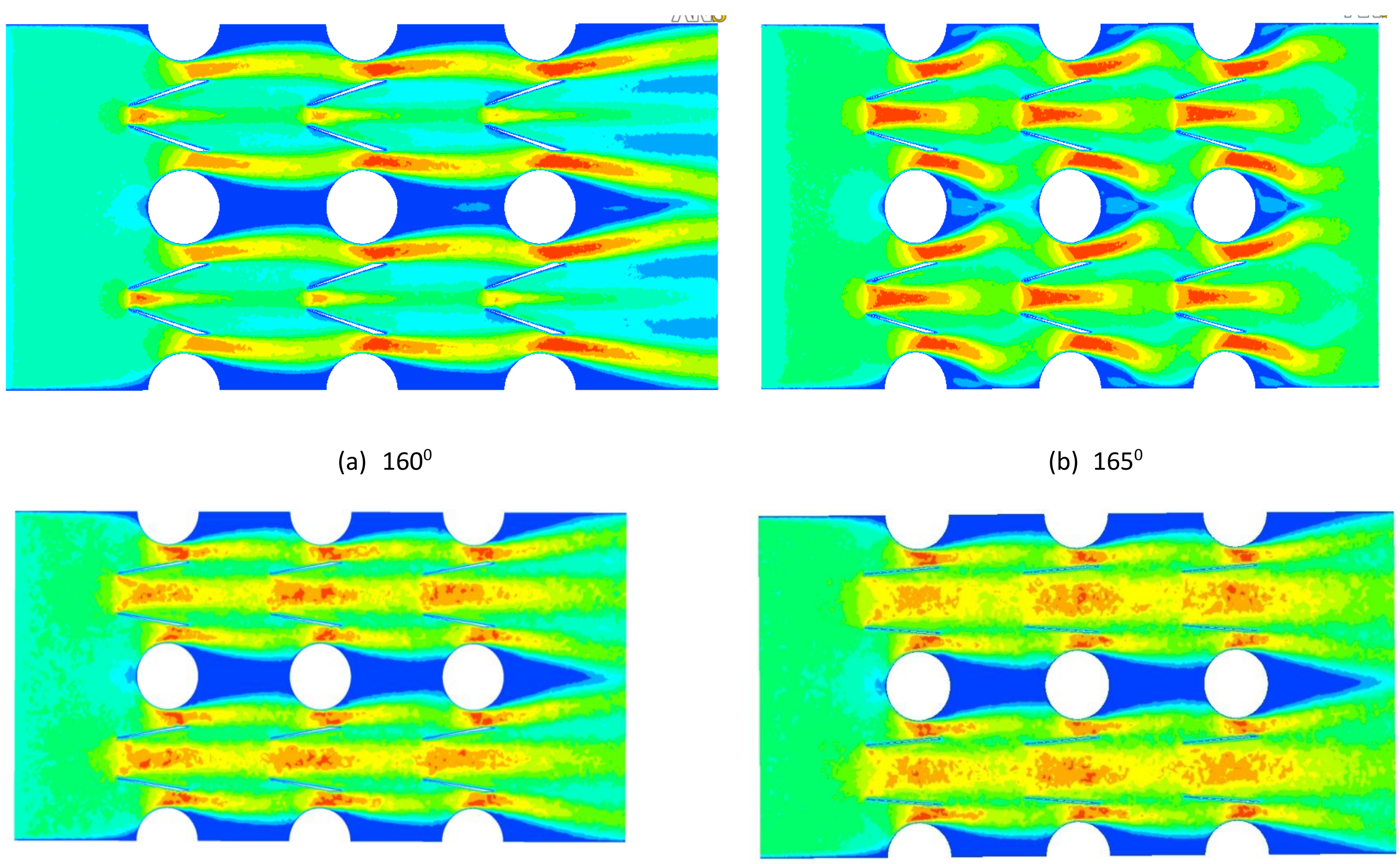


(a) 160^0 (b) 165^0

(c) 170^0 (d) 175^0

Figure 9: Influence of different attack angles of VGs on temperature gradient

Velocity contour:



(a) 160^0 (b) 165^0

(c) 170^0 (d) 175^0

Figure 10: Influence of different attack angles of VGs on velocity distribution

5.5: Effects of tube shape (oval, circular and rectangular)

Following contours clearly demonstrate the influence of tubes shapes on velocity, pressure and temperature profile. It can be noticed that circular and oval tubes yield better performance in terms of heat transfer performance and moderate pressure drop as compared to the square tubes. It could be explained as flow blockage by circular tubes is not so severe as compared to the square tubes i.e. flow profile across these tubes tends to diminish the wake region behind the tubes. while in case of square tubes this case does not applied.

Pressure contour:

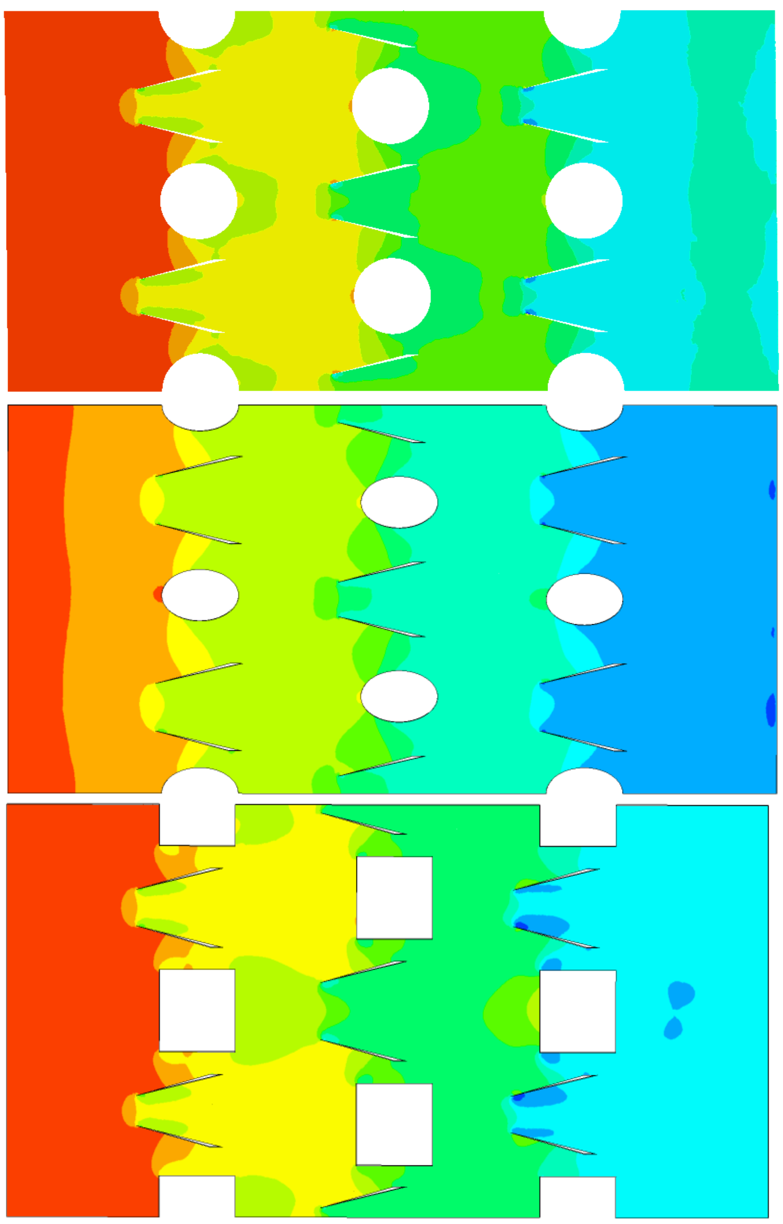


Figure 11: Influence of tubes shape on pressure drop

Velocity Contour:

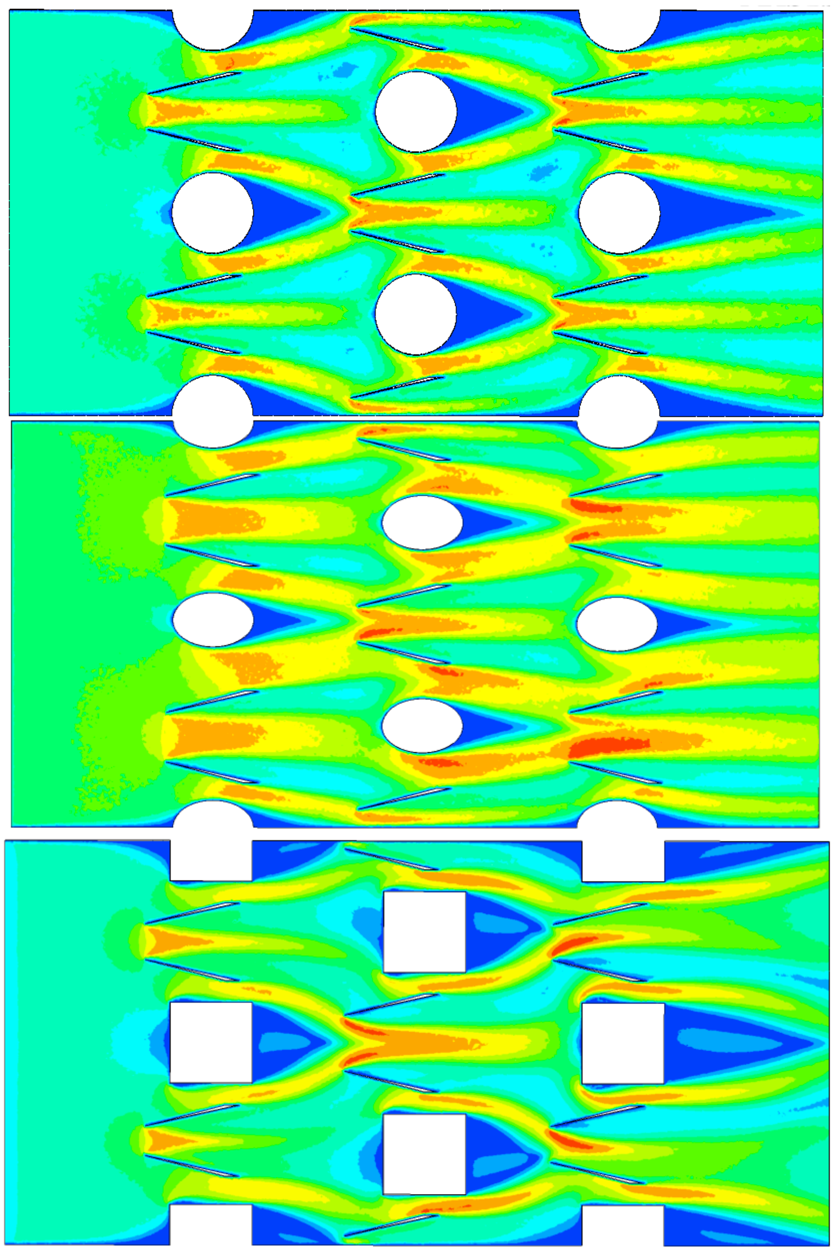


Figure 12: Influence of tubes shape on velocity contour

REVIEW STUDY:

2.1: Results and Discussion of Review study:

After carefully and thoroughly studying the recent scientific work regarding evaluating performance features of compact heat exchangers through different means and ways we've come up with following important results. These reviewed results would be very beneficial for the future researchers for designing compact heat exchangers with more compact size, low cost, high efficiency and performance rate etc.

The influence of different types of fins such as slit fins, wavy fins, herringbone fins, louver fins, plan fins etc. on compact heat exchanger performance is reviewed and their crucial results are listed below.

- The j and f factors decrease with increasing Re, in the tested range of Re, Re = 800–6500. And the j and f factors increases with fin space increasing at the same Re; the j factor increases with fin height, while the fin height has little effect on the f factor as a function of Re.
- Correlations of heat transfer and pressure drop for the wavy fins are developed. The proposed correlations give fairly good predictive ability against the present test data. The mean deviations of the correlations for j and f factors are 4.4% and 5.1%, and the average deviations are 0.4% and 0.3%, respectively.
- The effect of fin pitch on j and f factor is negligible. Considering the small variation of the fin pitch (from 1.3 mm to 1.7 mm), the foregoing argument should hold true for the present samples.
- Louver fin samples yield higher j and f factors than slit fin samples. For one row configuration, the average f factor ratio between slit fin sample and louver fin sample is 0.55. The ratio is 0.67 for two row configurations, and 0.71 for three row configurations. As for the j factor, the ratios are 0.79, 0.99 and 0.87 for one, two and three row configurations, respectively.
- The j factor decreases as the number of tube rows increases. The row effect on f factor is dependent on fin pattern. For slit fin, f factor is independent of the number of tube row. For louver fin, however, f factor decreases as the number of tube rows increases.
- The j=f ratios of louver fin samples are larger (38% for one row, 43% for two rows and 33% for three row) than those of slit fin samples. This suggests that slit ¯n geometry is advantageous than louver fin geometry on the basis of equal pressure drop.
- At the same frontal velocity, the pressure drops DP increases with increasing tube row numbers.
- For an inline arrangement, the pressure drops increase with the rise of tube diameter but the associated heat transfer coefficients decrease with it. The increase of fin height also gives rise to considerable increase of pressure drop but decrease of heat transfer coefficient.
- For the inline arrangement, the effect of fin spacing on the air side performance varies with the transverse tube pitch. For a larger transverse tube pitch of 71.4 mm, there is an effect on heat transfer coefficient but no detectable influence of the fin spacing on frictional characteristics. It is likely that this phenomenon is related to the considerable airflow bypass between tube rows. On the contrary, at a smaller transverse tube pitch of 50 mm, one can see smaller fin spacing results in higher pressure drops and lower heat transfer coefficients.

- For the staggered arrangement, the effect of tube diameter on the air side performance is analogous to that of inline arrangement but to a comparatively small extent. This is because the recirculation zone behind the tube row is much smaller in a staggered arrangement. The effect of the fin height on the pressure drops is much smaller than that of inline arrangement due to the major contribution to the pressure drops is from the blockage of the subsequent tube row in a staggered arrangement.
- The effect of fin spacing on the air side performance for staggered arrangement also varies with the transverse tube pitch. For a smaller transverse tube pitch of 50 mm, there is no appreciable influence of the fin spacing on heat transfer. On the contrary, at a larger transverse tube pitch of 84 mm, one can see smaller fin spacing leads to lower heat transfer coefficients. This is also attributed to the presence of airflow pass effect.
- The change of fin pitch does not significantly affect the Colburn factor. The friction factor increases with increasing fin pitch when the Reynolds number (Re_{Dc}) is higher than 2500,
- The Colburn factor and the friction factor decrease with increasing number of tube rows when the Reynolds number (Re_{Dc}) is less than 4000,
- The Colburn factor and the friction factor are changed as the fin thickness is varied from 0.115mm to 0.250mm. However, the trends of the effects of fin pitch and number of tube rows on the Colburn factor and friction factor remain the same.
- This review paper is presented to provide designers innovative techniques of designing CHXs incorporating highly effective heat transfer performance, moderate pressure drops and less pumping power with low area to volume density (β) and cost-effective products. Followings are the important concluded remarks of this paper,
- Genetic Algorithm (GA) technique and ε-NTU model can play significant role in finding out most crucial optimized parameters for better designing of heat exchangers.
- Discrete, wavy and corrugated fins certainly yield higher transfer performance regardless of fin pattern, fin spacing, fin thickness and number of tubes compared to flat plate fins. However, very low fin spacing and Re number this superiority might get lost. Louver fins provide higher heat transfer rate as compared to plate fins, offset-strip fins and slit fins at same Re number. While for large number of tube rows slit fins are beneficial compared to louver fins. Convex louver fins are highly effective than louver and plane fins in terms of heat transfer rate and pressure loss. As they are capable of intensifying turbulent intensity of fluid flow.
- It has been reported by many investigators and researchers that influence of fin spacing is negligible on heat transfer and friction characteristics. While some authors have reported that only at higher Re number (~>1000), Fp has no effect on j and f factor and at lower Re number (<1000) j-factor decreases with the reduction of fin spacing.
- Staggered alignment as compared to inline alignment of tubes in CHEs leads to the higher heat transfer performance at the expense of larger pressure drop.
- It was reported by many investigators that number of tube rows has negligible impact on friction factor (f).
- Elliptical tubes provide lesser drag as compared to circular tubes which is the main reason that oval tubes yield better heat transfer characteristics. Position of tubes in downstream region instead of upstream region is preferred as heat transfer enhancement appears due to the significant horseshoe vortex formation.
- Fin thickness and waffle height along with the variation of other important perimeter like tube row number, fin spacing, alignment and pattern of fins and tubes etc. have significant influence on heat transfer performance of heat exchangers.

2.2: IMPORTANT RESULTS WHEN VORTEX GENERATORS ARE INCORPORATED:

A comprehensive review is performed to evaluate novel techniques of enhancing heat transfer performance and pressure drop reduction of compact heat exchangers through researcher's experimental and numerical work. The effectuality of vortex generators on reducing air side thermal resistance and improving thermodynamic features of heat exchangers is widely discussed. Followings are the most important concluding remarks of this review,

- **In** case of staggered tube banks, the heat transfer was augmented by 30% to 10%, with the winglet pairs of the present configuration, and yet the pressure loss was reduced by 55% to 34%, for the Reynolds number (based on two times the channel height) ranging from 350 to 2100.
- In case of in-line tube banks, the heat transfer was augmented by 20% to 10% together with the pressure loss reduction of 15% to 8% in the same range of Reynolds number above.
- Applying the configuration proposed by Fiebig et al. the heat transfer was augmented by 25% to 10% but the pressure loss was also increased by 35% to 20% in the same Reynolds number range.
- Fin-and-tube heat exchanger elements with fiat and round tubes of the same cross-sectional areas in staggered arrangements show that round tubes give a roughly 20% larger Nusselt number and a friction factor nearly four times that obtained with fiat tubes. Wing-type vortex generators increase fin heat transfer and pressure losses only marginally for round tubes.
- However, these vortex generators can dramatically increase fin heat transfer by a factor of 2 or more for flat tubes. Pressure loss also increases by a factor of 2. This keeps the flow loss of a fin-and-tube element with fiat tubes and VGs still 50% smaller than that for an element with round tubes. Fin heat transfer is more than double the heat transfer with round tubes. For this enhancement, only 2.6% of the fin surface needs to be modified to form vortex generators.
- Three-row tube bundle in an in-line arrangement without winglets, the heat transfer was augmented by 72% at the Reynolds number of 2700, and the pressure loss penalty increased by 210%, in comparison with the planar straight fin (channel) without tubes. The corresponding increases for the staggered arrangement were 95% and 310%, respectively.
- It was also found that the performance enhancement was higher for the in-line arrangement than for the staggered arrangement, because the pressure loss penalty for the staggered arrangement was considerably larger than for the in-line arrangement. By applying the winglets recommended by the previous studies, the heat transfer increased by 10 to 25%, and the pressure loss also increased by 20% to 35% for a three-row in-line tube bundle at Reynolds numbers (based on two times the channel height) ranging from 300 to 2700.
- Protrusion from heat transfer surfaces like vortex generators (VGs) play prominent role in enhancing heat transfer performance and design heat exchanger with even smaller area to volume density, low thermal resistance on both air and liquid side and finally increase the thermodynamic performance of heat exchangers.
- LVGs have ability to originate three heat transfer mechanisms such as, secondary flow formation, developing boundary layer and intensifying turbulent intensity of fluid flow. Reduction in boundary layer thickness is caused by main and corner vortices induced by longitudinal vortices of VGs.

- Poor heat transfer region or wake region behind the tubes can be significantly reduced by VGs. They are also capable of enhancing significant amount of heat transfer in downstream region.
- Careful location, size and attack angle of VGs is crucial to enhance significant amount of heat transfer performance regardless of CFU or CFD configuration at the penalty of moderate pressure drop. However, locations of VGs have negligible influence on pressure drop.
- Staggered arrangement of winglets is responsible for bringing more heat transfer rate than inline arrangement of winglets. Moreover, Delta winglet pairs (DWPs) are superior to Rectangular winglet pairs (RWPs) and also Winglets are preferable over wings in term of higher heat transfer rate and lower pressure drop.
- Longitudinal vortices induced by longitudinal vortex generators such as delta winglets are superior to transverse vortices produced by transverse vortex generators at the expense of same pressure drop.
- Different researchers proposed various VG's attack angles in their studies, however optimum attack angle lies in the range of 30^0-45^0.
- Flow visualization study play key role in visualizing flow pattern of fluid streamlines at microscopic level. The formation of horseshoe vortices, corner vortices, longitudinal vortices and transverse vortices induced by vortex generators can easily be analyzed and examined through this study.

CONCLUDED REMARKS:

This analytical study was performed to examine the influence of different crucial parameters on compact heat exchangers. The impact of tube shapes and tube arrangements were discussed. Three types of tube shapes were considered i.e. Oval, Circular and Square type and both inline and staggered arrangement were demonstrated. The effect of Delta winglet vortex generators (DWVGs) was also acknowledged by varying their attack angles (i.e. 30^0,45^0 and 50^0), number of row (one and three) and configurations (CFU and CFD). Followings are the key points of this study.

- Circular and oval tubes yield better performance as compared to square tubes in terms of heat transfer performance and pressure drop. As poor heat transfer zone is certainly reduced by circular and oval tubes. While square tubes do not impart quiet better role in eliminating wake regions.
- Heat transfer performance rises as the flow inlet velocity increases i.e. higher velocity flow leads towards higher turbulence intensity by increasing the strength of horseshoe vortices induced by tubes. However, this heat transfer augmentation comes with higher pressure drop.
- Staggered arrangement of tubes provides better heat transfer performance than that of inline arrangement of tubes. As staggered tubes tend to induce better flow pattern. However, the pressure drops provided by staggered tubes is higher than inline tubes.
- When vortex generators are used the performance of compact heat exchangers increases. As the attack angle of VGs increases heat transfer performance rises along with pressure drop. It was seen that optimum attack angle at witch heat transfer performance is higher with moderate pressure that was 30^0.
- Common flow down (CFD) configuration of VGs gives better performance than Common flow down (CFD) configuration. As longitudinal vortices induced by DWVGs with CFD configuration stay longer induces better flow mixing in downstream region of channels.

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