Forced Convection Studies with Different Geometries in Tube Bank Arrangement Using CFD

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ABSTRACT: Heat transfer to or from a bank (or bundles) of tubes in cross flow heat exchanger is relevant to numerous industrial applications; such as, steam generation in a boiler or air cooling in the coil of an air conditioner. The purpose of this research work is to find out the effect of horizontal pitch on the Nusselt number and friction factor of tube. The compact tube bank arrangement experiences high convective resistance on air side, due to low convective heat transfer coefficient. The arrangement of seven cylinders in triangular arrangement is made in such a way that maximizes the turbulence of air in the tube bank arrangement. The numerical analysis of this tube bank arrangement with Ansys fluent is carried out. This work consists of circular and diamond shape tubes, with different value of pitch for maximum heat transfer. The increase in turbulence will increase the convective heat transfer coefficient of air due to better mixing of air. The circular tube & diamond shape tube bank have maximum Nusselt no at horizontal pitch of 2.5D & 3D respectively. The value of coefficient of friction should be minimum so that there will be less pressure drop. The circular & diamond shape tube have least values of coefficient of friction at horizontal pitch of 3D

Key words: Convection, Geometry, CFD, Nusselt Number, HT

I. INTRODUCTION

The heat transfer in tube bank arrangement in cross flow heat exchanger has wide range of applications, such as steam generation in boiler or air cooling in the coil of an air conditioner. In energy transfer related applications, heat exchanger performance is of great importance in meeting today's stringent energy efficiency standards with low cost and less environmental impact. In the liquid-to-gas and phase-change heat exchangers, typical to many Heating, Ventilating and Air Conditioning & Refrigeration (HVAC&R) systems, the gas-side thermal resistance contributes heavily to the overall thermal resistance. Fin-and-tube heat exchangers have been widely applied in lots of fields, such as energy, power, chemical, food, and refrigeration. Improved heat transfer performance will have a significant impact on the energy crisis. For the gas-to-liquid fin-and tube heat exchanger, a high thermal resistance exists because of the poor thermo physical property of air. For example, the airside thermal resistance can comprise 75% of the total thermal resistance in an evaporator and 95% in a condenser for typical refrigeration systems. Researchers are devoted to developing the enhanced heat transfer surfaces, especially on the air side. Generally, in tube bank arrangement the fluid moves inside the tubes while second fluid moves outside the tubes. So, with the help of better mixing of fluids by providing different tube bank arrangement the heat transfer can be enhanced. The spacing between the rows of tubes have significant effect on the heat transfer and skin friction coefficient of tubes. The prime motive of this work is to find out the effect of horizontal pitch on the convective heat transfer coefficient as well as on the skin friction coefficient. So the development of the higher effective heat exchanger requires a larger contact area for heat transfer with reduced volume. Such larger contact area is easily achieved by passing one fluid through a number of small channels or tubes. This process with an array of the cylinder leads towards the development of a new class of heat exchanger, as tube banks. The tube bank is a special case of the heat exchanger, where the heat interaction is between the hot fluids flowing through a number of small tubes and cold fluid usually passes over the tube surface along the cross direction. There are several techniques commonly used to increase the heat transfer rate by improving thermal contact between the heat exchanger fluid and wall. Themost common methods typically manipulate the surface, including its roughness, use of coiled tubes and vortex generators.

II. LITERATURE REVIEW

Smith et al[2] have investigated experimentally the effect of Reynolds number on the nusselt number and skin friction coefficient in tandem arrangement with uniform heat flux over the tubes. The results presented by the author suggest that Nusselt number have strong dependency on cone angles, tail length ratio and Reynolds number. The heat transfer and friction factor will increase with cone angle

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and reduction with tail length ratio due to augmentation in turbulence. The heat transfer enhancement of 67%, 57% and 46% were obtained at the cone angle of 45 degree with tail length ratio of 1, 1.5 and 2.0 respectively.

Pongjet et al [3] investigated experimentally the heat transfer and friction factor in combined twisted tape & winglet vortex generator for constant heat flux square duct. The author concluded that the nusselt number & friction factor for the combined twisted tape and V- winglet increases with increasing duct to height ratios & decreasing the pitch to tape width ratio. The maximum value of heat transfer & friction factor is obtained at duct height ratio of 0.2, pitch to tape width ratio of 2 and twist ratio of 4. The maximum thermal performance is obtained at duct height ratio of 0.1, pitch to tape width ratio of 2 and twist ratio of 4. The thermal performance of combined vortex flow deice gives 17% higher than twisted tape alone.

Bayat et al[4] had experimentally studied the fluid flow and heat transfer from cam-shaped tube bank in staggered arrangement. Tubes were located in test section of an open loop wind tunnel with two longitudinal pitch ratios 1.5 and 2. Reynolds number varies in range of 27,000 Re_D 42,500 and tubes surface temperature is betweenn78 and 85 ^{0}C. Results show that both drag coefficient and Nusselt number depends on position of tube in tube bank and Reynolds number. Tubes in the first column have maximum value of drag coefficient, while its Nusselt number is minimum compared to other tubes in tube bank. Moreover, pressure drop from this tube bank is about 92–93% lower than circular tube bank and as a result thermal–hydraulic performance of this tube bank is about 6 times greater than circular tube bank.

Zdanski et al[5] had performed an experimental study addressing the effects of delta winglet vortex generators on convective heat transfer rate at in-line tube bank in external cross flow. The main goal of the work is to evaluate a study assessing the influences of the following parameters on convective heat transfer enhancement: the distance from the vortex generators to the tube bank, the pitch and the incidence angle of the delta winglet, as well as the free-stream velocity inside the wind tunnel (Reynolds number). The validation of the experimental methodology employed was performed by comparing the present results with empiric correlations available in the literature. The main results indicate that Nusselt number was enhanced when using turbulence promoters, being the maximum increment around 30%. Otherwise, the pressure drop through the tube bank was enhanced accordingly, being the maximum increment around 40%. Finally, it is worth mentioning that all the experimental results for the Nusselt number were condensed in a new empirical correlation for practical applications with good accuracy (maximum error around ±6.0%).

The main aim of this paper is-

1. To study the effect of Reynolds number on nusselt number & skin friction coefficient for circular shaped tube with triangular arrangement by CFD analysis.
2. To study the effect of Reynolds number on nusselt number & skin friction coefficient for diamond shaped tube with triangular arrangement by CFD analysis.
3. To compare the effect of shape of tube on nusselt number and skin friction coefficient

III. METHODOLOGY

CIRCULAR SHAPE TRIANGULAR TUBE BANK ARRANGEMENT

In this paper the analysis is done on circular shape tubes and diamond shape tubes. If we consider the cylinder than we have a variation of pressure with increase in length. For the analysis of cylinder, the free stream fluid is brought to rest at forward stagnation point, with rise in pressure as per Bernoulli equation. From stagnation point the pressure starts decreasing with the motion of fluid in forward direction. So, in this case the boundary layer develops under the influence of favorable pressure gradient (dp/dx<0) and reaches to the minimum value[1]. After the point of minimum pressure towards the rear end further development of hydrodynamic boundary layer takes place under the influence of adverse pressure gradient (dp/dx>0). The adverse pressure gradient will promote the turbulence which will enhance the heat transfer due to better mixing.

IV. ANALYSIS OF RESULT

The assumption is made that the flow is taking place from left hand side to right hand side and flow through the circular tube. The inlet velocity is varied with the desired Reynolds number. As from the contour we can see that there will be increase in velocity between the circular tubes simultaneously there will be decrease in pressure between the tubes. The increase in velocity will increase the turbulence in the flow region with the large number of eddies formation.

Comparison of velocity contour of 4m/s at different pitch

At longitudinal pitch of 3D

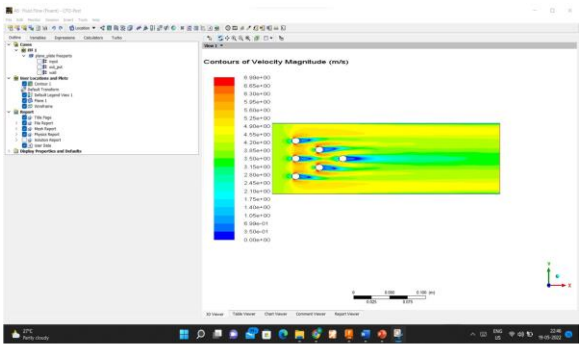

Figure 1Comparison of velocity contour of 4m/s at pitch 3D

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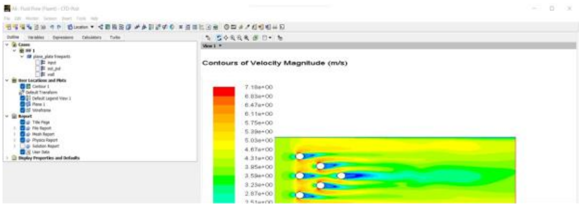

Figure 2 Comparison of velocity contour of 4m/s at pitch 2.5D

At longitudinal pitch of 2D

The velocity of a fluid is varied from 4 m/s, 6 m/s, 8 m/s, 10 m/s and 11 m/s with temperature of surrounding is considered as 300 K. the assumption is that air flows from left hand side to the right-hand side and pass over the triangular tube bank arrangement. The increase in velocity is observed in between the tubes because of decrease in area between the tubes simultaneously there will be decrease in the pressure between the tubes as per the Bernoulli equation.

The main objective of the present study is to determine the effects shape of the tube and their arrangements on the flow structure and the overall heat transfer escalation for a fin-tube type heat exchanger. Thus, investigations with two different shape of fin-tube heat exchanger, one with circular shape of tube and the other with diamond shape of tubes along with triangular arrangement under different values of longitudinal pitch have been conducted. The longitudinal pitch for circular tubes are varied as 2D, 2.5D and 3D. The present study has been carried out with the objective of assessing the effects of the positions of the tubes on flow and heat transfer characteristics in a representative periodic element of a fin-tube heat exchanger. The flow structures and heat transfer behaviors have been examined at different cross-flow planes with the help of velocity vectors, streamlines, temperature contours and Nusselt number distribution etc. The performance of the effect of tube shape has been evaluated based on the comparison of Nusselt number and friction factor.

In the case of heat transfer, understanding the distribution of the temperature of air close to the tubes and development of the vortex within the column of tubes is studied. The distribution of the temperature along the test array, determines the optimum spacing of the tubes within the array. Additionally, the temperature contour provides an estimate of the thermal boundary layer formation over the tube surface.

The temperature of the air in the cross flow across the tube bank increases, by gaining the heat from the tube surface. The maximum temperature of 356 K, which is the boundary condition, is observed at the wall boundary of the tubes. The intensity of the rise in the temperature of the air increases as the air past the tube column. With the increase in the Reynolds number for the fluid flow, temperature intensity surrounding the circular and diamond tubes gradually decreases. The domain wall on above and below the tube array has no impact on the temperature distribution over the tube surface.

V. OPTIMIZATION FOR CIRCULAR SHAPE TUBE

As we can see the horizontal pitch of 2.5D will provide the highest value of nusselt number so the value of nusselt number increases by increasing the value of horizontal pitch from 2D due to increase in turbulence which would result in large inter-mixing and hence better heat transfer. As expected in the staggered case, there is very strong interactions between the vortices. Moreover, both clockwise and counter clockwise vortices are interacting thus the temperature is more dispersed. The same can be visualized from the temperature contours of different configurations in different longitudinal pitch. It may be noted that the average temperature at the exit of the tube bank arrangement cross flow heat exchanger with transverse pitch of 2.5D is the highest due to better intermixing.

During the channel flows, the swirling motions generated by the staggered tube bank arrangement disrupt the thermal boundary layer, intensify mixing and bring about enhancement of heat transfer with relatively less pressure penalty.

The fin surfaces are subjected to flow impingement; accelerated and decelerated flow due to the presence of turbulence. All these effects cause differential local distribution of heat transfer on the top and bottom channel walls.

The flow past over the tubes generates the turbulence in the fluid flow, across the tube bank. The intensity of the turbulence increases with the increase in the Reynolds number for the fluid flow. Further, the array of the tubes increases the intensity of the turbulence much higher. The turbulence has direct impact on rate of heat transfer rate of the system.

It is seen that the intensity of the turbulent kinetic energy increase with the increase in the Reynolds number for the fluid flow. The maximum intensity of the turbulence is observed at the downstream of the last column of the tube bank. This may be due to the fact that the vortex shedding phenomenon does not take place in the last column of the tube bank. The higher intensity vortex increases the turbulence in the fluid flow. The high turbulence is observed after the separation point, which may be due to the fluid recirculation within the tube bank, thereby promoting better energy interaction. This may also result in the higher pressure drop within the tube bank.

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The fluid flow is always associated with the friction coefficient over which the flow is established. It is clear from the figure that the friction factor is maximum in case of the circular tubes. The highest friction factor is reported for the circular tubes with a pitch ratio of 2. The friction factor for the fluid decreases, with an increase in the Reynolds number for the flow. This is usually due to the dominant pressure force, which reduces the friction.

The minimum value of skin friction coefficient will be the $186.36*10^{-4}$ at longitudinal pitch ratio of 3 at Reynolds number of 6477 for diamond shape triangular tube bank arrangement. The value of nusselt number will be maximum for longitudinal pitch ratio of 3 for diamond shaped tube.

There will be around 35% decrease in skin friction coefficient from circular tube to diamond shape tube triangular tube bank

VI. CONCLUSION

1. There will be around 35% decrease in skin friction coefficient from circular tube to diamond shape tube triangular tube bank arrangement.
2. The value of nusselt number is maximum for longitudinal pitch of 2.5D for all the values of respective Reynolds numbers for circular shape triangular tube bank arrangement
3. The value of skin friction coefficient is maximum for longitudinal pitch of 2.5D for all the values of respective Reynolds numbers for circular shape triangular tube bank arrangement
4. In case of diamond shape tube initially the value of nusselt is maximum for longitudinal pitch of 2D up to the Reynolds number of 12000 but after that the longitudinal pitch of 3D will provide maximum values of nusselt number.
5. In case of diamond shape tubes the minimum value of skin friction coefficient is obtained with the longitudinal pitch of 2.5D

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