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Analysis of Effect of Longitudinal Pitch in a Staggered Tube Array of Cross Flow

H S Bachhav¹, P J Bansod²

M. Tech, Department of Mechanical Engineering, G.H Raisoni college of Engg& Management, Pune, India¹

Professor, Department of Mechanical Engineering, G.H Raisoni college of Engg& Management, Pune, India²

ABSTRACT: Heat transfer in a tube array of cross flow heat exchanger is familiar to number of industrial applications like steam generation in a boiler, air cooling in the coil of an A/C. 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 due to low convective heat transfer coefficient experiences high convective resistance on air side. The arrangement of tubes in staggered is made in such a way that to increase the turbulence of air in the tube bank. The analysis of this tube array arrangement is carried out with ansys fluent. This work consists an array of tubes of longitudinal pitch (at 2d, 2.5d & 3d). The increase in turbulence due to better mixing of air will increase the convective heat transfer coefficient of air. The tube array has maximum nusselt number at horizontal pitch of 2.5d. The value of coefficient of friction should be minimum so that there will be less pressure drop. The tube has least values of coefficient of friction at horizontal pitch of 3d.

KEYWORDS: Heat Exchanger, CFD analysis, Inline and staggered tube, tube bank, heat transfer.

I. INTRODUCTION

Heat exchangers consisting of a heated tube bank are widely used in practice to transfer heat from one medium to another. Typically, one fluid flows over the tube bank, while another fluid of a different temperature passes through the tubes. The tubes are arranged as a bank in an inline or staggered manner to heat the surroundings, dissipate excessive heat, or recover waste heat. Tube banks in crossflow are used in many fields, such as industry, biomedicine, mechanisms for heat transfer increase, and nano fluid applications. They can be found in the economizer and evaporator of an air conditioning system [1].

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 has significant effect on the heat transfer and skin friction coefficient of tubes. The prime motive of this project 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. The most common methods typically manipulate the surface, including its roughness, use of coiled tubes and vortex generators.

Tube banks are commonly-employed design elements in heat exchangers. Both plain and finned tube banks are widely found. Tube bundles are a sub-component in shell and tube heat exchangers, where the flow resembles crossflow at some places, and longitudinal flow elsewhere. (The term tube bank is often used, in the literature, to denote a crossflow situation and bundle to indicate longitudinal flow, however this convention is far from universal.) The flow may be single-phase or multiphase: boilers and condensers containing tube banks find a wide range of applications in industry, in addition there may be combustion, e.g., in a furnace heat exchanger.

II. COMPUTATIONAL FLUID DYNAMICS (CFD)

CFD is a highly useful technique used to numerically solve the equations of the fluid systems involving heat transfer, mass transfer, concentration, and fluid flow depending on the results of computer-based simulation. This tool can be used to cover a wide range of industrial and non-industrial applications including: lift and drag in aerodynamic field, combustion process in gas turbine and internal combustion engines, the flow inside rotating tubes and diffusers and other applications in marine engineering, chemical processes, environmental engineering, etc.

III. LITERATURE REVIEW

L.H. Tang, M. Zeng, Q.W. Wang [2] in his investigation characteristics of air-side heat transfer and friction of five types of fin and tube heat exchangers like crimped spiral fin, slit fin, plain fin, fin with delta wing longitudinal vortex generators and mixed fin, with the 12 number of rows and the diameter of tubes is 18 mm, having Reynolds number range from 4×10^3 – 10×10^3 . In this investigation it was found that crimped spiral fin gives the better result for heat transfer compared to other fins. P Pongsoi, Chi-Chuan Wang, S Pikulakajorn, S Wongwises [3] in his paper the effect of fin pitches with a multi pass parallel and counter cross-flow configuration on the air side performance of crimped type spiral fin tube in heat exchangers. This study examines the effect of fin material, fin pitches on the air coming side on performance of crimped fin tube with the range of high Re numbers varying from 4×10^3 to 13×10^3 . P Pongsoi, S Pikulakajorn, Chi-Chuan Wang, S Wongwises [4] in his paper covers the effect on the air side performance of number of tube rows with a multi pass parallel and counter cross flow configuration for crimped spiral fin tube heat exchanger. The test samples materials are of aluminium and copper with separate number of tube rows. The effects on the friction characteristics and heat transfer of number of tube rows and fin material are studied. The results show that there is insignificant effect on the heat transfer performance for number of tube rows and fin materials is found at high Reynolds number. P Pongsoi, P Promopattum, S Pikulakajorn, S Wongwises [5] reports on Effect on the air side performance of fin pitches of L footed spiral fin tube heat exchangers. The purpose of this experiment was to investigate the effects on the frictional characteristics and air side heat transfer performance of fin pitch of L-footed spiral fin and tube heat exchangers with Re number varying from 4×10^3 to 15×10^3 . A working fluid on the air side and the tube side are ambient air and hot water used, respectively. However, there is no such significant effect of fin pitch on the average pressure drop, heat transfer rate, and friction factor according to this investigation. Ashish Dixit & Anil Kumar Patel [6] conduct an experimental investigation of a plate fin on heat transfer characteristics having two broad faces grooves of different configurations. Transverse grooved, V-grooved, multi V-grooved, inclined grooved fins are used in analysis. They gathered heat transfer data by changing Reynolds number from 1.5×10^3 to 5×10^3 , and these outcomes are compared with a smooth conventional fin to gauge the heat transfer performance of modified fin. Ratnesh Raj [7] conduct experiment and measured pressure drop characteristics and heat transfer in doubly enhanced tube heat exchanger, which shows for the analysis the doubly enhanced tube is used which compared to the bare tube heat exchanger. The maximum reduction in pumping power on a constant heat duty basis was in the laminar regime 23% with Ethylene Glycol and in the turbulent regime 54% with water.

IV. METHODOLOGY

A. Aim and Objective:

The objectives of the work have been as the following:

- To study the effect of Reynolds number on heat transfer for staggered tube array by CFD analysis.

- To study & compare the effect of longitudinal pitch on heat transfer for staggered tube array.
- The outcome results are helpful for the modification and redesign for the improvement in the performance of inline & staggered tube array arrangement.

B. Governing Equations:

The governing equations of a mathematical model describe how the values of the unknown variables (i.e. the dependent variables) change when one or more of the known (i.e. independent) variables change. The governing equations of fluid flow represent mathematical statements of the conservation laws of physics. The mass conservation or continuity equation is given by:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$$

Conservation of Momentum Equations:

X-momentum:

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right)$$

Y-momentum:

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right)$$

Z-momentum:

$$u \frac{\partial \omega}{\partial x} + v \frac{\partial \omega}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left(\frac{\partial^2 \omega}{\partial x^2} + \frac{\partial^2 \omega}{\partial y^2} \right)$$

Energy Equation:

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)$$

C. Assumptions:

Numerical model was developed under the following assumptions

- The physical properties of the working fluid and the tube are constant.
- The incompressible flow is limited as laminar flow.
- The flow and heat transfer are in steady state.
- The viscous dissipation is negligible.

For the expected results, the governing equations describing the fluid flow and heat transfer in domain should be in non-dimensional form. The tube bank consists of tube arranged subjected to cross flow conditions. The case is having an inline tube array and staggered tube array. The various steps that are involved in project completion like selection of model, CFD analysis of the tube array arrangement are involved. The steps that are followed to complete this dissertation and writing work of dissertation is explained in a sequence manner and represented by flow chart.

D. Geometry Modelling for Staggered Tube Array:

Staggered tube array model was made using ANSYS design modeler software. For model, a 60 mm entry section was added to ensure a fully developed velocity profile to maintain a steady analysis, and a 300 mm section after the tubes was also added to study the flow in that region and its effect on the results for tube array model, as shown in Figure 1 tube diameter (d) of 15 mm was used with longitudinal pitch $SL = 2d$ with fixed transverse pitch $ST = 30$ mm. The analysis on staggered tube arrangement is for different values of horizontal pitch like $2d$, $2.5d$ & $3d$

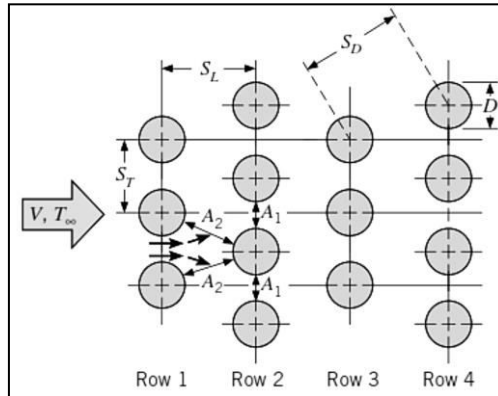


Figure 1: Tube array arrangements Staggered.

E. Mesh Generation for Staggered Tube Array:

A mesh is generated over the staggered tube array arrangement and the domain, also 10 inflation layer on tube surface. After mesh generation the mesh is solved numerically with CFD solution. Figure 8 shows schematic drawing of the present computational domain.

F. Boundary Condition:

The working fluid material is air and it is passing through the domain having the velocity from 4 m/s, 6 m/s, 8 m/s, 10 m/s and 11 m/s. The pressure based solver is used for the simulation with SST K-ε model and flow is second order upwind. The tube surface is considered as no slip boundary condition and domain boundaries are considered as slip condition. The temperature of air at inlet is taken as 300 K and the temperature of tube is taken as 400 K. The simulation process consists of 4 steps geometry, modelling, mesh generation, CFD solution and CFD analysis of the result.

V. RESULT AND DISCUSSION

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 increase in velocity is observed in between the tubes because of decrease in area between the tubes will be resulted in the pressure drop as per the Bernoulli equation. The main objective of the present study is to determine the effects of longitudinal pitch arrangements on the flow structure and the overall heat transfer in heat exchanger. Thus investigations with staggered tube arrangement under different values of longitudinal pitch have been conducted. The longitudinal pitches for tubes are varied as 2d, 2.5d and 3d. The flow structures and heat transfer behaviours 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 longitudinal pitch has been evaluated based on the comparison of Nusselt number and friction factor.

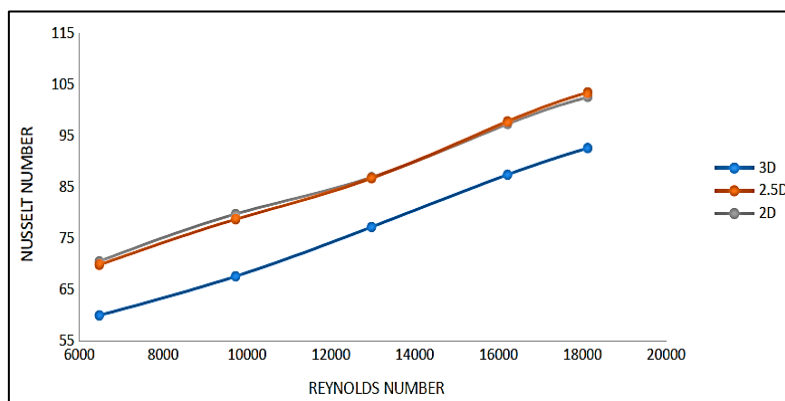


Figure 2: Comparison of Nusselt number vs. Reynolds number.

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 tubes gradually decreases. The domain wall on above and below the tube array has no impact on the temperature distribution over the tube surface.

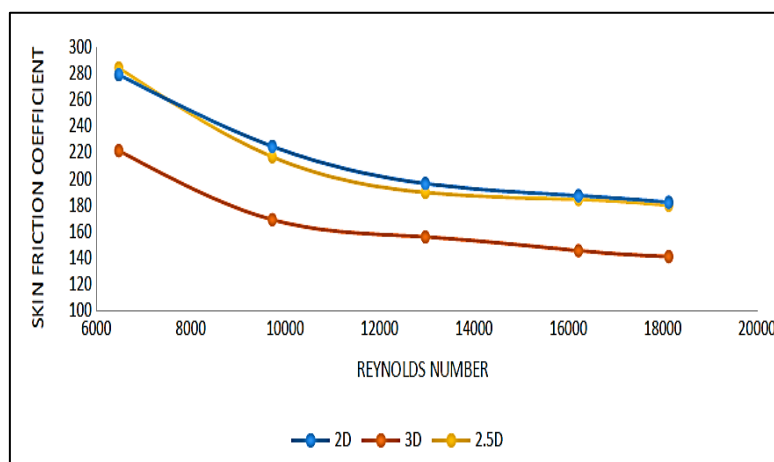


Figure 3: Comparison of skin friction coefficient Vs. Reynolds number.

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 are very strong interactions between the vortices. All over, 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.

VI. CONCLUSION

The determination of cross flow heat transfer to air from staggered tube array was achieved. The relationship between the flow rate with longitudinal pitch and their effect to the heat transfer rate were determined as followed:

- The increase in the pitch ratio of the tubes may increase the possibility of higher vortex formation, which may enhance the heat transfer.
- For the heat exchangers with staggered tube bank arrangement, the performance of the heat exchangers increases with increase in longitudinal pitch from 2d to 2.5d although with a pressure penalty. This is irrespective of the Reynolds number applicable for general purpose industrial heat exchangers (Re number varying from 6×10^3 to 19×10^3).
- There will be increment in the value of nusselt number with increase in horizontal pitch from 2d to 2.5d due to the horizontal pitch of 2.5d will provide the better turbulence with the inter mixing of fluid as compared to the other horizontal pitch. The value of nusselt number is maximum for longitudinal pitch value of 2.5d for all the values of respective Reynolds numbers for the staggered tube array arrangement.
- The value of skin friction coefficient should be minimum so that there will be less pressure drop. The tube has least values of coefficient of friction at horizontal pitch of 3d.
- All the combinations are liable to pressure drop penalty



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