Flow and thermal characteristics analysis of plate-finned tube and annular-finned tube heat exchangers for in-line and staggered configurations

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Abstract- *As the effective selection of fin can greatly enhance the performance of heat exchanger, heat transfer and pressure drop performance on the air-side of annular and rectangular finned tube heat exchangers were numerically investigated. Two types of tube arrangement (in-line and staggered alignment) were examined for 6 different air flow rate for both the heat exchangers using computational fluid dynamics software package ANSYS FLUENT. Renormalization group theory (RNG) based k-ε turbulence model was employed to handle the unsteady three-dimensional flow and the conjugate heat transfer characteristics. The exit temperature were determined from the simulated results and then the LMTD, heat transfer rate and air-side heat transfer coefficient were calculated. The numerical flow visualization results revealed few important aspects, such as, the development boundary layers between the fins, the formation of the horseshoe vortex system, and the local variations of the velocity and temperature on the fin geometries. The result shows that as the air flow rate increased the exit temperature decreased but the overall heat transfer increased. Staggered configuration shows higher heat transfer characteristics over the in-line configuration. The rectangular finned tube shows 17 to 24% improvement in heat transfer over the annular finned tube.*

Keywords: Finned tube heat exchanger, Tube arrangement, Exit temperature, Heat transfer coefficient

1. INTRODUCTION

Heat exchangers are the common way of cooling or heating of fluids in the field of engineering. Among the different type of heat exchangers, tubular or shell and tube type and extended surface or finned tube type heat exchangers are widely in use and finned tube heat exchangers are commonly used in many industrial applications such air conditioning system, radiators and industrial heater. Generally, a liquid flows within the tubes while gas is directed across the finned-tubes. For poor thermal conductivity and heat transfer coefficient of the gasses, it is necessary to apply the extended surfaces on the gas side of the heat exchangers to enhance the heat transfer without losing its compactness. In addition, heat transfer rate in the finned tube heat exchangers largely depends on the geometry of the fin, flow characteristics, and the alignment of the tubes and fins [reference]. Moreover, flow patterns for the in-line and staggered tube arrays are complicated and will have to be dealt in the heat exchanger design. Thus, further investigations on the heat transfer enhancement are of broad interest to design more compact and efficient heat exchangers. Until this day, a large number of work has been performed to improve the heat transfer rate in the finned-tube heat exchangers.

Sin and Kim [1] investigated the effect of fin and tube alignment on the heat transfer performance of finned tube heat exchangers with large fin pitch. The study revealed that the Colburn *j*-factor decreases with an increase of number of tube rows from 1 to 4 at a given fin pitch and the effect of number of rows is significant at low Reynolds number. They found that by applying staggered fin alignment, the heat transfer performance increases by 7% as compared with the in-lined alignment. Moreover, continuous flat plate finned-tube shows lower heat transfer performance than the discrete flat plate finned-tube. By applying staggered fin and tube alignments, heat transfer performance can be improved by 20% compared to the continuous flat plate finned-tube [1]. Mon and Gross [2] numerically analyzed the fin-spacing effects in annular-finned tube heat exchangers. The study found that the boundary layer and horseshoe vortices between the fins were substantially dependent on the Reynolds number and the ratio of fin pitch to height. The horseshoe vortex effect is more obvious in the largest fin spacing and at the higher velocities. The boundary layer development on the fin and tube surfaces mainly depends on the fin spacing to height ratio (s/h_f). Choi et al. [3] investigated the heat transfer characteristics of discrete plate finned-tube heat exchangers with large fin pitches. The study concluded that the discrete plate finned-tube heat exchanger shows higher j-factors than the continuous plate finned-tube heat exchanger due to the boundary layer break and higher flow mixing intensity. For fin pitches of 7.5–15 mm, the j-factors of the discrete plate finned-tube heat exchangers were 6.0–11.6% higher than those of the continuous plate finned-tube heat exchangers. The j-factor of the discrete plate finned-tube heat exchangers decreased with the increase of the number of tube rows. Rao, Saxena and Kirar [4] studied elliptical pin fin heat sink by taking minor axis as the perameter. It found that the thermal resistance of pin fin decreases but the pressure drop increases with the increase in minor axis. Joardar and Jacobi [5] investigated the potential of winglet type vortex generator (VG) arrays for air-side heat transfer enhancement. They studied two enhanced cases: single row leading tube referred as ‘‘leading-edge’’ and three-row-alternate-tube inline array arrangement referred as ‘‘three-row-array’’. The thermal performance of the heat exchangers after attaching the VGs for both cases are compared to its baseline performance without the generators. For the leading-edge case, the vortex generators caused an increase of 11.7–32.7% in heat transfer coefficient over the range of face velocities considered. In contrast, the three-row-array resulted in a more uniform enhancement of about 38% over the baseline case for the tested range of airside Reynolds number. Lee et al. [6] studied the heat transfer characteristics of spirally-coiled circular fin-tube heat exchangers operating under frosting conditions. The study revealed that the frost thickness for large fin pitches was greater than that for small fin pitches. The air-side pressure drop per unit length increased as the fin pitch decreased. Heat transfer rate of the heat exchanger increased as the fin pitch decreased and the number of tube rows increased. The Nusselt number of the heat exchangers decreased with the elapsed time for frost growth. Bahirat and Joshi [7] analyzed the plate fin tube heat exchangers for various fin inclinations. Lin et al. [8] numerically investigated the heat transfer enhancement of circular tube bank fin heat exchanger with interrupted annular groove fin. The study found that the interrupted annular grooves have dual efficacy of fluid flow guiding and detached eddy inhibition to reduce the size of wake region, and could continuously produce the developing boundary layer. At lower Reynolds numbers, the interrupted annular groove fin surface could not efficiently enhance heat transfer but the excellent performance of the interrupted annular groove fin can be achieved at higher Reynolds numbers. Mendez et al. [9] investigated the effect of fin spacing on convection in a plate fin and tube heat exchanger. They studied the hydrodynamics and heat convection around a cylinder between flat plates representing a single-row plate-fin and tube heat exchanger. The pair of flat plates affect each other for small spacing and the nature of the flow strongly changes as the distance between fins is increased. Upstream of the tube, there is no vortex at first as the systems due to the two fins cancel themselves out. The vortices showed up as the fin spacing is further increased. The region downstream of the tube is dominated by the wake. The local Nusselt number on the fin is highest at the leading edge due to the thin boundary layer, and at the front of the tube when a horseshoe vortex system is present there.

The main purpose of this study is to numerically simulate the annular finned tube and staggered finned tube heat exchanger in six different air flow rate at constant tube wall temperature using the ANSYS FLUENT software. Then analyze their flow characteristics and the temperature distribution on a four row bundles numerically for both inline and staggered configuration of annular-finned tube and rectangular-finned tube and calculate the heat transfer output for all the cases and compare their characteristics.

2. NUMERICAL SIMULATION

The available computational fluid dynamics software package ANSYS FLUENT is used to determine the related problems. ANSYS FLUENT uses a finite volume method to solve the governing equations for a fluid which are derived from the conservation mass equation, the conservation of momentum and the conservation of energy equations.

Equation for conservation of mass or continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = S_m \qquad (1)$$

Equation 1 is the general form of the mass conservation equation and is valid for incompressible as well as compressible flows. Where ρ is the density and $\vec{V}$ is the velocity vector. And the source S_m is the mass added to the continuous phase from the dispersed second phase for example, due to vaporization of liquid droplets and any user-defined sources.

Equation for conservation of momentum or Navier-Stokes equation:

$$\frac{\partial}{\partial t}(\rho \vec{V}) + \nabla.(\rho \vec{V}\vec{V}) = -\nabla p + \nabla.(\vec{\tau}) + \rho \vec{g} + \vec{F} \qquad (2)$$

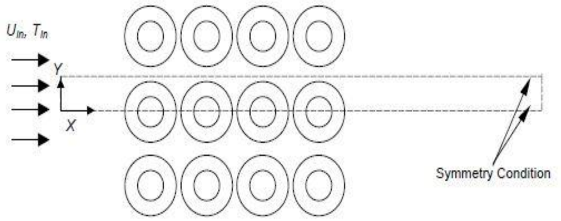
Where, p is the static pressure, $\vec{\tau}$ is the stress tensor, and $\rho\vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces (for example, that arise from interaction with the dispersed phase), respectively. $\vec{F}$ also contains other model-dependent source terms such as porous-media and user-defined sources.

Equation for conservation of energy or energy equation:

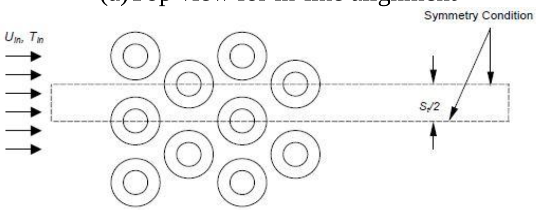
$$\frac{\partial}{\partial t}(\rho E) + \nabla.(\vec{V}(\rho E + p)) = \nabla.(k_{eff}\nabla T - \sum_j h_j\vec{J_j} + \vec{\tau}_{eff}.\vec{V} + S_h \quad (3)$$

Where, k_{eff} is the effective conductivity and $\vec{J_j}$ is the diffusion flux of species j. The first three terms on the right-hand side of 3 represent energy transfer due to conduction, species diffusion, and viscous dissipation, respectively. S_h the heat of chemical reaction, and any other volumetric heat sources.

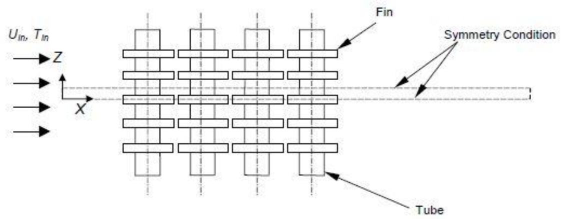
The geometry of the fin and tube is shown in Figure.1. A schematic view of the proposed in-line and staggered tube bundles for annular fin from the top position are shown in Figure 1(a) and 1(b). Computational domains to be considered in this study are displayed by dotted lines with symmetry conditions.



(a)Top view for in-line alignment



(b)Top view for staggered alignment



(c)Side view

Figure.1: Computational domain

Table.1: Dimensions for the geometry

	In-line alignment (cm)	Staggered alignment (cm)
Tube diameter	1.5	1.5
Diameter of the fin [For annular fin]	2	2
Length and width of the fin [For rectangular fin]	2	2
Fin thickness	0.075	0.075
Transverse tube pitch	5	5
Longitudinal tube pitch	5	2.5

For this study path conforming method with some body and face sizing is used to discrete the computational domain into a finite number of control volumes. The geometries of computational domains are carefully modeled and meshed within the ANSYS software. The mesh sizes are made finer near the fin and tube wall to resolve the secondary flows such as horseshoe vortices and flow separations where the high gradients are expected. The coarse mesh sizes were selected for the case where the flow is relatively uniform.

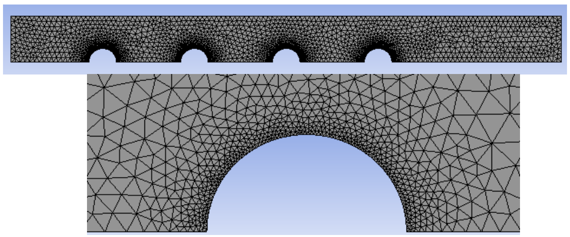


Figure.2: Mesh

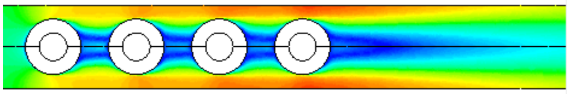
In this study, the air is forced to pass between the fins, which gets heated or cooled. And the important physical properties such as thermal conductivity (0.02808 W/m.K), density (1.204 kg/m^3), viscosity (2.008*10^{-05}), specific heat (1007 kJ/kg.K) is assumed. However, in this study, it is assumed to be the dry air and no attempts were prepared for condensation effects. For the solid's part, aluminum is chosen for both the tube and the fin.

To solve any numerical problem some preliminary conditions of the physical model have to be defined appropriately. For this problem at the upstream boundary conditions, the air entering the computational domain is assumed to have uniform velocity U_{in}, temperature T_{in} (293.15 K). The fluid region consists of the inlet, outlet, and symmetry zone. The solid region includes the fin. At the solid surfaces or wall, no-slip conditions for the velocity are specified. Heat convection from the fin and tube is considered. And the wall is assumed to have a uniform temperature of T_{wall} (393.15 K).

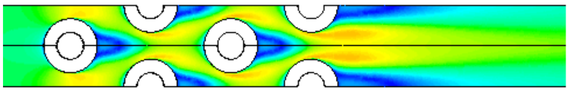
Numerical investigations are performed for both the staggered and the inline tube arrangement for both annular finned tube and rectangular finned tube heat exchangers. All simulations are carried out for a range of air flow rate 4.5 ms^{-1} to 9.5 ms^{-1}. Renormalization group theory (RNG) based k–ε turbulence model with is employed to predict the heat transfer and fluid flow characteristics.

3. RESULTS and DISCUSSION

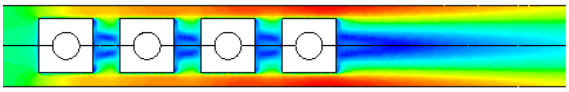
It is essential to understand the characteristics of velocity distribution for understanding of heat transfer phenomena. Velocity field between the cross section mid plane between two adjacent fin is shown in Figure.3. And it is seen that the flow pattern for the first row is almost same for both in-line and staggered alignment. In the staggered arrangement the first row acts as a turbulence promoter as well as the velocity is increased for the further rows because of its blockage effect. For the in-line alignment there much larger wake region present after the first row than staggered alignment. From Figure.4 similar kind of vortex formation near the fin can be seen in the first row for both circular and rectangular fin in bot in-line and staggered alignment. But for in-line case as the deeper rows present in the wake rezone no vortex can be seen after the first row. But for the staggered alignment vortex can be evident in each rows. Vortex generation keeps a vital role in heat transfer.



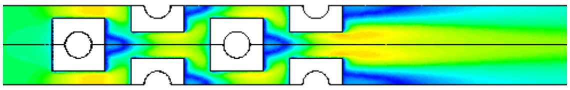
(a) Annular finned tube [In-line alignment]



(b) Annular finned tube [Staggered alignment]



(c) Rectangular finned tube [In-line alignment]



(d) Rectangular finned tube [Staggered alignment]

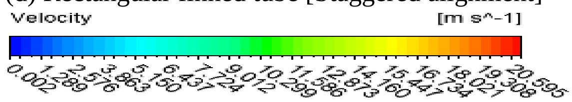
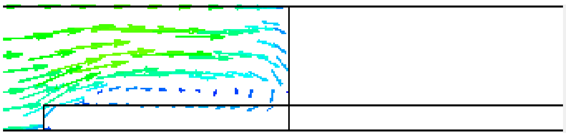
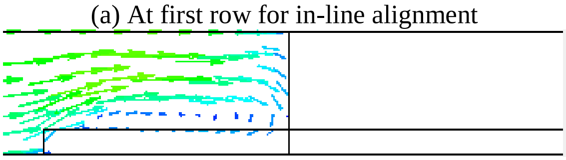


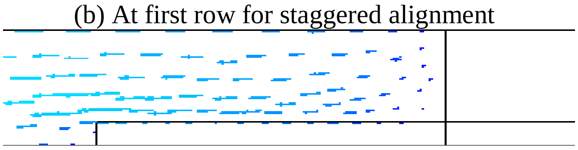
Figure.3: Velocity contour



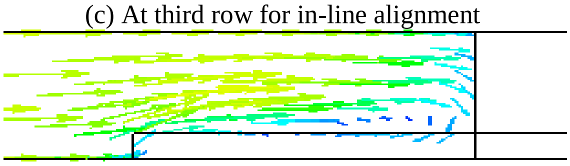
(a) At first row for in-line alignment



(b) At first row for staggered alignment



(c) At third row for in-line alignment



(d) At third row for staggered alignment

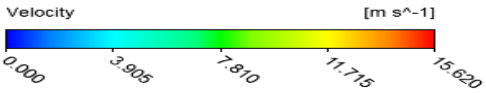


Figure.4: Velocity vector

Figure.5 and Figure.6 shows the heat transfer characteristic for the heat exchangers at different air flow rate and Figure.7 shows the temperature distribution at the mid fin plane. It is evident that as the velocity increases the exit temperature reduces but overall heat exchanging characteristics increases almost linearly. For high air flow rate the exit temperature may reduce but overall mass transfer increases which cover up the effect of lower exit temperature's effect in overall heat transfer. It is also evident that the staggered alignment shows higher heat transfer characteristic than the in-line alignment. Formation of more vortex and higher turbulence in the flow plays a vital role for the higher heat transfer in the staggered alignment. And the rectangular fins provide higher heat transfer that the circular fins. This may happen due to the higher heat transfer surface area in rectangular fins.

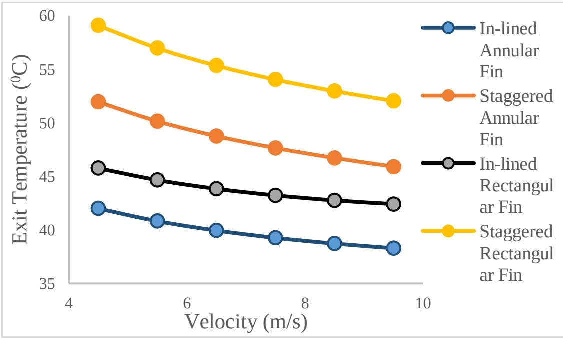


Figure.5: Exit temperature vs Velocity

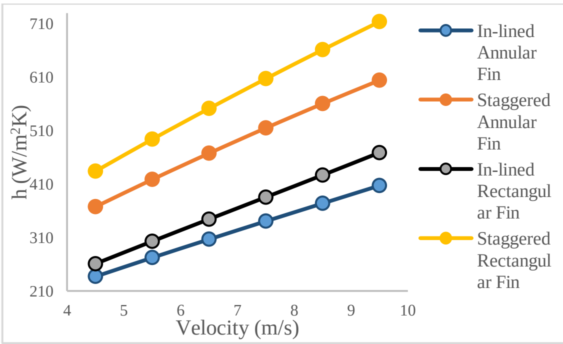
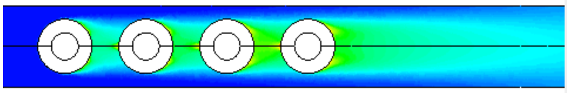
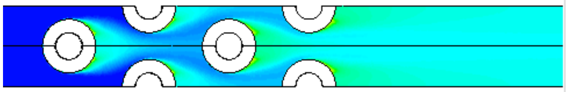


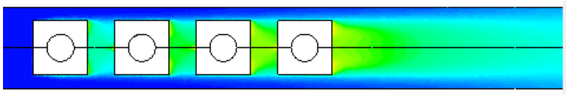
Figure.6: Heat transfer coefficient vs Velocity



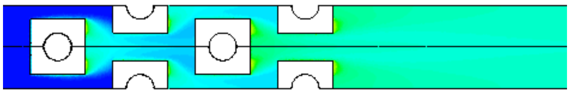
(a) Annular finned tube [In-line alignment]



(b) Annular finned tube [Staggered alignment]



(c) Rectangular finned tube [In-line alignment]



(d) Rectangular finned tube [Staggered alignment]

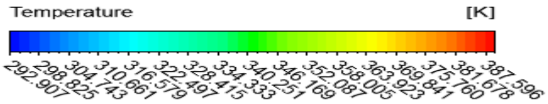


Figure.7: Temperature contour

4. CONCLUSION

Flow and heat transfer characteristics of annular and rectangular fin for different air flow rate was numerically studied. The flow is highly 3 dimensional and greatly affected by the type of the fin and arrangement of the fin. Fin arrangement has plays a major role on the turbulence of the flow and vortex formation. High turbulence and more vortex high main reason for the higher heat transfer in staggered tube arrangement. It is also evident that exit temperature decreases as the air flow rate increases but overall heat transfer increases with the air flow rate for all kind of tube and fin arrangement.

Rectangular finned tube shows 17-24% higher heat transfer characteristics than annular finned tube in general. It is mainly for the higher cross section area and weighted perimeter in the rectangular arrangement.

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8. NOMENCLATURE

Symbol	Meaning	Unit
T	Temperature	(K)
P	Pressure	(Pa)
ρ	Density	{kg/m^3}
V	Velocity	{m/s}
C_p	Specific heat	{kJ/kg.K}
k	Thermal Conductivity	{W/mK}
h	Heat Transfer Coefficient	{W/m^{2}K}