Numerical Experiment on H-Type Fin and Tube Heat Exchanger by Passing Coolant through the Tubes and blowing colder air across Fin

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ABSTRACT

Fin-and-tube heat exchangers with plain fins are widely used in direct air-cooled condenser system in the power plant, because of its relatively simple property compared to some other fins with variable cross-sectional area channel. In this paper, H-type finned tube heat exchanger on which its thermal efficiency and thermal effectiveness has been calculated, consists of an insulated casing inside which is a bundle of tubes with H-type rectangular fins. Water(coolant) which is used as a heating medium, is introduced into the inlet chamber and passes through the tubes and goes to the outlet chamber. The colder air flows across the finned tubes. The efficiency and effectiveness of the heat exchanger will be compared with respect to the different air flow velocity. Using Ansys, heat transfer performance and pressure drop for this near structure of fin-and-tube heat exchanger on the fin surface has been numerically analyzed. It is interesting to note that our proposed H-type finned structure gives better heat transfer rate.

Keywords: H-type Fin, Tube, Heat exchanger, Fin efficiency.

1. INTRODUCTION

Heat exchanger is a device to facilitate to exchange heat between two fluids without mixing at different temperature. Heat exchangers are used for both cooling and heating purpose. In order to improve the heat exchanger efficiency and extend surface heat transfer, H-type finned tubes have been widely used in boilers and waste heat recovery in recent years[1]. H-type fines are also known as square fin tube. H-type finned tubes are derived largely from rectangular finned tubes. The surfaces to tailor this problem is the H-type finned tube banks, which provide anti-wear and anti-fouling features due to the presence of unique groove-type structures on their fin surfaces. Yet, the surfaces also possess certain self-cleaning properties due to their special geometrical structure. In essence, this makes H-type finned tube banks one of the best candidates for offering appropriate, reliable, and relatively safe use in severe industry environments while still maintaining a high heat transfer capacity. Because of their unique groove structure on the fin surface, H-type finned tubes have excellent anti-wear and anti-fouling performance.[6] Numerous experimental and numerical studies have been conducted on the heat transfer and resistance characteristics of finned tubes. These studies mainly focus on spiral finned tubes, plain finned tubes and serrated finned tubes, while fewer studies exist, however, on the heat transfer and resistance characteristics of H-type finned tubes studied by Yu, X.; Yuan[2]. Heng Chen, Yungang Wang [3] studied the heat transfer and pressure drop characteristics of H-type finned tube banks by numerical simulation. The effects of geometric parameters and Reynolds number were examined by Yu *et al*., Chen and Lai [4] , performed experimental tests to examine the heat transfer and resistance characteristics of H-type finned tube banks and provided some reference for the design of the H-type finned tube bundles. In recent years, considerable efforts have been conducted to better understand and improve the airside heat transfer characteristics of H-type finned tube banks. Many experimental and numerical studies on H-type finned tube banks focused on the fin layout in the heat exchangers and the effect of varying geometric parameters like fin width, fin height, fin pitch, fin thickness, and air velocity on the overall heat transfer performance. Much research is limited to single factor experiments, limiting the obtained experimental correlations due to restricted experimental conditions and different fin structures[5]. Wu et al.[8] performed experimental studies concerning the airside performance of two novel fin-tube heat exchangers with delta winglet pairs. It was found that the heat transfer was enhanced with a reduction of pressure loss. Normally, the H-type finned tube bank houses a huge volume and weight that is not only costly, but also bulky as far as installation is concerned.

Varying test conditions cause the results of H-type finned tube’s heat exchange coefficient and resistance coefficient to be quite different, necessitating comprehensive experimental research on the heat transfer and resistance characteristics of H-type finned tubes.

2. Problem Description and Mathematical Modeling

The H-type finned tube heat exchangers on which its thermal efficiency and thermal effectiveness will be calculated, consists of an insulated casing inside which is a bundle of tubes with H-type rectangular fins. Water as a heating medium is introduced into the inlet chamber, passes through the tubes and goes to the outlet chamber. Across the finned tubes flows colder air. The efficiency and effectiveness of the heat exchanger will be compared with respect to the different air flow velocity. The numerical analysis is focused on air flow across tubes because the air side resistance of the heat exchanger makes up 80 % of the total resistance to the heat exchange.

2.1. Assumptions: The mathematical model is based on the following assumption:

- The fin material is homogenous and isotropic and the fin operates under steady state condition.
- There is no internal heat generation or absorption in the fin, loses heat by convection to its surroundings. The convective heat transfer coefficient is a constant.
- The temperature of the surrounding fluid remains constant during the heat rejection Process.
- Heat loss from the tip of the fin is negligible.

2.2. Governing Equations:

The mathematical model used to describe a physical problem is a set of differential equations and constitutive relations and initial and boundary conditions. The basic equations of fluid dynamics are derived from:

Conservation of Mass: $\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_j)}{\partial x_j} = 0$

Conservation of Momentum: $\frac{\partial(\rho u_j)}{\partial t} + \frac{\partial(\rho u_j u_i)}{\partial x_j} = \rho f_i + \frac{\partial \sigma_{ji}}{\partial x_j}$

Conservation of Energy: $\frac{\partial(\rho e)}{\partial t} + \frac{\partial(\rho e u_j)}{\partial x_j} = -\rho f_i u_i + \frac{\partial(\rho u_j u_i)}{\partial x_j} - \frac{\partial q_j}{\partial x_j}$

2.3 Boundary conditions:

Boundary and initial conditions are determined when defining the computational model and the physical properties of fluids and materials of the exchange surfaces. Boundary and initial conditions include defining:

- The air entering the computational domain is assumed to have uniform velocity over the cross section and ranges from 6m/s to 14m/s, and turbulence intensity 5 %.
- The air temperature at the inlet to the heat exchanger is 288 K.
- Hot water at the inlet to tubes has temperature 353 K. Because water has a high thermal capacity, it is assumed that the temperature of the tube inner wall is constant and equal to the water temperature.
- At the sides of the computational domain it is set symmetry boundary condition.
- The gauge pressure P_{out}at the outlet of channel is set to zero. This corresponds to atmospheric conditionP_{atm}.
- Hydraulically smooth walls were defined for outer tube and the fin walls.
- Symmetry condition is set for top, bottom, left and right side of computational domain. Symmetry condition is applied due to simplifying the calculation where it is possible to reduce computational domain.
- Physical properties of air are defined as polynomial function of temperature (and pressure) and thus set in Ansys Fluent software. Density of the air is based on incompressible ideal gas law.
- The physical properties of the fin and tube material are set constant (K=16.2 W/(m·K), ρ_t=7860 kg/m3).

2.4 Solution Methods:

The established mathematical model is solved by using the finite volume method that is based on dividing the computational domain into small volumes and integration of equations of conservation at these volumes thus obtaining a system of discrete algebraic equations that are then solved iterative. Computational domain is meshed by ANSYS software. Meshing was performed by using a hybrid mesh where most of the volume is structured mesh, and the smaller part around the fins is unstructured mesh.

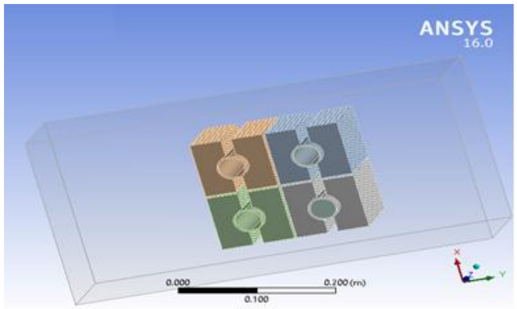

Figure. 1: 3D model of computational domain

An important aspect when performing numerical simulations is to judge whether the simulation is complete and converged, or not. There are several different ways to check this and ensure convergence.

Table 1-: Solutions methods and convergence criteria

Item	Solutions methods / convergence criterion
Pressure velocity coupling	SIMPLE
Momentum	Second order upwind
Turbulent kinetic energy	Second order upwind
Turbulent dissipation rate	Second order upwind
Convergence judgment conditions	Continuity, k, ω, velocity for x, y, z: 1×10^{-4} Energy: 1×10^{-9}

2.5 Calculation for Efficiency and Effectiveness:

. Temperatures of the H-type fins were taken from the ANSYS analysis. Then the relationship between fin efficiency and air velocity and the experimental correlation of fin efficiency was derived.

The relationship between fin efficiency η_f and air velocity v . As the air velocity, fin height and fin width increase, fin efficiency decreases. Fin pitch has little effect on fin efficiency. So fin efficiency is primarily associated with air velocity, fin height and fin width. With the increase in height and fin width, the average temperatures of the fin increases. According to the definition of fin efficiency, it will decrease with the average temperatures of the fins.

$\eta_f = \frac{Q_{act}}{Q_{max}}$; $Q_{act} = KA_c m\theta_0 [\frac{1+\frac{mk}{h}\tan hml}{\frac{mk}{h}+\tan hml}]$; h=5$^{watt}/_{m^2k}$; k=25$^{watt}/_{mk}$;

$A_c = Area\ of\ crosssection = h_1 * \delta$; $h_1 = 0.089m$; $\delta = 0.0025m$; m=$\sqrt{\frac{hp}{kA_c}}$; p= Perimeter=2$*$ $(h_1 + \delta)$; $\theta_0 = Temperature\ difference = (T_{max} - T_{atm})$; $Q_{max} = hA(T_{max} - T_{atm})$; A = Total Area of fin =2$*$ $h_1 * h_2$; $h_2 = 0.095m$

$\in = \frac{Q_{with\ fin}}{Q_{without\ fin}}$; $Q_{with\ fin} = Q_{actual}$; $Q_{without\ fin} = hA_c\theta_0$.

3. Results and Discussions

Fig 2 -4 shows that the comparison results for different velocity of air across the Fin and the contour of air flow and temperature distribution.

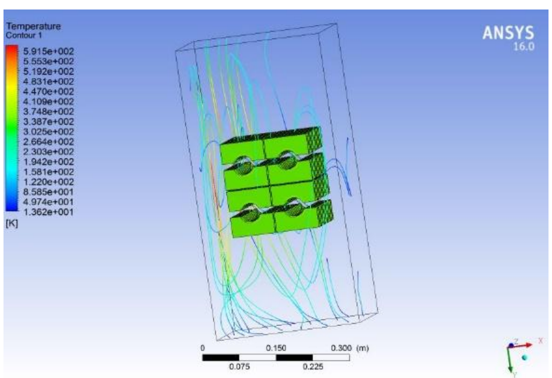

Figure 2 . for velocity 6m/s when $T_{max} = 591.5K; T_{min} = 13.62K; T_{base} = 447K$

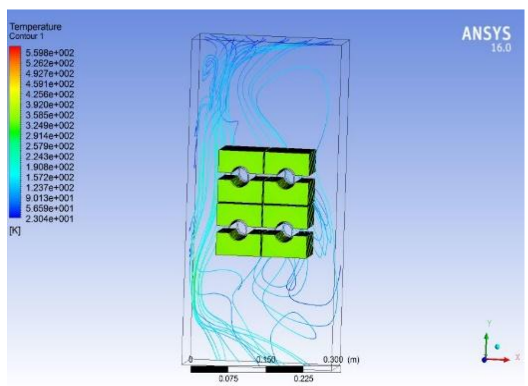

Figure 3 for velocity 8m/s when $T_{max} = 559.5K;\ T_{min} = 23.04K; T_{base} = 392.0K$

The below table shows the calculated values of efficiency and effectiveness of the designed H-type finned tube heat exchanger with respect to different air velocities.

Table 2 Result Table

Velocity	6m/s	8m/s	10m/s	12m/s	14m/s

Efficiency	67.62 %	63.24%	59.92%	54.66%	53.64%
Effectiveness	5.13	4.81	4.55	4.15	4.074

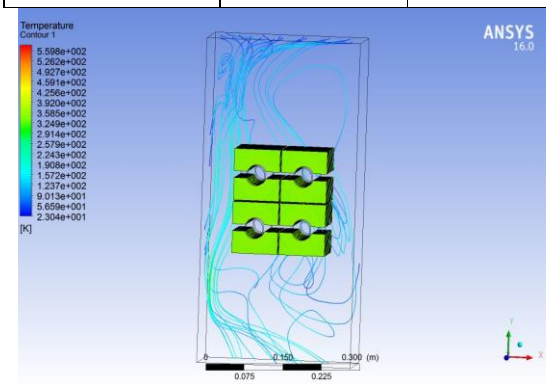

Figure 4 for velocity 10m/s when

$T_{max} = 425.6K; T_{min} = 23.04K; T_{base} = 425K$

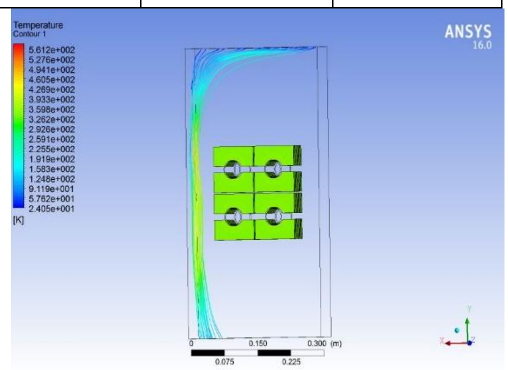

Figure 5 for velocity 12m/s

$T_{max} = 561.5K; T_{min} = 24.04K; T_{base} = 447K$

The graph in figure 6 (a) shows the relationship between the calculated efficiency values versus varying velocity. It shows that the efficiency of the heat exchanger varies inversely to the increasing velocity. The graph in figure 6.(b) shows the relationship between the calculated effectiveness values versus varying velocity. It shows that the effectiveness of the heat exchanger varies inversely to the increasing velocity

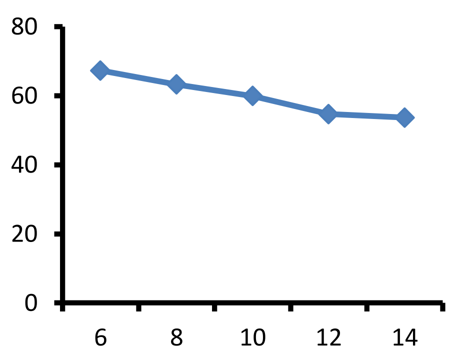

Figure 6 (a): Efficiency (In Y-axis) vs. Velocity (In X-axis) Graph;

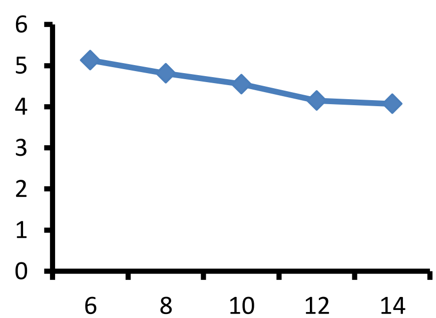

Figure 6 (b): Effectiveness (In Y-axis) vs. Velocity (In X-axis) Graph;

4. Conclusions

A set of Ansys fluent analysis has been performed to evaluate the influence of air velocity on the heat transfer, pressure drop and overall thermal performance of H-type finned tube heat exchanger. Based on the analysis, it is concluded that as air velocity v increases, fin efficiency η_f decreases for a definite fin design.

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