Optimization of Fin and Tube Heat Exchanger

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Abstract:- The objective of this paper is to develop an optimization methodology for the design of a finned tube heat exchanger in a residential air-conditioning unit. Three-dimensional simulations are carried out to investigate heat transfer and fluid flow characteristics of a two-row plain fin-and-tube heat exchanger. Heat transfer and pressure drop characteristics of the heat exchanger are investigated for Reynolds numbers ranging from 330 to 7000. Model geometry is created, meshed, calculated, and post-processed. Fluid flow and heat transfer are simulated and results compared using both laminar and turbulent flow models with steady-state solvers to calculate pressure drop, flow, and temperature fields. As far today's demand is to manufacture an effective heat exchanger to maximize heat transfer rate with low weight and high effectiveness. Finned tube heat exchanger widely used in Heating, Ventilation and Air Conditioning, Aerospace and Cryogenic applications. In automobile air conditioning R-134a is widely used refrigerant which is not environmental friendly because of its higher GWP (Global Warming Potential) value. Design of finned tube heat exchanger is done by using various correlations to calculate the heat transfer coefficient from air side and refrigerant side. R-22 and Propane is used as refrigerant because its thermophysical properties are better as compared to recently used refrigerant in automobile heat exchanger. It is eco-friendly natural refrigerant which has very low Global Warming Potential. In this paper, various details of processes involved in the fabrication of finned and tube heat exchanger is also mentioned.

Keywords:- *Optimization, Effectiveness, Ventilation, Global warming potential.*

I. INTRODUCTION

It facilitate the heat exchange between the two fluids at different temperatures while protecting them from mixing with each other. Fin and tube heat exchangers are mostly used in heating, ventilation and air-conditioning systems in a household, to chemical processing units and power production in larger plants. Heat exchangers are differ from the mixing chambers because they do not allow the two fluids involved to mix with each other. Heat transfer in fin and tube heat exchanger usually involves convection from tube surfaces to air and conduction through the tube walls. The Heat transfer rate between the two fluids at a location in a heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the heat exchanger. The simplest type of heat exchanger consists of two concentric pipes of different diameters and these heat exchangers are called Double-Pipe heat exchanger. From small diameter pipe one fluid flows and the other fluid flows via the spacing between the smaller and larger diameter pipes. There are Two types of flows are possible. In parallel flow heat exchanger both the hot and cold fluids enters the heat exchanger from the same end and they move in the same direction of flow but In counter flow heat exchanger, the hot and cold fluids enter the heat exchanger from opposite ends and these fluids flow in opposite directions of flow to each other. The heat exchangers which are designed to provide the larger heat transfer surface area per unit volume of heat exchanger are called Compact Heat Exchanger. The ratio of the heat transfer surface area of the heat exchanger to the volume of Heat Exchanger is called Area Density of heat exchanger. A heat exchanger with area density more than 700 m2/m3 is classified as being compact. Compact heat exchangers enable us to achieve high heat transfer rates between two fluids in a small volume, and they are commonly used in applications with strict limitations on the weight and volume of heat exchangers.

In compact heat exchangers, the two fluids usually move perpendicular to each other, and such flow configuration is called cross-flow. The cross-flow is further classified as unmixed and mixed flow. In industrial applications the shell-and-tube type heat exchangers are widely used. Shell-and-Tube type heat exchangers contain very large number of tubes (around 100) packed in a shell and tube heat exchanger with the axis of tubes parallel to that of the shell of heat exchanger. Baffles are used in the shell and tube heat exchanger to force the fluid from the shell side to flow across the shell of the shell and tube heat exchanger to enhance the heat transfer rate and to maintain the uniform spacing between the tubes of heat exchanger. But the shell-and-tube type heat exchangers are not suitable to be used in automobile and aircraft applications because shell and tube heat exchangers have relatively large size and weight. Fin and tube heat exchangers are widely used in Aerospace, Cryogenic applications, In Refrigeration, Heating, Ventilation and Air Conditioning Systems, due to lesser size and weight than shell and tube heat exchanger. Finned Tube heat exchangers are widely used in residential refrigeration and air conditioning systems.

II. TECHNIQUES OF HEAT EXCHANGER ENHANCEMENT

Researchers have been studying to reduce the air-side thermal resistance of the fin and tube heat exchanger . There are two type of techniques that are used to increase the heat transfer rate. Passive techniques and Active techniques. In Passive techniques the changes are done in the structure, shape of the fin and tube used in fin and tube type heat exchanger. Fluid additives can also be added. In Active techniques surface vibrations analysis and pumping power analysis is done. In our research paper only passive techniques are analysed and investigated for increase the heat transfer rate of fin and tube type heat exchanger. By creating more turbulence the heat transfer rate can also increase. Generally the interrupted fins are used in fin and tube type heat exchanger to increase the convective heat transfer coefficient on the air-side of the heat exchanger.

The average thickness of the boundary layer reduces by surface interruption of boundary layer. Some other methods can also be used to increase the fin surface interruptions like by using louvers, slits, waves, a combination of louvered and wavy fins geometries used in fin and tube type heat exchanger.

- *Materials used and Geometry of the Fin and Tube type Heat Exchanger*

Fins are generally made of aluminium or by using the alloys of aluminium, on the other hand the tubes of fin and tube heat exchanger are made of copper. Geometrical Parameters : Fin density of 6 to 16 fins per inch , Fin thickness 0.12 to 0.14 mm, Outside diameter of tubes: 11 mm, The Transverse pitch 24 mm, and Longitudinal pitch 21 mm.

Geometrical Parameters		
Fin thickness	t	*0.140 mm*
Number of tube rows		*2*
Longitudinal Pitch	P_l	*21.89 mm*
Tube wall thickness	δ	*0.295 mm*
Transverse pitch	P_t	*24.35 mm*
Longitudinal Pitch	P_l	*21.89 mm*
Tube wall thickness	δ	*0.295 mm*

Table 1. Dimensions of the Fin and Tube heat exchanger model

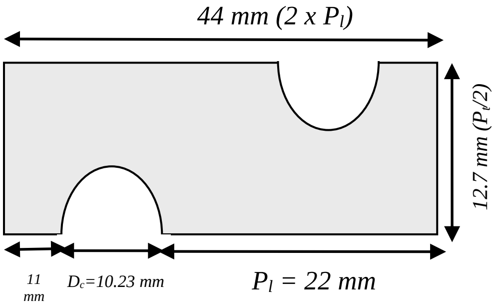


Fig 1:- Illustration of the main computational domain and geometric parameters of the heat exchanger model studied

A. *Processes Involved In Fabrication of Finned Tube Condenser*

- *Tube Cutting :-* In tube cutting process the Copper Tubes are cutted with measuring the required length of Copper Tubes.

- *Tube Bending* :- In tube bending there is a tendency of folding and buckling of tubes hence the strength of the tubes should be selected in such a way that it can resist the buckling and folding effects. The tubes are bend into the U shape. The tubes used in heat exchanger are made of Copper.

- *Deburring* :-Burrs are formed along the edges of the tubes after the process of tube bending, tube machining, trimming and casting. Generally these burrs are triangular in shape.

- *Tube Swaging* :- In tube swaging process the tube thickness and the internal diameter of the tube is reduced. In tube swaging internal mandrels may or may not be used.

- *Brazing* **:-** In Brazing process the base material does not melt only the filler material is melted and it is filled between the gap of the workpieces by the capillary action. The melting temperature of the filler material that is used in brazing process remains more than 475 degree. The filler material that is used in brazing process is called Spelter. The filler material remains the alloy of Copper and Zinc.

- *Vertical Expender Machine* **:-** After manufacturing of fin and Cu tubes, Cu tubes drive out in the hole of fins and then place in Vertical Expender Machine.

- *Oven Heating* :- After coming out expender machine fin and tube type of heat-exchanger is put in the oven where the temperature is between 150° to 170°c. This process is performed for removal of water & oil from it.

- *Leakage Testing* :- After passing from the oven leakage testing is being done if there would be any leakage then processes will be repeated again.

III. FIN AND TUBE HE PERFORMANCE PARAMETERS

The performance of fin and tube type heat exchanger is dependent on various parameters like: the velocity of air which is flowing near to the tubes and results in convective heat transfer, the drop in pressure of the fluid as it flows along the length of the tubes. This section also includes the various dimensionless numbers calculations like Reynolds number, Nusselt number, Prandtl number, equations for fins efficiency calculation , and equations for friction factor (f) and colburn *j*-factor calculation. The graph is also drawn to understand the effect of Reynolds number on the friction factor and on the colburn j factor.

Reynolds number Re

Equation-1

$$\mathrm{Re} = \frac{\rho \cdot V \cdot D_c}{\mu}$$

Where, *V* is the minimum free-flow air velocity (in the minimum flow cross-section of the tube row)

Equation -2

$$V = V \cdot \left\{ \frac{P_t \cdot F_p}{P_t \cdot F_p - D_c \cdot F_p - t(P_t - D_c)} \right\}$$

A. Fanning friction factor f

Equation-3

$$f = \frac{A_c}{A_o} \frac{\rho_m}{\rho_{in}} \left[\frac{2\rho_{in}\Delta p}{G^2} - (K_c + 1 - \sigma^2) - 2\left(\frac{\rho_{in}}{\rho_{out}} - 1 \right) + (1 - \sigma^2 - K_e) \frac{\rho_{in}}{\rho_{out}} \right]$$

B. Colburn j-factor

Equation-4

$$j = \frac{Nu}{Re_{D_c} \cdot Pr^{1/3}}$$

C. Nusselt number Nu

Equation 5

$$Nu = \frac{h}{k / D_h}$$

Equation 6

$$D_h = \frac{4(F_p - t)(P_t - D_c)P_l}{2(P_l P_t - \pi D_c^2 / 4) + \pi D_c (F_p - t)}$$

D. Prandtl number Pr

Equation 7

$$\mathrm{Pr} = \frac{\nu}{\alpha} = \frac{\mu C_p}{k}$$

E. Heat Transfer and Efficiency

Equation-8

$$\dot{Q} = (\dot{m}C_p)_h (T_{h,in} - T_{h,out}) = (\dot{m}C_p)_c (T_{c,out} - T_{c,in})$$

Equation 9

$$\dot{Q} = UA\Delta T_m$$

where ΔT_m refers to the true mean temperature difference.

F. LMTD equation

Equation-10

$$\varepsilon = 1 - \exp \frac{NTU^{0.22}}{C*[\exp(-C*NTU^{0.78}) - 1]}$$

Equation-11

$$C* = \frac{C_{\min}}{C_{\max}} = \frac{(\dot{m}C_p)_{air}}{(\dot{m}C_p)_{water}}$$

Equation-12

$$\varepsilon = \frac{\dot{Q}_{avg}}{\dot{Q}_{\max}} = \frac{\dot{Q}_{avg:(\dot{Q}_{wtr} + \dot{Q}_{air})/2}}{\dot{m}_{wtr} C_{p,wtr} (T_{wtr,in} - T_{air,in})}$$

Equation-13

$$NTU = UA / C_{\min}$$

Equation-14

$$\frac{1}{UA} = \frac{1}{\eta_0 h_o A_o} + \frac{\delta_w}{k_w A_w} + \frac{1}{h_i A_i}$$

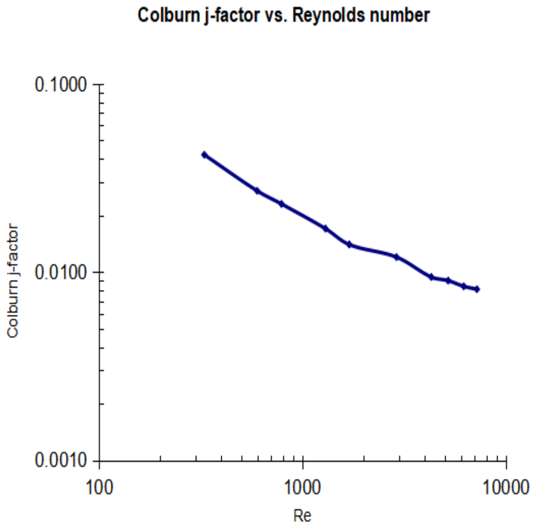
Equation-15

$$h_i = \left(\frac{k}{D} \right)_i \frac{(\mathrm{Re}_i - 1000)\,\mathrm{Pr}(f_i / 2)}{1 + 12.7\sqrt{f_i / 2}(\mathrm{Pr}^{2/3} - 1)}$$

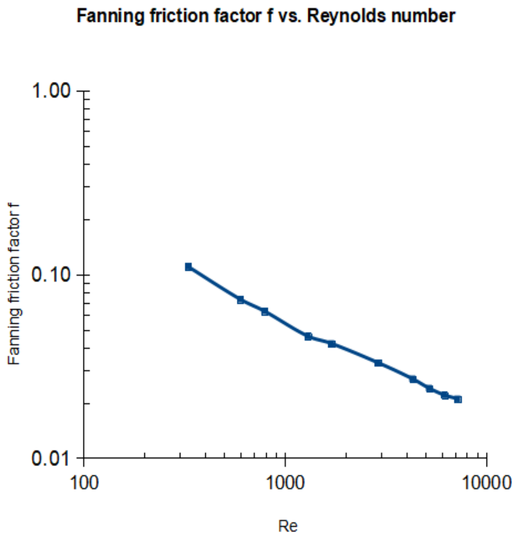
Equation-16

$$f_i = (1.58 \ln(\mathrm{Re}_{D_i}) - 3.28)^{-2}$$

IV. EFFECT OF REYNOLDS NUMBERS ON COLBURN J-FACTOR



Effect of Reynolds numbers on friction factor



V. RESULTS ANALYSIS

The computational domain has contains boundary conditions as shown in Figure 4.3 with the following conditions:

- For Tube surfaces, Dirichlet Boundary Conditions:

  Air velocity: $u = v = w = 0$
  $T = T_{wall}$,.

- For Fins, Dirichlet Boundary Conditions:

  Air velocity: $u = v = w = 0$
  $T = T_{finwall}$
  Air velocity: $u = v = w = 0$

- At Inlet, Dirichlet Boundary Conditions:

  Uniform velocity of air
  $u = u_{inlet}$,
  $v = w = 0$
  $T = 6\ ^{\circ}\mathrm{C}$.

- No Slip Conditionsat the top and bottom surfaces of fins :
  $(\partial u/\partial z)=0,\ (\partial v/\partial z)=0,\ w = 0,\ (\partial T/\partial z) = 0.$

- Symmetry Planes Boundary Conditions:

$(\partial u/\partial y)=0,\ v = 0,\ (\partial w/\partial y) =0,\ (\partial T/\partial y) = 0$

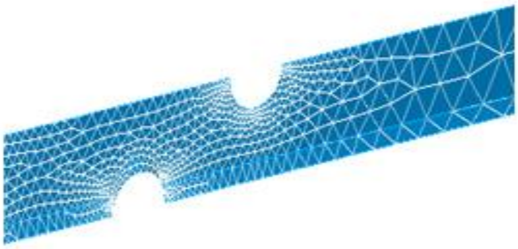


Figure-2: A mesh made up of 8465 tetrahedral cells

Tetrahedral meshes were created with cell numbers ranging from approximately 8,000 to 150,000 cells to be used for the grid independence test.

When the CFD simulations were first attempted, other problems with the calculations arose due to geometrical parameters. The cases could not make calculations, and it was necessary to investigate the problems.

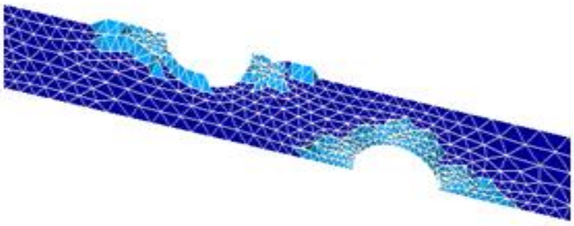


Figure-3: Illustration of a meshing problem: the underside of the mesh

VI. RESULTS AND DISCUSSIONS

There is a gradual decrease in friction factor as Reynolds number increases. The graph between friction factor and Reynolds number is plotted by taking total ten samples of Reynold number values. The range of the Reynold number which is being used ranging from 330 to 7K. In two-row tube-fin heat exchanger simulations are carried out which are having fin pitch 2.23 mm. The Reynolds number lies between 330 to 7000, which resembles to frontal air velocity which lies between 0.3 – 6.2 m/s at the inlet.

The reason for this study was to prepare a tool for optimization and the guidelines for designing finned-tube condenser heat exchangers during the time of finding effects of fin enhancements on performance of system along with investigating the use of an isolated component based fitness function to decrease calculation time and complexity of the model.

VII. CONCLUSIONS

It was found that the heat transfer rate and flow model accuracy depends on the flow regime and the friction factor .

There is a gradual decrease in friction factor as Reynolds number increases and with increase in Reynolds number the turbulence in the flow of fluid increases.

The increase in turbulence of the fluid results in the increase in heat transfer rate.

VIII. FUTURE SCOPE AND RECOMMANDATIONS

The properties of air are highly temperature-dependent, and many of the calculations do not account for these changes, but instead use an average value, which can substantially affect the flow at a particular cross section according to the temperature profile, hence to improve result accuracy the dependency of air properties can taken into consideration.
The effect of angle of inclination of the copper tubes and change in the shape of the fins in fin and tube heat exchanger can also be taken into the consideration for improving heat transfer rate.

The efficiency equation assumes a uniform air and fin temperature, which is not the case practically. The local convective heat transfer coefficient changes across the fin according to the temperature variation. The heat transfer coefficient is determined by assuming the steady fluid flow analysis. But actually the temperature of the fins and copper tubes changes with respect to time, hence transient flow analysis can also be taken into consideration. Further fine mesh can also be used to increase the accuracy of results. In our project structural grids are used but hybrid or unstructured grids can be used.

IX. LIMITATIONS

The change in shape of the fin is only possible if there is a modification in the die of the Fin Press Machine. The change in die configuration leads to the increase in the cost of fabrication of finned and tube heat exchanger.

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