DESIGN AND ANALYSIS OF DOUBLE PIPE HEAT EXCHANGER USING COMPUTATIONAL METHOD

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Abstract—Heat transfer equipment is defined by the function it fulfills in a process. On the similar path, Heat exchangers are the equipment used in industrial processes to recover heat between two process fluids. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, and natural gas processing. The operating efficiency of these exchangers plays a very key role in the overall running cost of a plant. So the designers are on a trend of developing heat exchangers which are highly efficient, compact, and cost effective. A common problem in industries is to extract maximum heat from a utility stream coming out of a particular process, and to heat a process stream. Therefore the objective of present work involves study of refinery process and applies phenomena of heat transfer to a double pipe heat exchanger.

Keywords—Thermal, Heat transfer, Computational Flow Dynamics (CFD), Modeling, Heat Flux, Heat transfer Coefficient

I INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions.

Typical applications involve heating or cooling of a fluid stream and evaporation or condensation of single- or multicomponent fluid streams. In other applications, the objective may be to recover or reject heat, or sterilize, pasteurize, fractionate, distill, concentrate, crystallize, or control a process fluid. In a few heat exchangers, the fluids exchanging heat are in direct contact.

In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak. Such exchangers are referred to as direct transfer type, or simply recuperators. In contrast, exchangers in which there is intermittent heat exchange between the hot and cold fluids—via thermal energy storage and release through the exchanger surface or matrix are referred to as indirect transfer type, or simply regenerators. Such exchangers usually have fluid leakage from one fluid stream to the other due to pressure differences and matrix rotation/valve switching.
Common examples of heat exchangers are shell and tube exchangers, automobile radiators, condensers, evaporators, air pre heaters, and cooling towers. If no phase change occurs in any of the fluids in the exchanger, it is sometimes referred to as a sensible heat exchanger. There could be internal thermal energy sources in the exchangers, such as in electric heaters and nuclear fuel elements. Combustion and chemical reaction may take place within the exchanger, such as in boilers, fired heaters, and fluidized-bed exchangers.

Mechanical devices may be used in some exchangers such as in scraped surface exchangers, agitated vessels, and stirred tank reactors. Heat transfer in the separating wall of a recuperator generally takes place by conduction. However, in a heat pipe heat exchanger, the heat pipe not only acts as a separating wall, but also facilitates the transfer of heat by condensation, evaporation, and conduction of the working fluid inside the heat pipe. In general, if the fluids are immiscible, the separating wall may be eliminated, and the interface between the fluids replaces a heat transfer surface, as in a direct-contact heat exchanger.

CLASSIFICATIONS OF HEAT EXCHANGERS
There are a number exchanger types based on the type of flow configuration, method of heat transfer and constructional features. The process designer has to select the best suitable type that meets the performance and operational requirements. Following is the list of heat transfer equipment.

Based on Principles of Operation (Transfer process)

- Recuperative Type (Direct Transfer)
- Regenerative type (Storage)
- Fluidized Bed Type

Based on Fluid Flow Arrangement

- Counter flow

- Parallel flow
- Cross Flow

Double pipe heat exchanger
This project work is based on design and thermal analysis of double pipe heat exchanger. Double pipe Heat exchangers (often also referred to as “double pipes”) are characterized by a construction form which imparts a U-shaped appearance to the heat exchanger. In its classical sense, the term double pipe refers to a heat exchanger consisting of a pipe within a pipe, usually of a straight-leg construction with no bends. However, due to the need for removable bundle construction and the ability to handle differential thermal expansion while avoiding the use of expansion joints (often the weak point of the exchanger), the current U-shaped configuration has become the standard in the industry. A further departure from the classical definition comes when more than one pipe or tube is used to make a tube bundle, complete with tube sheets and tube supports similar to the TEMA type exchanger.

Hairpin heat exchangers consist of two shell assemblies housing a common set of tubes and interconnected by a return-bend cover referred to as the bonnet. The shell is supported by means of bracket assemblies designed to cradle both shells simultaneously. These brackets are configured to permit the modular assembly of many hairpin sections into an exchanger bank for inexpensive future-expansion capability and for providing the very long thermal lengths demanded by special process applications.

One benefit of the hairpin exchanger is its ability to handle high tube side pressures at a lower cost than other removable-bundle exchangers. This is due in part to the lack of pass partitions at the tube sheets which complicate the gasketing design process. Present mechanical design technology has allowed the building of dependable,
removable bundle, hairpin multitubes at tube side pressures of 825 bar (12,000 psi).

The best known use of the hairpin is its operation in true countercurrent flow which yields the most efficient design for processes that have a close temperature approach or temperature cross. However, maintaining countercurrent flow in a tubular heat exchanger usually implies one tube pass for each shell pass.

![Double pipe heat exchanger](image)

**MATERIALS AND MANUFACTURING PROCESS**

The selection of materials of construction is a very important aspect to be considered before undertaking the design of heat exchangers. In each individual case, the choice of proper material shall be made, bearing in mind the specific requirements which are normally as follows:

1. Mechanical resistance i.e. strength and sufficient toughness at operating temperatures.
2. Chemical resistance under operating conditions with regard to corrosive media, concentration, temperature, foreign substances, flow behavior etc. the rate of corrosion must be negligible over a prolonged period of time and the material must be resistant against other corrosion phenomenon.
3. No detrimental interference by the material on the process or on the products.
4. Easy supply of material within time permitted and in the time required.
5. Good workability.
6. Lowest possible costs.

**PROBLEM DESCRIPTION**

The present work is based on industrial requirement. In the petroleum refinery, after distillation, different grades of oil come out at different high temperature which comes in to a pump and supplied at required level. The aim is to design a double pipe heat exchanger for an already existing suction pool of pump in which hot hydrocarbons are passing after distillation. The heat recovered from high temperature hydrocarbons is utilized to increase the temperature of crude oil up to required limit. The pool data, drawings, temperature of hydrocarbon and required temperature rise of crude oil were given by industry.

**LITERATURE SURVEY**

The goal of this chapter is to summarise some of the relevant studies from the extensive literature and material available on heat exchangers. This includes the latest studies on individual components using numerical techniques. The rationale is to put the present work in perspective with the state-of-the-art. For the past decades, several analyses of heat transfer and flow phenomena were carried out in components of heat exchanger tube side, shell side, fins, and baffles using numerical codes. The review is carried out on the literature available on heat exchanger. The complex nature of flow poses a challenge for both the numerical code and for the turbulence models. A review of previous works will highlight some of the drawbacks and challenges of obtaining a numerical solution of the flow. The review will also guide this work.
in presenting constructive results and conclusions on heat and flow modeling.

Behzadmehr et al., [1] have established numerically the critical Grashof numbers for transition from laminar to turbulent convection and relaminarization of fully developed mixed convection in a vertical pipe with uniform wall heat flux. A study of upward mixed convection of air in a long vertical tube with uniform wall heat flux has been conducted for two very low Reynolds numbers ($Re=1000$ and $Re=1500$) over a wide range of Grashof numbers ($Gr=10^8$) using a low Reynolds number $k$–$\varepsilon$ model with proven capabilities of accurately simulating both laminar and turbulent flows. The results in the fully developed region define three critical Grashof numbers for each Reynolds number. The smallest critical value distinguishes the $Re–Gr$ combinations that lead to a pressure decrease over the tube length from those leading to a pressure increase. The middle one corresponds to transition from laminar to turbulent conditions while the largest indicates the conditions for which relaminarization takes place.

**Objective:**
The objective of present work is based on design and thermal analysis of double pipe heat exchanger. The inner pipe is a suction pipe of the pump in which hot hydrocarbon flows and in outer annulus cold crude oil passes from opposite direction. The heat recovery from hot fluid is used to increase the temperature of cold fluid. Design was carried out based on the outlet temperature requirement of the cold fluid. With the help of computation fluid dynamics, the study and unsteady simulation was carried out for the designed heat exchanger and based on the simulation results, thermal analysis was carried out.

**DESIGN ASPECTS OF PRESENT HEAT EXCHANGER**

The thermal design is based upon a certain process parameters, the thermal and physical properties of the process fluids and the basic governing equations. Conventionally design method of heat exchanger is largely based on the use of empirical equations as well as experimental data which is available mostly in the form of graphs and chart.

**SALIENT FEATURES OF HEAT EXCHANGER**
The heat exchanger considered for analysis here has the following parameters

**Shell side:**
- Type of fluid: crude oil.
- Inlet temperature =313K
- Outlet temperature =553K.
- Mass flow rate=0.320kg/s.
- Diameter of shell=0.3m.

**Tube side:**
- Type of fluid: diesel oil.
- Inlet temperature =618K
- Outlet temperature =?.
- Mass flow rate=145.6kg/s.
- Specific gravity=0.874
- Tube inside diameter =0.2027m.
- Tube outside diameter=0.2191m
DESIGN OF DOUBLE PIPE HEAT EXCHANGER

Thermal design calculations
For this the following data are required:

<table>
<thead>
<tr>
<th></th>
<th>Hot side (tube side)</th>
<th>Cold side (shell side)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet temperature</td>
<td>$T_{in}$ = 615 K</td>
<td>$T_{in}$ = 313 K</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>$T_{out}$</td>
<td>$T_{out}$</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>$M_w$ = 1.456 kg/s</td>
<td>$M_c$ = 0.320 kg</td>
</tr>
<tr>
<td>Specific heat</td>
<td>$G_w$ = 3277.33 kg/k</td>
<td></td>
</tr>
<tr>
<td>Density</td>
<td>$\rho$ = 0.74 kg/m$^3$</td>
<td>$\rho$ = 969 kg/m$^3$</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$k$ = 0.1107 W/m$^2$</td>
<td>$k$ = 0.130 W/m$^2$</td>
</tr>
<tr>
<td>Viscosity</td>
<td>$\mu$ = 4.5 x 10^{-5} Ns/m$^2$ (oil)</td>
<td>$\mu$ = 9.75 x 10^{-5} Ns/m$^2$ (oil)</td>
</tr>
<tr>
<td>Thickness of fin</td>
<td>$t$ = 0.002 m</td>
<td></td>
</tr>
<tr>
<td>Fin pitch</td>
<td>$s$ = 0.142 m</td>
<td></td>
</tr>
<tr>
<td>Number of fins</td>
<td>$N_f$ = 34</td>
<td></td>
</tr>
</tbody>
</table>

$$Q_w = M_w C_w (T_{in} - T_{out})$$
$$Q_c = 0.320 \times 2491 (553 - 313) = 191308.79 W$$

$$T_{ho} = T_{hi} - \frac{Q_c}{M_h C_{ph}}$$

For counter flow LMTD

$$\Delta T_1 = T_{h,i} - T_{c,o}$$
$$\Delta T_1 = 618 - 553 = 65$$

$$\Delta T_2 = T_{h,o} - T_{c,i}$$
$$\Delta T_2 = 617.6 - 313 = 304.6$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \left(\frac{\Delta T_1}{\Delta T_2}\right)} = 155.12 K$$

$$T_{ho} = \frac{191308.79}{145.6 \times 3277.3} = 617.6 K$$

Flow area of fin

Consider as 10% fin cut, so the area of cut is calculated by the following

$$BF = H / D \times 100$$
$$H = 0.1 \times 0.3 = 0.03$$

$$1 - \cos \frac{\theta}{2} = 0.03 \Rightarrow 0.15 \Rightarrow 0.2$$

$$\cos \frac{\theta}{2} = 0.8$$

$$\theta = 36.86 \times 2 = 73.73$$

$$3.2$$

$$A_3 = \frac{R^2}{2} \left(\theta - \sin \theta\right)$$

$$0.15^2 \div (1.28 - \sin 73.73) = 3.676 \times 10^{-3} m^2$$

Area of fin

$$A_f = \frac{\pi}{4} (D^2 - d_o^2) - A_t \times 2 \times N_f$$

$$A_f = \frac{\pi}{4} (0.34^2 - 0.2191^2) - 0.00367 \times 2 \times 34 = 1.994138 m^2$$

Bare area (or) unfin area

$$A_B = \pi \times d_o \times \left(L - N_f (s - t)\right)$$

$$A_B = \pi \times 0.2191 \times [3.634(0142 - 0.002)] = 0.165114 m^2$$

Total fin area

$$A_t = A_f + A_B$$

$$A_t = 1.99413 + 0.165114 = 2.159 m^2$$

Perimeter

$$P = (2 \times 2 \times N_f \times h) + 2L$$

$$P = (2 \times 2 \times 34 \times 0.04045) + 2 \times 5 = 15.501 m$$
\[ D_e = \frac{2(AF + AB)}{(\pi F)} \]
\[ D_e = \frac{2(1.994183 + 0.165114)}{(\pi \times 15.5)} = 0.088732m \]

.velocity
\[ u_c = \frac{m_c}{\rho A_t} \]
\[ u_c = \frac{0.320}{698 \times 0.00367} = 0.124m/s \]

.Reynolds number
\[ \text{Re} = \frac{\rho u_c D_e}{\mu_c} \]
\[ \text{Re} = \frac{698 \times 0.124 \times 0.088732}{0.75 \times 10^{-3}} = 10239.91 \]

.Hence it is turbulent flow

.Friction factor
\[ f = (1.82 \ln \text{Re} - 1.64)^{-2} \]
\[ f = (1.82 \ln 10239.91 - 1.64)^{-2} = 0.03123 \]

.Colburn factor
\[ j = f / 2 \]
\[ j = 0.03123 / 2 = 0.015615 \]

.Prandtl number
\[ \text{Pr} = \frac{\mu_c C_{pc}}{k_c} \]
\[ \text{Pr} = \frac{0.75 \times 10^{-3} \times 2491}{0.130} = 14.37 \]

.Heat transfer coefficient
\[ h_o = j \times R_c \times p_c^{\frac{2}{3}} \times \frac{k_c}{D_e} \]
\[ h_o = 0.015615 \times 10239.91 \times 14.37^{\frac{2}{3}} \times \frac{0.130}{0.088732} = 1384.49W/m^2K \]

.Effective convection coefficient on inside
\[ \frac{h_o}{h_{fo}} = \frac{h_o}{h_o + h_{fo}} \]
\[ \frac{1}{1384.49 + \frac{1}{0.00058}} = 767.84W/m^2K \]

.Prandtl number
\[ \text{Pr} = \frac{\mu_h C_{ph}}{k_h} \]

.Reynolds number
\[ \text{Re}_b = \frac{\rho u_m d_i}{\mu_h} \]
\[ \text{Re}_b = \frac{874 \times 5.16 \times 145.6 \times 0.2027}{4.5 \times 10^{-3}} = 203143.237 \] Hence flow is turbulent.

.Prandtl number
\[ \text{Pr}_b = \frac{\mu_h C_{ph}}{k_h} \]
Friction factor
\[ f = (1.82 \ln R e_b - 1.64)^{-2} \]
\[ f = (1.82 \ln 203143.237 - 1.64)^{-2} = 0.015546 \]

Nusselt number
\[ N u_b = \frac{((f / 2)) (R e_b) Pr_b}{1.07 + 12.7 (f / 2)^{1/2} (Pr_b^{2/3} - 1)} \]
\[ N u_b = \frac{(0.015546 / 2)(203143.237) \times 134}{1.07 + 12.7(0.15546 / 2)^{1/2}(134^{2/3} - 1)} = 3448.87 \]

Heat transfer coefficient
\[ h_i = \frac{N_u k_b}{d_i} \]
\[ h_i = \frac{3448.87 \times 0.1107}{0.2027} = 1871.61 w / m^2 k \]

Effective convection coefficient on inside
\[ h_{ie} = \frac{h_i \times 1}{h_i + h_i} \]
\[ h_{ie} = \frac{1871.61 \times 0.00038}{1871.61 + 0.00038} = 1093.7 W / m^2 K \]

Efficiency of fin
\[ \eta_f = \frac{\tanh (m H_f)}{m H_f} \]
\[ m = \frac{2h_i}{tk_f} \]
\[ m = \frac{2 \times 1384.49}{0.002 \times 16.8} = 294.44 \]
\[ \eta_f = \frac{\tanh (294.44 \times 0.04045)}{294.44 \times 0.04045} = 0.761 \]

Overall efficiency of fin
\[ \eta_o = \left[ 1 - (1 - \eta_f) \frac{A_f}{A_e} \right] \]
\[ \eta_o = \left[ 1 - (1 - 0.761) \frac{1.994183}{2.15929} \right] = 0.779 \]

Outside convection coefficient referred to inside area
\[ \frac{h}{A} = \frac{(\eta (A e + \frac{1}{2})) h_{oe}}{A} \]
\[ h = \frac{0.779(1.99413 + 0.165144)}{\pi \times 0.2027 \times 5} = 426.71 W / m^2 K \]

Overall heat transfer coefficient
\[ u = \frac{\bar{h} \times h_{ie}}{h + h_{ie}} \]
\[ u = \frac{426.71 \times 1093.7}{426.71 + 1093.7} = 306.9 W / m^2 K \]

So the overall heat transfer coefficient is in the range

GEOMETRY:

The designed geometry under consideration in this thesis is a double pipe heat exchanger type or concentric tube heat exchanger with a circular fins or baffles the domain is sub divided in to two sections with a shell and tube channels.

Figures 3.2 show schematic two-dimensional views of the heat exchanger to be analyzed. This type of heat exchanger has been designed to recovery of heat from hot source (hot fluid) which is flowing through the tube and the shell side fluid is cold the flow is counter flow. The heat exchanger is one shell pass and one tube pass based on these conditions the heat exchanger is designed.
CFD ANALYSIS OF PLAIN TUBE

The temperature contours with respective magnitudes are shown in fig.4.3 the temperature on the Plane shows a small gain in temperature nearly 11 K, along the heat exchanger. So from the above fig the targeted temperature is not reached due to that reason chosen the circular fin with different material as well as different thickness the analysis has been carried out. And here circular fin will act as baffle to guide the flow.

NUMERICAL SIMULATION PROCEDURE

MODELING AND MESH GENERATION

Model of the heat exchanger in discussion is a double pipe heat exchanger with circular fins along the heat exchanger. The geometry was created in the GAMBIT. It also has the capability of handling direct CAD geometry inputs, geometry creation and editing, regions and zone definitions. The geometry in the study is complex and so is divided into two parts for simplification in geometrical modeling and mesh generation, the shell section, and tube section. The shell sections contain fins inside of the shell i.e. (it is on the tube section) two different sections in the present geometry. The dimensions of the heat exchanger were provided by the industry through drawings. The dimensions were tabulated as discussed in the previous chapter. The geometry creation in the GAMBIT is a very tedious procedure. The geometry in the GAMBIT builds up by first creating the vertices, edges, faces and then the volumes. All these volumes are joined by defining interfaces at the common faces.

The computational meshes for the heat exchanger domain were also created in the GAMBIT module. This software consists of modules that edit the mesh parameters and also compute a hexahedral element mesh (HEXA) or tetrahedral element mesh (TETA). The HEXA module is a 3D object-based, semi-automatic, multi-block structured and unstructured, surface and volume mesher. In addition, the T-Grid of this software allows to easily create complex grid topology. This feature makes it suitable for handling complex geometries such as heat exchanger without difficulty. The grid generation also follows a similar topology as such of geometry creation i.e., the edge, faces and then the volumes. Edge meshing for the both the shell and tube portions were done uniformly. It should be mentioned that great effort was made to resolve the grid close to the wall in the shell sections to capture the wall turbulence effects. The surface mesh of the entire tube was done by adopting HEXA mapped scheme. The volume mesh generation was the most challenging part, the tube sections were successfully meshed using structured mesh hexahedral element mesh (HEXA) scheme. The semi automated T-Grid scheme was implemented for the shell
and the fin sections creating tetrahedral mesh elements. Mesh quality which includes analysis of negative volumes and proximity, skewness and aspect ratio was performed. The maximum skewness was below 0.6, the corresponding aspect ratio was greater than 1 and there were no negative volumes. A validation of the mesh quality was performed and finally the output solver (.MSH) file was written. The outline of the domain and the grid with mesh are shown in the Fig 5.1(a, b, c).the total number of elements is 0.777820 millions.

NUMERICAL METHOD

There are a number of different numerical methods which are used in CFD codes to solve the governing differential equations. One common approach is the finite volume technique. The CFD code Fluent uses a control-volume-based technique to convert the governing equations to algebraic equations that can be solved numerically. Governing equations are solved sequentially (segregated solver). The CFD code solves the governing differential equations of turbulence in addition to the basic equations of motion.

![Geometry of heat exchanger without mesh](image1)

![Geometry of heat exchanger with mesh](image2)

Boundary conditions

Boundary-type specifications define the physical and operational characteristics of the model that represent on the model boundaries, groups of entities that are to be included in the boundary-type specification entity set. For the present model boundary condition defined on faces velocity inlet for both cold and hot fluid, at the exit of the shell and tube side pressure outlet and at shell outer face and tube outer surfaces is wall and interface is defined at the tube side.

RESULTS:

<table>
<thead>
<tr>
<th>Material/fin thickness</th>
<th>Steel</th>
<th>Aluminum</th>
<th>Copper</th>
</tr>
</thead>
<tbody>
<tr>
<td>t₁</td>
<td>0.002</td>
<td>0.002</td>
<td>0.002</td>
</tr>
<tr>
<td>t₂</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
<tr>
<td>t₃</td>
<td>0.004</td>
<td>0.004</td>
<td>0.004</td>
</tr>
<tr>
<td>t₄</td>
<td>0.005</td>
<td>0.005</td>
<td>0.005</td>
</tr>
</tbody>
</table>

Details of Comparative Study

The Plane and position of line in the flow domain

MATERIAL STEEL
Temperature variation on plane along the heat exchanger

Temperature variation across Outlet for varying Fin thickness.

The temperature contours with respective magnitudes are shown in fig 6.2. The temperature on the Plane shows a gradual increase in temperature along the heat exchanger. From fig 6.3 temperatures with respect to different thickness of the fin at the outlet on position of the line is observed. Initially the temperature variation with 0.002m fin thickness is taken from the position of the line and thereafter with increment of 0.001m fin thickness there is increase in value of temperature as fin thickness increases. As we go on increasing the fin thickness there is gain in temperature of 1.75K, 1.25K and 1K respectively. So by this increasing fin thickness the difference of temperature gain is less compared with earlier case. Likewise, if we go on increasing fin thickness, there is gain in temperature, increase in surface area and heat transfer. But as the quantity of material increases, the cost of material will increase.

MATERIAL ALUMINUM:

Temperature variations on plane along the heat exchanger

Temperature variation across Outlet for varying Fin thickness.

The temperature contours with respective magnitudes are shown in fig 6.4. The temperature on the Plane shows a gradual increase in temperature, along the heat exchanger. From fig 6.5 temperatures with respect to different thickness of the fin at the outlet on position of the line is observed. Initially the temperature variation with 0.002m fin thickness is taken from the position of the line and thereafter with increment of 0.001m fin thickness there is increase in value of temperature as increment of fin thickness. As we go on increasing the fin thickness the gain in temperature is 1K then 0.5K and 0.3K respectively. So by this increasing fin thickness the difference of temperature gain is less compared with earlier case. Likewise if we go on increasing fin thickness there is gain in temperature, increase in surface area and heat transfer. But as the quantity of material increases, the cost of material will increase.

MATERIAL COPPER

Temperature variations on plane along the heat exchanger
Temperature variation across Outlet for varying Fin thickness

The temperature contours with respective magnitudes are shown in fig 6.6. The temperature on the Plane shows a gradual increase in temperature, along the heat exchanger. From fig 6.7 temperature with respect to different thickness of the fin at the outlet on position of the line is observed. Initially the temperature variation with 0.002m fin thickness is taken from the position of the line and thereafter with increment of 0.001m fin thickness there is increase in value of temperature as increment of fin thickness. As we go on increasing the fin thickness the gain in temperature is 1K then 0.5K and 0.3K respectively. So by this increasing fin thickness the difference of temperature gain is less compared with earlier case. Likewise if we go on increasing fin thickness there is gain in temperature, increase in surface area and heat transfer. But if the quantity of material increases, the cost of material will increase

COMPARISON OF MATERIALS

Temperature variation on fin along the heat exchanger presents the temperature gradient in the cold fluid along the length of the heat exchanger with three different fin materials. The gradient is high in case (c) as compared to the other two cases. The contours of maximum temperature in the cold fluid shown are by the respective magnitudes. The cold fluid gains the heat from the hot fluid by convection and also by conduction through the fin material. The rate of heat conduction from fin to the cold fluid depends up on the thermal conductivity of the fin material. The higher conductivity lowers the resistance so faster the heat transfer. As conductivity of copper i.e. case (c) is highest among three cases, transfer of heat from copper fin to cold fluid will be faster. So the point of maximum temperature in the cold fluid from the entry will be occurring earlier in copper case as compared to other two cases. The contours are in good agreement with the above discussion. So the point of above temperature in case of steel is at a farther point from the inlet section when compared to other cases.
Variation of Heat Transfer with varying fin thickness

The above graph shows the relation between the heat transfer and fin thickness with different materials. As the fin thickness increases the value of heat transfer is increasing. For copper heat transfer is maximum value when compared to other materials and steel is having the least value of heat transfer.

COMPARISON OF MATERIAL WITH THICKNESS

Temperature variation at a position with fin material and thickness shows comparison of different material for all four thicknesses of fin. As shown in the Fig, if we move from steel to aluminum there is increase of approximately 10 to 12K. Plot shows that for all the thicknesses, gain in temperature with copper compared to aluminum is around 2 to 3K.

PRESSURE AND VELOCITY

Heat exchanger domain

Pressure variation along the domain
Velocity variation along the domain

Comparison of temperature with massflow rates for steel material

(a) massflow 0.120kg/s  (b) massflow 0.220kg/s

(a) massflow 0.120kg/s and (b) 0.220kg/s with temperature variation with different thickness. Above plots predicts that the temperature variation for different thickness with varying mass flow rates. In fig 6.12(a) predicts that as the fin thickness increases the value of temperature increasing 1.5K, 1.20K and 1K for both the cases (a) and (b). But, comparing with mass flow rates there is 40K higher with (a) than with (b).

Comparison of temperature with massflow rates for Aluminum material

(a) massflow 0.120kg/s  (b) massflow 0.220kg/s

massflow 0.120kg/s and (b) 0.220kg/s with temperature variation with different thickness. Above plots predicts that the temperature variation for different thickness with varying mass flow rates. In fig 6.12(a) predicts that as the fin thickness increases the value of temperature increasing 1K, 0.5K and 0.3K for both
Comparison of temperature with mass flow rates for copper material

SUMMARY RESULTS:

<table>
<thead>
<tr>
<th>Sr. no</th>
<th>Thickness of Ins (m)</th>
<th>Temperature (°C)</th>
<th>Pressure (Pascal)</th>
<th>Velocity(m/s)</th>
<th>Heat transfer (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.002</td>
<td>320.83</td>
<td>37.85</td>
<td>0.14</td>
<td>32.03</td>
</tr>
<tr>
<td>2</td>
<td>0.003</td>
<td>282.23</td>
<td>38</td>
<td>0.14</td>
<td>28.22</td>
</tr>
<tr>
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(a) Steel

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<th>Pressure (Pascal)</th>
<th>Velocity(m/s)</th>
<th>Heat transfer (W)</th>
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(b) Aluminium

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(c) Copper

Fig 6.14 Temperature variation with different thickness
CONCLUSIONS

1. As we increase the fin thickness the temperature of the cold fluid at the outlet of the heat exchanger increases.

2. We get high temperature profile at outlet in case of Aluminum and copper compared to steel material.

3. There is very minor changes occur in the pressure and velocity profile with increase of fin thickness as well as change of material that is pressure and velocity doesn’t get much affected by thickness of fin and material of fin.

4. The simulated outlet temperature is 543k which is very near to design outlet temperature 553k. There is less than 3% variation occurs than design value.

5. By decreasing the mass flow rate for there is increasing the value of temperature up to 609k and 577k.

6. After 5min there is no variation in temperature with respect to time.

FUTURE SCOPE

Heat transfer between to different liquids in parallel flow

Heat transfer between to different liquids in counter flow

Experimentation thermal analysis of double pipe heat exchanger

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