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Design and Analysis of Oil Cooling Shell and Tube Type Heat exchanger

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Abstract: Heat exchanger used for heat transfer take place for cooling or heating purpose. Shell and tube type heat exchanger is most commonly used in industrial application. The tube diameter, tube length, shell types etc. are all standardized and are available only in certain sizes and geometry, so the design of a shell-and-tube heat exchanger usually involves a trial and error procedure. A set of Design calculation were carried out to Design Shell And tube type heat exchanger For Screw compressor for cooling the oil and comparison were made between various parameters .By calculating the heat transfer coefficient & Pressure drop by changing parameter we came to know that,which parameter is safe for design of Shell and tube type heat exchanger. In Screw compressor oil used for compression process. Earlier practice in screw compressor oil is cooled by air medium. By using Shell and tube type heat exchanger we can successfully cool the oil by water medium.

This study has been undertaken to study design and analysis of the shell and tube heat exchanger. Shell and tube heat exchangers are found to be a widely used heat exchanger in industry for heat exchange purpose. This study shows the effect of various parameters on shell and tube type heat exchanger such as heat transfercoefficient ,pressure drop , pitch layout and baffle spacing. Standard Design calculations are used to study the same. The study also shows the simulation work carried out using 'Solidworks' for Shell and tube type heat exchanger.

Keywords: shell and tube type heat exchanger, flow simulation, solidworks, heat exchanger design

I. INTRODUCTION

A heat exchanger is a device used to transfer heat between a solid object and a fluid or between two or more fluids. The fluids may be separated by a solid wall to prevent mixing or they may be in direct contact. They are widely used in refrigeration, air conditioning, , chemical plants, petrochemical plants, power stations, petroleum refineries, natural-gas processing, and sewage treatment, in space for transferring heat. The best application of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolantflows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air.

A shell and tube heat exchanger is a class of heat exchanger designs. It is the most commonly used heat exchanger in oil refineries and other large chemical processes and also it is suited for higher-pressure applications. As its name suggest, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes, and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids.

JK Files Pvt. Ltd was established in 1978. JK files, a subsidiary of Raymond Ltd is today the largest producers of files in the world. They can manufacture according to any specifications provided by the customer. It has an impressive 32% global market share. The products are made from high quality materials and with modern manufacturing processes and conform to international standards.

The compressors used in the manufacturing plant uses an oil for the purpose of lubrication of parts of compressor. Since the temperature of oil gets increased, in earlier practice fan blower is used to lower the temperature of the oil using water. This cooled oil is again used for lubrication of compressor parts.

The heat exchanger is used in different industries for cooling fluid process. This heat exchanger is to be design for cooling of oil which is being circulated within the compressor. For this, the heat exchanger of shell and tube type is designed which wants to make it practicality in industry in cooling the hot oil from compressor.

A lot of research work has been carried out on the heat exchangers and its optimization with respect to thermal performance of such researches is explained in detailed in section II below.

II. LITERATURE REVIEW

Andre L.H. Costa and Eduardo M. Queiroz [1] presented a paper which deals with study about the design optimization of shell-and-tube heat exchangers. The formulated problem consists of the minimization of the thermal surface area for a certain service, involving discrete decision variables. Additional constraints represent geometrical features and velocity conditions which must be complied in order to reach a more realistic solution for the process task. The optimization algorithm is based on a search along the tube count table where the established constraints and the investigated design candidates are employed to eliminate non optimal alternatives, thus reducing the number of rating runs executed.

Abhishek Arya [2] carried out experimental work on fixed designed STHX and calculate the heat transfer coefficient and effectiveness. Validation is carried out which gives the result comparison with that of experimental result. Here flow parameters are not varied but size and number of tubes are varied and best efficient model is selected as Optimized value

III. METHODOLOGY

It is possible to study the effect of various parameters on the performance of heat exchanger with the advancement in simulation software's . This research work concern with design, analysis of flow simulation of the heat exchanger and studying the effects on various parameters. In order to validate the theoretical calculations flow simulation is carried out in Solidworks software, comparision have been conducted using different mass flow rates of hot and cold fluids on the heat exchanger

A. Details regarding Screw compressor

Specifications of Screw compressor

1) Application- Air screw Compressor
2) Make- ELGI
3) Rated Power- 125HP
4) CFM- 570
5) Outlet air pressure-7 bar
6) Max. air pressure- 8 bar
7) Oil type- Crude Lube Oil
8) Outlet oil temperature- 80-90˚C
9) Req. inlet oil temperature- 30 ˚C
10) Oil pressure- 2 bar
11) Oil flow rate- 5 LPM

Based on this parameters further design calculations are done using standard analytical reference book.this calculations are mentioned in below chapter 4.

IV. THEORETICAL DESIGN

A. Properties Of Crude Oil (shell side)

Table 4.1 represents the properties of crude oil which is being circulated in shell side of the heat exchanger. it represents the properties of crude oil such as temperature, specific heat, thermal conductivity, density, viscosity.

Table 4.1 Properties of crude oil

Crude Oil	I/P	O/P
Temperature(°C)	90	30
Specific Heat(J/KgK)	2110	1988.9
Thermal conductivity(W/mK)	0.132	0.1355
Density(Kg/m³)	787	787.4
Viscosity(Nm/s²)	0.0014	0.00487

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B. Properties Of Water (Tube Side)

Table 4.2 represents the properties of water which is being circulated in tube side of the heat exchanger.it represents the properties of water such as temperature, specific heat, thermal conductivity, density, viscosity.

Table 4.2 Properties of water

Water	I/P	Avg	O/P
Temperature(°C)	25	27.6	30.2
Specific Heat(J/KgK)	4180	4180	4180
Thermal conductivity (W/mK)	0.6115	0.61826	0.618
Density(Kg/m³)	996.4	995.776	995.15
Viscosity(Nm/s²)	0.00086	0.00083	0.0008

C. Calculation for Counter Flow

1) Heat Transfer Coefficient (q)

$= m.c_p.(T_{in} - T_{out})_{oil}$

$=0.075*2052*(90-30)$

$=9234$ W

2) Heat absorbed by water (Water)

$= m.c_p.(t_{in} - t_{out})_{water}$

$9234 = 0.425*4180*(t_{out} - 25)$

$\rightarrow t_{out} = 30.2°C$

3) Logarithmic Mean Temperature Difference (LMTD)

$$\Delta T_m = \frac{(T_{in} - t_{out}) - (T_{out} - t_{in})}{\ln\left(\frac{T_{in} - t_{out}}{T_{out} - t_{in}}\right)}$$

$$= \frac{(90 - 30.2) - (30 - 25)}{\ln\left(\frac{90 - 30.2}{30 - 25}\right)}$$

$=22.0828$ °C

4) Rand S are considered as dimensionless temperature ratios

$$R = \frac{T_{in} - T_{out}}{t_{out} - t_{in}}$$

$$= \frac{90-30}{30.2-25}$$

$=11.538$

$$S = \frac{tout - tin}{Tin - tin}$$

$$= \frac{30.2-25}{90-25}$$

$=0.08$

5) The log mean temperature correction factor (Ft) can be given as,

$$F_t = \frac{\sqrt{(R^2+1)}\,ln\left(\frac{1-S}{1-RS}\right)}{(R-1)\,ln\left[\frac{2-S[R+1-\sqrt{(R^2+1)}]}{2-S[R+1+\sqrt{(R^2+1)}]}\right]}$$

$$= \frac{\sqrt{(11.538^2+1)}\,ln\left(\frac{1-0.08}{1-11.538*0.08}\right)}{(11.538-1)\,ln\left[\frac{2-0.08[11.538+1-\sqrt{(11.538^2+1)}]}{2-0.08[11.538+1+\sqrt{(11.538^2+1)}]}\right]}$$

$=0.95$

$\Delta T_{mc} = F_t * LMTD$

$=0.95*22.0828$

$=20.97°C$

6) Overall Heat Transfer coefficient Assumed

$(U_{assumed})$= 60 W/m^2∘C

7) Provisional Area

$$A=\frac{q}{U*\Delta T_{mc}}$$
$$=\frac{9234}{60*20.97}$$
$$=7.33m^2 \cong 7.35\ m^2$$

8) Tube Dimensions

Tube Inner Diameter (d_i)= 0.012m

Tube Outer Diameter (d_0)= 0.016m

Tube length=1.5m

9) Surface Area

$$A_{sc} = \prod \times d_o \times L$$
$$= \prod \times 0.016 \times 1.5$$
$$=0.0754\ m^2$$

10) No. of Tubes

$$N=\frac{A}{Asc}$$
$$=\frac{7.35}{0.0754}$$
$$= 97.48 \cong 98$$

As the Shell side Fluid is relatively clean use Triangular Pitch and considering 1:2 pass (1shell pass and 2 tube passes)

(K= 0.249 , n = 2.207)

11) Bundle Diameter

$$D_b= d_o \left(\frac{N}{k}\right)^{1/n}$$
$$= 0.016\left(\frac{98}{0.249}\right)^{1/2.207}$$
$$=0.2398 \cong 0.24m$$

12) Bundle Diameter Clearance

c =0.044.8 m

13) Shell Diameter

$$D_s=D_b + c$$
$$=0.24 + 0.044.8$$
$$=0.2848\ m$$

D. Tube Fluid film Coefficient

1) Mean Water Temp$=\frac{tout + tin}{2}$

$$=\frac{25+30.2}{2}$$
$$=27.6∘C$$

2) Cross Sectional Area Of Tube

$$A_c = \frac{\Pi}{4}\times d_i^2$$
$$=\frac{\Pi}{4}\times 0.012^2$$
$$= 0.0001130\ m^2$$

3) Tube Velocity (Water linear velocity)

$$(u_w)=\frac{m_w}{\rho_w*\frac{N}{2}*Ac}$$
$$=\frac{0.425}{995.78*\frac{98}{2}*0.000113}$$
$$=0.07708\ m/s$$

4) Reynolds Number

$$Re = \frac{\rho.u.d_i}{\mu}$$

$=\frac{995.776\times 0.07701\times 0.012}{0.00083156}$

$=1106.7$

5) *Tube Side Heat Transfer Coefficient*

$h_i = \frac{c_p . u_w^{0.8} . (1.35 + 0.02 \times \frac{tout + tin}{2})}{d_i^{0.2}}$

$=\frac{4180 * 0.077^{0.8} * (1.35 + 0.02 \times \frac{30.2 + 25}{2})}{12^{0.2}}$

$=622.03$ W/m^2∘C

E. Shell Side Coefficient

1) Baffle Spacing

$l_B = \frac{Ds}{5}$

$=\frac{0.2448}{5}$

$=0.04896$ m

2) *Tube Pitch*

$T_p = 1.25 \times d_o$

$= 1.25 \times 0.016$

$= 0.02$ m

3) *Cross Flow Area*

$A_s = \frac{(T_p - d_o)}{T_p} \times D_s \times l_B$

$= \frac{(0.02 - 0.016)}{0.02} \times 0.2848 \times 0.059$

$=0.003244 m^2$

4) *Mass Velocity*

$G_s = \frac{m_o}{A_s}$

$=\frac{0.075}{0.003244}$

$=23$ Kg/s m^2

5) *Equivalent Diameter*

$d_e = \frac{1.1}{d_o}(T_p^2 - 0.917 \times d_o^2)$

$=\frac{1.1}{16}(20^2 - 0.917 \times 16^2)$

$= 11.36$ mm $= 0.01136$ m

6) *Reynolds Number*

$Re = \frac{G_s * d_e}{\mu}$

$=\frac{23 * 0.01136}{0.00316}$

$=83.32$

7) *Prandtl Number*

$P_r = \frac{c_p * \mu}{k_f}$

$=\frac{2052 \times 0.00316}{0.134}$

$=48.39$

Choose 25% baffle Cut

8) $J_H = 0.5\,(1 + \frac{l_B}{D_s})(0.08 Re^{0.6821} + 0.7\ Re^{0.1772})$

$=0.5\,(1 + \frac{1}{5})(0.08 * 75.7^{0.6821} + 0.7 * 75.7^{0.1772})$

$=1.89$

F. Shell Side Heat Transfer Coefficient Can Be Given As

1) $h_s = \frac{k \times J_H \times {P_r}^{\frac{1}{3}}}{d_e}$

$= \frac{0.134 \times 1.88 \times 48.39^{\frac{1}{3}}}{0.01136}$

$= 81.53 W/m^2 \circ C$

2) *Overall Coefficient*

$$\frac{1}{U_o} = \frac{1}{h_s} + \frac{1}{h_{od}} + \frac{d_o * ln\left(\frac{d_o}{d_i}\right)}{2k_w} + \left(\frac{d_o}{d_i}\right) * \frac{1}{h_{id}} + \left(\frac{d_o}{d_i}\right) * \frac{1}{h_i}$$

$$\frac{1}{U_o} = \frac{1}{81.57} + \frac{1}{5000} + \frac{0.016 * ln\left(\frac{16}{12}\right)}{2 * 45} + \left(\frac{16}{12}\right) * \frac{1}{3000} + \left(\frac{16}{12}\right) * \frac{1}{622.03}$$

$U_o = 66.5 W/m^2 \circ C$

$$(U_o = 66.5) > (U_{o\ assumed} = 60)$$

G. Pressure Drop

1) *Tube side Pressure Drop*

From Fig (12.24)(R.K.Sinnot) for Re = 1106.7

$J_f = 0.009$

$\Delta P_t = N_p \times (8 \times J_f (\frac{L}{di}) + 2.5) \times \rho . \frac{{\mu_t}^2}{2}$

$= 2 \times (8 \times 0.009 (\frac{1.5}{0.012}) + 2.5) \times 995.7 * \frac{0.077^2}{2}$

$= 67.89 N/m^2 = 67.89 Pa$

2) *Shell Side Pressure Drop*

For Re = 75.70 from fig 12.30 $J_f = 0.32$

$\Delta P_s = 8 \times J_f \times (\frac{Ds}{de}) \times (\frac{L}{l_b}) \times (\rho . \frac{{\mu_s}^2}{2})$

$= 8 \times J_f \times (\frac{Ds}{de}) \times (\frac{L}{l_b}) \times (\rho . \frac{\left\{\frac{G_s}{\rho}\right\}^2}{2})$

$= 8 \times 0.32 \times (\frac{0.284.8}{0.01136}) \times (\frac{1.5}{0.059}) \times (818.95 * \frac{\left\{\frac{21.05}{818.95}\right\}^2}{2})$

$= 487.28 N/m^2 = 487.28 Pa$

H. Component Design

1) *Shell*

Shell Thickness $= \frac{p * D_i}{2f.J - p} + c$

$= \frac{.5 * 284.8}{2 * 130 * .75 - .5} + 5$

$= 5.73 mm \cong 10 mm$

Shell Outer Diameter = 284.8+20mm

=304.8 mm

2) *Shell Cover*

Shell Cover inner diameter

Ds=284.8 mm

Thickness of torispherical head (h)

Ri=284.8 mm

$r_i = 0.06 * Ri$

=0.06* 284.8

=17.0688 mm

$$W=\frac{1}{4}\left(3+\sqrt{\frac{Ri}{ri}}\right)$$

$$=\frac{1}{4}\left(3+\sqrt{\frac{284.8}{17.066}}\right)$$

$$=1.77$$

Corrosion Allowance=3mm

Thickness of a torispherical head

$$t_h=\frac{p*Ri*w}{2*f*j-0.2*p}+c$$

$$=\frac{0.5*284.8*1.77}{2*130*0.7-0.2*p}+3$$

$$=4.43mm$$

Inside depth of the head (hi)

$$h_i=R_i-\left[\left(Ri-\frac{Ds}{2}\right)\left(Ri+\frac{Ds}{2}\right)+2ri\right]^{1/2}$$

$$h_i=284.8-\left[\left(284.8-\frac{284.8}{2}\right)\left(284.8+\frac{284.8}{2}\right)+2*17.688\right]^{1/2}$$

$$=40mm$$

3) Channel cover

Outer diameter=304.8 mm

Thickness (t_{cc})

$$t_{cc}=\frac{Dc}{10}\frac{\sqrt{c1*p}}{f}+c$$

$$=\frac{304.8}{10}\frac{\sqrt{0.3*5}}{13}+3$$

$$=5.87mm$$

4) Pass partition plate= 10mm

5) Tube sheet thickness

$$tss=\frac{F*Gp}{3}\sqrt{\frac{p}{k*f}}$$

$$=\frac{1.25*284..8}{3}\sqrt{\frac{0.5}{0.419*130}}$$

$$=11.76mm \cong 15mm$$

6) Nozzle

Nozzle Diameter=50.8 mm (2 inch)

Thickness

$$\boldsymbol{tn}=\frac{p*Dn}{2Fj-p}+c$$

$$=\frac{0.55*50.8}{2*138*0.85-0.55}+3$$

$$=3.11 \text{ mm}$$

7) Design of gaskets

Gasket factor ($\boldsymbol{m}$) =2.5; Maximum design seating stress ($\boldsymbol{Y}$), kgf/mm^2= 2.04

$$\frac{Dog}{Dig}=\sqrt{\frac{Y-pm}{Y-p(m+1)}}$$

$$\frac{Dog}{285.05}=\sqrt{\frac{2.04-0.056*2.5}{2.04-0.056*3.5}}$$

$$D_{og}=290.8mm=292.8mm$$

Width 8mm below

8) Flange Thickness

Flange thickness=25m

9) Bolts

bolt circle diameter (Cb) = 375 mm

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Root diameter r=18 mm
Root Diameter =18 mm

10) Baffles

Baffle Spacing

$l_B = \frac{Ds}{5}$

$=\frac{0.2448}{5}$

=0.04896 m

Tube vertical pitch

$T_p{}'$=0.87 *T_p

=87*20

=17.4 mm

Baffle cut height

Hc=0.25* D_s

=0.25*284.8

=71.2mm

Height between baffle tips

H_{bt}= D_s – 2*H_c

=284.8-2*71.2

=142.4mm

Number of constrictions crossed

Ncv=$\frac{H_{bt}}{T_p{}'}$

$=\frac{142.4}{17.4}$

=8.183

From Figure 12.32 (R.K.Sinnot) F_n=0.99 for Ncv=8.183

Height from the baffle chord to the top of the tube bundle

H_b=$\frac{D_b}{2}$ - D_s(0.5 - B_c)

$=\frac{240}{2}$ – 284.8(0.5 -0.25)

=48.8mm ≅49 mm

Bundle cut =$\frac{H_b}{D_b}$

$=\frac{49}{240}$

=0.2041

From Figure 12.41 (R.K.Sinnot) at cut of 0.2041 R_a'=0.14

Tubes in one window area,

N_w=N*R_a'

=98*0.14

=13.72≅14

Tubes in cross-flow area

N_c= N – 2*N_w

= 98 – 2*14

=70

Number of baffles

Nb=$\frac{L}{l_B}$ -1

$=\frac{1500}{56.96}$ -1

=25.33≅25

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Table 4.1 represents the various components its dimensions and its material

Table:41 Components and its material

Component	Specimen	Dimension	Material
SHELL	Inner Diameter	284.8 mm	Mild Steel
	Outer Diameter	304.8 mm	
TUBE	Inner Diameter	16 mm	Stainless Steel A 320
	Outer Diameter	12 mm	
	pitch	20 mm	
BAFFLE	Baffle Spacing	500 mm	Mild Steel
	No. Of Baffles	2	
	Baffle thickness	6 mm	
HEAD	Thickness	10 mm	Stainless steel
	Crown radius	284.8 mm	
	Knuckle radius	20 mm	
	Inside Depth	100 mm	
PASS PARTITION PLATE	Thickness	10 mm	Stainless steel
GASKET	Inside diameter	284.8 mm	Soft Al
	Outide diameter	292.8mm	
FLANGE	Thickness	25 mm	Gray Cast Iron HT150
TUBE SHEET	thickness	15 mm	Gray Cast Iron HT150
BOLT (M16)	No of Bolts	12	Hot Rolled Carbon Steel
	Pitch Circle	375 mm	
	Root Diameter	18 mm	

V. CAD Model

Fig .5.1 and 5.2 represents the cad model of Tubes and outer components respectively.

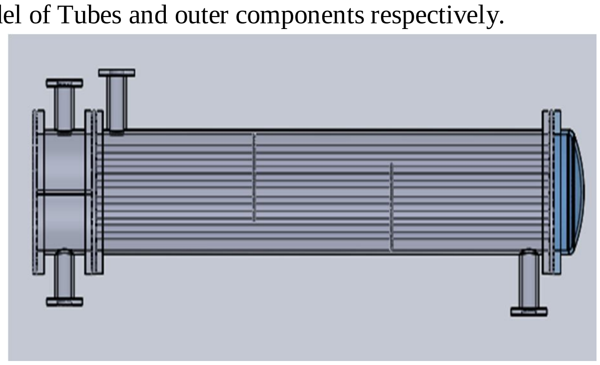

Fig. 5.1 :CAD Design (Tubes and Baffles)

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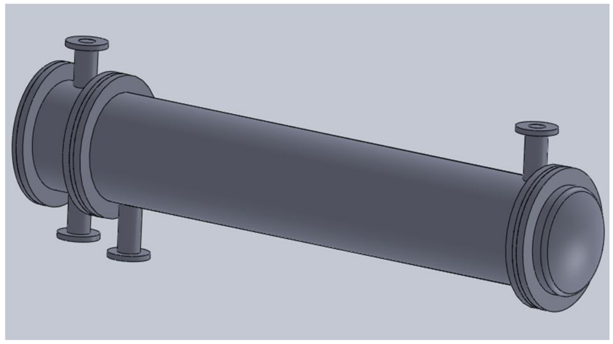

Fig 5.2 :CAD Design (Outer Components)

VI. RESULT

A. *Effect Of Different Parameters On Heat Transfer Coefficient*

1) Effect of tube outside diameter for same mass flow rate of water.

Table 6.1 Tube Outer Dia. and heat transfer coeff.

Sr No.	1	2	3	4	5
Tube Outer dia(mm)	16	20	25	30	38
Heat Trans Coeff.	65.29	51	42.13	35.54	27.96

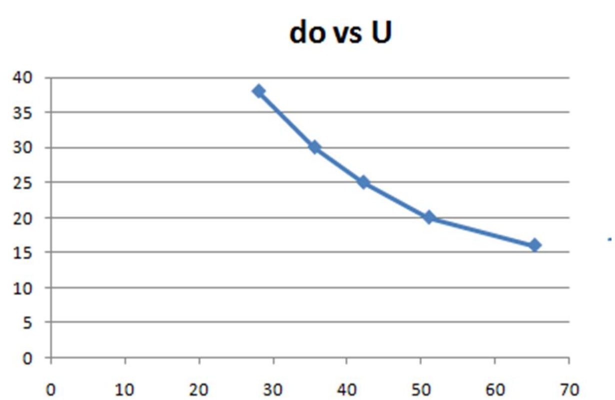

Fig.6.1. Graph of **$d_o$vs U**

2) Effect of mass flow rate for same tube outside diameter

Table 6.2 mass flow rate and same tube outside diameter

Sr No.	1	2	3	4	5
Mass flow rate (LPM)	10	15	20	25	30
Heat Transfer Coe.	55.9	56.4	60.68	62.72	63.99

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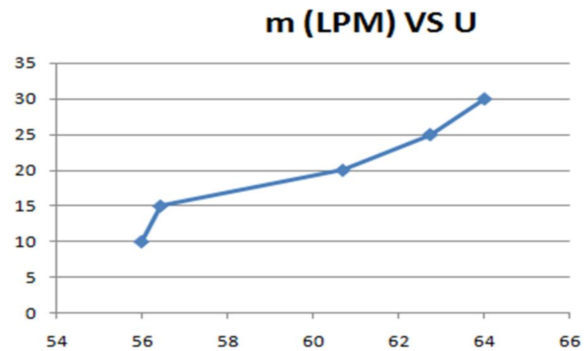
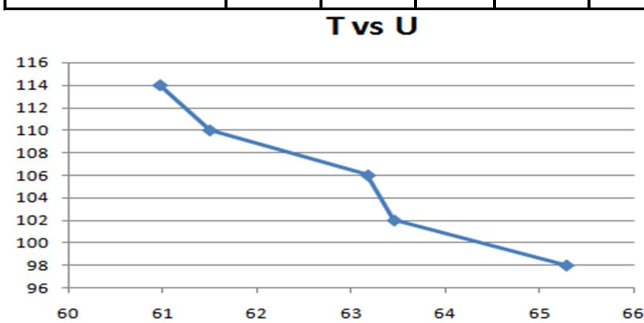


Fig.6.2. Graph of m**vs U**

3) Effect of no. of tubes for same mass flow ı U

Table:6.3 No. of tubes for same mass flow rate

Sr No.	1	2	3	4	5
No. of Tubes	98	102	106	110	114
Heat Tran. C	65.2	63.4	63.	61.5	60.9



U

Fig.6.3. Graph of T **vs U**

B. Results from flow Simulation

Fig 6.4 and Fig 6.5 represents Solidworks flow simulation results for different tube material i.e. steel and copper respectively

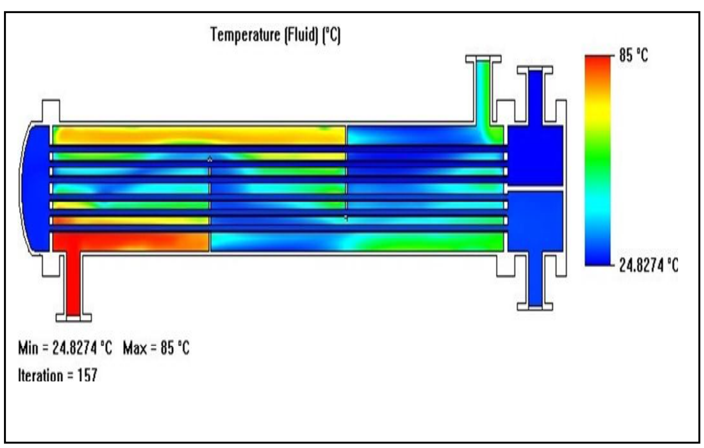


Fig.6.4Simulate when, tube material- Steel(K=30w/mk)

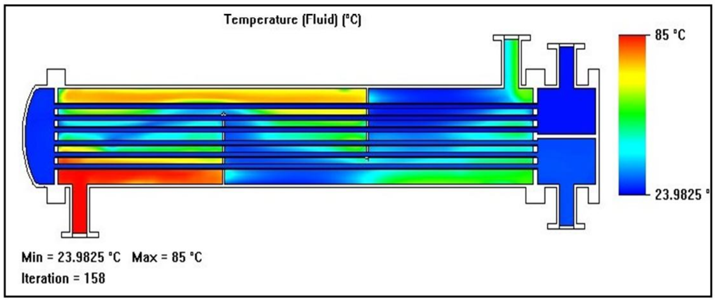


Fig.6.5 Simulate when, tube material- copper(K=220w/mk

VII. CONCLUSION

From the results of flow simulation of the heat exchanger it can be concluded that the simulation gives results close to those obtained from the theoretical calculations.

From the calculation it can be concluded that the overall heat transfer coefficient gets affected due to various parameters such as baffle spacing and tube pitch layout. It can be found that using triangular pitch overall heat transfer coefficient increases, also it increases with decrease in the baffle spacing.

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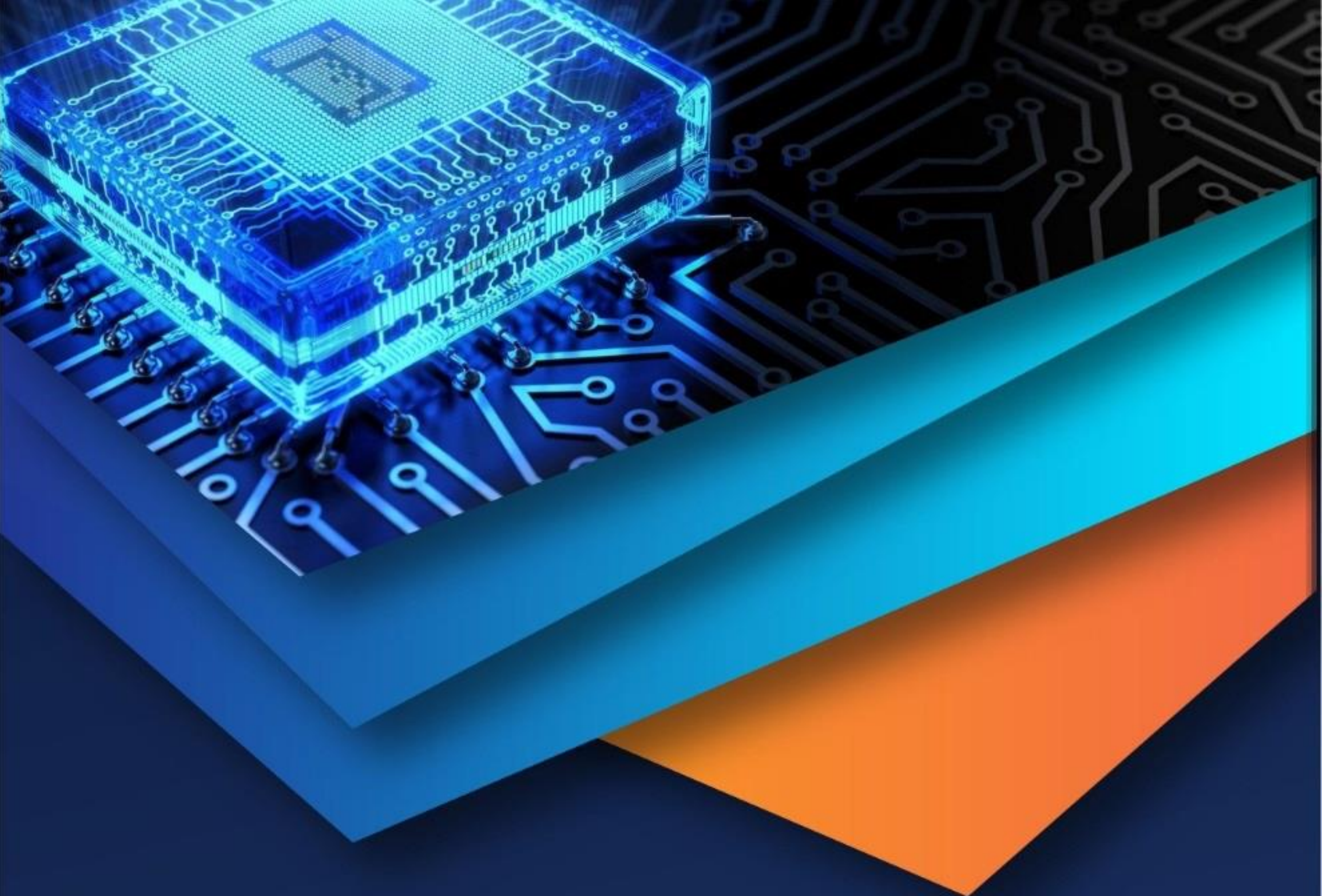

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