BANGLADESH UNIVERSITY OF ENGINEERING AND TECHNOLOGY

Thermo-Fluid System Design

Course No: ME 310

Submitted by:

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Abstract:

Double pipe hairpin heat exchangers with bare inner tubes are sometimes used in large scale industry for cooling purpose of hot oil. To design such type of heat exchanger for industrial level, one has to perform rigorous hand calculation with a lot of design considerations. To verify this calculation, one needs to run simulation. Economic analysis is also needed with engineering formulation to evaluate the efficiency of the whole heat exchanger system. We have used EXCEL for manual hand calculation and HTRI for simulation purpose.

Steel pipe is used for both inner and outer tube to make the system thermally and economically efficient. SAE30 oil is flowing through the inner tube and cooling water in the annulus area. To maintain the flow of fluid, two small size of pump are also attached. We have selected these parameters to obtain our purposes to cool the oil effectively and also to maintain the given limitations.

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Problem Statement:

Fluid	SAE-30 oil	Sea-Water
Inlet temperature,ºC	65	20
Outlet temperature,ºC	55	30
Pressure drop limitations,kPa	140	5
Total mass flow rate,kg/s	2.5	–

For hand calculations, assume the length of heat exchanger is 3m. Assume nominal pipe sizes of 3.5’’ and 2’’ as outer and inner tubes, respectively.

Select the proper fouling factors. The geometrical information is provided to initiate the analysis. These will be taken as variable parameters to come up with a suitable design; one can start with one inner tube and complete the hand calculations by then checking geometrical parameters to determine the effects of these changes on the design. The final report will include material selection, economical analysis (both hand and software analyses), thermo-hydraulic calculations (both hand and software), mechanical design, and technical drawings of the components and the assembly.

Calculate:

a) The velocities in the tube and in the annulus
b) The overall heat transfer co-efficient for a clean and fouled heat exchanger
c) The total heat transfer area of the heat exchanger with and without fouling (OS design)
d) The surface area of a hairpin and the number of hairpins
e) Pressure drop inside the tube and in the annulus
f) Pumping powers for both streams
g) Technical drawings.

Hand Calculation:

Using the correlations [1], the results are generated from MS Excel by manual programming. After several iteration, the selected design calculations are showing below:

<u>Hot Fluid: SEA-30 oil (Tube)</u> <u>Cold Fluid: Sea-Water (Annulus)</u>

$T_{h,i} = 65^oC$	$T_{c,i} = 20^oC$
$T_{h,o} = 55^oC$	$T_{c,o} = 30^oC$
$T_{avg,h} = 60^oC$	$T_{avg,c} = 25^oC$
$\dot{m}_h = 2.5$ kg/s	$\dot{m}_c = 1.27669$ kg/s
$\rho_h = 864.03$ kg/m^3	$\rho_c = 1023$ kg/m^3
$C_{p,h} = 2047.3$ J/kgk	$C_{p,c} = 4009$ J/kgk
$\mu_h = 0.07445$ Pa.s	$\mu_c = 0.00097$ Pa.s
$k_h = 0.1404$ W/mk	$k_c = 0.607$ W/mk
$Pr_h = 1081.4$	$Pr_c = 6.14$
$C_h = \dot{m}_h C_{p,h}$	$C_c = \dot{m}_c C_{p,c}$
$= (2.5\times2047.3)$ W/k	$= (1.27669\times4009)$ W/k
$= 5118.25$ W/k	$= 5118.25$ W/k

Assume, 5×3 std type K copper tubing is selected

ID_a = 4.805"

= 0.122047m

ID_t = 2.907"

= 0.075717m

OD_a = 4.125"

= 0.130175m

OD_t = 3.125"

= 0.079375m

L = 5m,

$T_w = \frac{1}{2}\left(\frac{65+55}{2} + \frac{20+30}{2}\right) = 42.5^oC,$

μ_w = 0.1914 Pa.s (for oil)

$A_a = \frac{\pi}{4}(ID_a^2 - OD_t^2)$

$= \frac{\pi}{4} \times (0.122047^2 - 0.079375^2)\ m^2$

$= 0.006751 m^2$

$A_t = \frac{\pi}{4} ID_t^2$

$= \frac{\pi}{4} \times (0.075717^2)\ m^2$

$= 0.0045 m^2$

$V_a = \frac{\dot{m}_c}{\rho_c \times A_a}$

$= \frac{1.27669}{1023 \times 0.006757}$

= 0.184871 m/s

$V_t = \frac{\dot{m}_h}{\rho_h \times A_t}$

$= \frac{2.5}{864.03 \times 0.0045}$ m/s

= 0.642583 m/s

$De_a = \frac{{ID_a}^2 - {OD_t}^2}{OD_t}$

$= \frac{0.12204^2 - 0.079375^2}{0.079375}$ m

= 0.108284m

$Dh_a = ID_a - OD_t$

= (0.1220-0.079375)m

= 0.042672m

$$\mathrm{Re_a} = \frac{\rho_c \times V_a \times De_a}{\mu_c} = \frac{1023 \times 0.18487 \times 0.108284}{0.07445} = 8319.86 \text{ (turbulent)}$$

$$\mathrm{Re_t} = \frac{\rho_h \times V_t \times ID_t}{\mu_h} = \frac{864.03 \times 0.64258 \times 0.075717}{0.07445} = 564.6622 \text{ (laminar)}$$

$$\mathrm{Nu_a} = 1.86(Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$= 1.86\left(\frac{De_a Re_a Pr_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$= 1.86 \times \left(\frac{0.108284 \times 8319.9 \times 1081.4}{5}\right)^{1/3} \times \left(\frac{0.00097}{0.1914}\right)^{0.14} = 65.02613$$

$$\mathrm{Nu_a} = \frac{h_a De_a}{k_c}$$

$$\therefore\ \mathrm{h_a} = \mathrm{Nu_a} \times \frac{k_c}{De_a} = 65.03 \times \frac{0.607}{0.108284}\ \mathrm{W/m^2K}$$

$$= 364.5108\mathrm{W/m^2K}$$

$$\mathrm{Nu_t} = 0.023 \times Re_t^{\,0.8} \times \mathrm{Pr_h}^{0.3} \text{ (cooling)}$$

$$= 0.023 \times 564.6622^{0.8} \times 1081.4^{0.3} = 30.97313$$

$$\mathrm{Nu_t} = \frac{h_t ID_t}{k_h}$$

$$\therefore\ \mathrm{h_t} = \mathrm{Nu_t} \times \frac{k_h}{ID_t}$$

$$= 30.97313 \times \frac{0.1404}{0.075717}\ \mathrm{W/m^2K}$$

$= 57.43234\ W/m^2K$

Starting with $N_t = 1$

$A_o = N_t \pi OD_t L = (1\times\pi\times0.079375\times5)\ m^2 = 1.246823m^2$

$A_t = A_o = 1.246823m^2$

As there is no finned surface in the inner tube,

$\therefore \eta_f = 0$

$\therefore$ The overall surface efficiency, $\eta_o = 1$ [3]

Fouling resistance:

$R_{f,o} = 0.00088\ m^2K/W$ $R_{f,i} = 0.00035\ m^2K/W$

For 316 Stainless Steel: k=15W/mK

The overall heat transfer co-efficient (fouled) is:

$$U_{of} = \frac{1}{\frac{d_o}{d_i h_i}+\frac{d_o R_{fi}}{d_i}+\frac{d_o ln\left(\frac{d_0}{d_i}\right)}{2k}+\frac{R_{fo}}{\eta_o}+\frac{1}{\eta_o h_o}}$$

$$= \frac{1}{\frac{0.079375}{0.075717\times57.43234}+\frac{0.079375\times 0.00035}{0.075717}+\frac{0.079375 ln\left(\frac{0.079375}{0.075717}\right)}{2\times15}+\frac{0.00088}{1}+\frac{1}{364.511}}$$

$= 44.70669399\ W/m^2K$

<u>Under Clean Condition:</u>

$$U_{oc} = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o ln\left(\frac{d_0}{d_i}\right)}{2k} + \frac{1}{\eta_o h_o}} = \frac{1}{\frac{0.079375}{0.075717 \times 57.43234} + \frac{0.079375 \times ln\left(\frac{0.079375}{0.075717}\right)}{2 \times 15} + \frac{1}{364.511}}$$

$$= 47.346\ W/m^2K$$

Cleanliness Factor, CF,

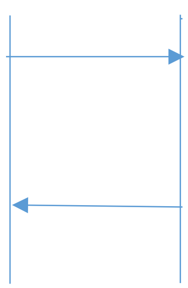
$$CF = \frac{U_{of}}{U_{oc}} = \frac{44.71}{47.346} = 0.944$$

$Q = \dot{m}_h C_{p,h} \Delta T_h = 2.5 \times 2047.3 \times 10 = 51182.5W$

Assume Counter flow,

$T_{h,i} = 65^oC$ $\qquad$ $T_{h,o} = 55^oC$ $\qquad$ $\Delta T_{lm} = \Delta T1 = \Delta T2$

$= 35^oC$



$T_{c,o} = 30^oC$ $\qquad$ $T_{c,i} = 20^oC$

Without fouling,

$$A_{oc} = \frac{Q}{U_{oc}\Delta T_{lm}} = \frac{51182.5}{47.346\times35} = 30.8866\text{m}^2$$

With fouling,

$$A_{of} = \frac{Q}{U_{of}\Delta T_{lm}} = \frac{51182.5}{44.71\times35} = 32.71003\text{m}^2$$

Total Area of Hairpin, $A_{hp} = A_o = 1.246823\text{m}^2$

Without fouling,

Total number of hairpins, $N_{hp} = \frac{A_{oc}}{A_{hp}} = \frac{30.8866}{1.246823} = 24.7723 \approx 25$

With fouling,

Total number of hairpins, $N_{hp} = \frac{A_{of}}{A_{hp}} = \frac{16.2}{1.0174} = 26.23471 \approx 26$

Pressure Drop Calculations:

In Pipe:

$k = \frac{OD_t}{ID_a} = \frac{0.079375}{0.122047} = 0.650364$

$\frac{1}{\varepsilon_{corr}} = \frac{1+k^2}{(1-k)^2} + \frac{1+k}{(1-k)lnk} = \frac{1+0.65^2}{(1-0.65)^2} + \frac{1+0.65}{(1-0.65)ln0.65} = 0.66871$

$\therefore \varepsilon_{corr} = 1.4954$

Friction factor, $f_p = \varepsilon_{corr}\,[64/Re_p]$

$$= 1.4954\times\frac{64}{564.6622} = 0.113$$

$\Delta P_p = f_p \frac{L}{ID_p} \frac{1}{2} \rho_p {V_p}^2$

$= 0.113 \times \frac{5}{0.042672} \times \frac{1}{2} \times 864.03 \times 0.642583^2$

$= 33.07408$KPa

Pumping Power:

$W_p = \frac{\Delta P_p \dot{m}_h}{\eta_a \rho_h} = \frac{33.07 \times 10^3 \times 2.5}{0.8 \times 864.03} = 119.6$W

In Annulus:

Friction factor, $f_a = (1.82 log_{10} Re_a - 1.64)^{-2}$

$= (1.82 log_{10}(8319.858) - 1.64)^{-2}$

$= 0.033$

$\Delta P_a = \left(f_a \frac{L}{Dh_a} + 1\right)\left(\frac{1}{2} \rho_a {V_a}^2\right)$

$= \left(0.033 \times \frac{5}{0.042672} + 1\right) \times \left(\frac{1}{2} \times 1023 \times 0.184871\right)$

$= 2.113808$KPa

Pumping Power:

$W_a = \frac{\Delta P_a \dot{m}_c}{\eta_a \rho_a} = \frac{2.113808 \times 10^3 \times 1.27669}{0.8 \times 1023} = 3.3$W

			m
Dep	Dhp	IDp	0.075717
		ODp	0.079375
Ap	0.004503	IDh	0.122047
Aa	0.006751	ODh	0.130175
Ao	1.246823	Dha	0.042672
L	5	Dea	0.108284
k	0.650364	Inner tube property	
ϵ	0.66871	k	15

muw	0.1914

Rw	0.00088
Ro	0.00035

Properties

Oil		Water	
rho	864.03	rho	1023
Cp	2047.3	Cp	4009
mu	0.07445	mu	0.00097
k	0.1404	k	0.607
Pr	1081.4	Pr	6.14

Sea Water		SAE 30 oil	
Ti	Te	Ti	Te
20	30	65	55
m	1.27669	m	2.5

Q	51182.5	Tw	42.5

Selected	V	Re	Nu	h	Uo	Uo, fouled
Water	0.184871	8319.858	65.02613	364.5108	47.346	44.70669399
Oil	0.642583	564.6622	30.97313	57.43234		

f	ΔP
0.033122729	2.113808
0.113342101	33.07408

	New	Fouled
A	30.8866	32.71003
Nhp	24.7723	26.23471

Selected Design:

HTRI	**Output Summary**					Page 1
	Released to the following HTRI Member Company:					
	BUET					
	Ruvo					

Xhpe E Ver. 6.00 12/16/2020 0:24 SN: Vals100+ **SI Units**

: Summary Unit
Simulation - Horizontal Hairpin With No Baffles

No Data Check Messages.
No Runtime Messages.

Process Conditions		**Cold Shellside**		**Hot Tubeside**	
Fluid name			Water		Oil
Flow rate	(kg/s)		1.2245		2.5000
Inlet/Outlet Y	(Wt. frac vap.)	0.000	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	20.00	30.00	65.00	55.00
Inlet P/Avg	(kPa)	120.002		230.003	178.183
dP/Allow.	(kPa)	3.660	40.000	103.641	140.000
Fouling	(m2-K/W)		0.000350		0.000879

Exchanger Performance

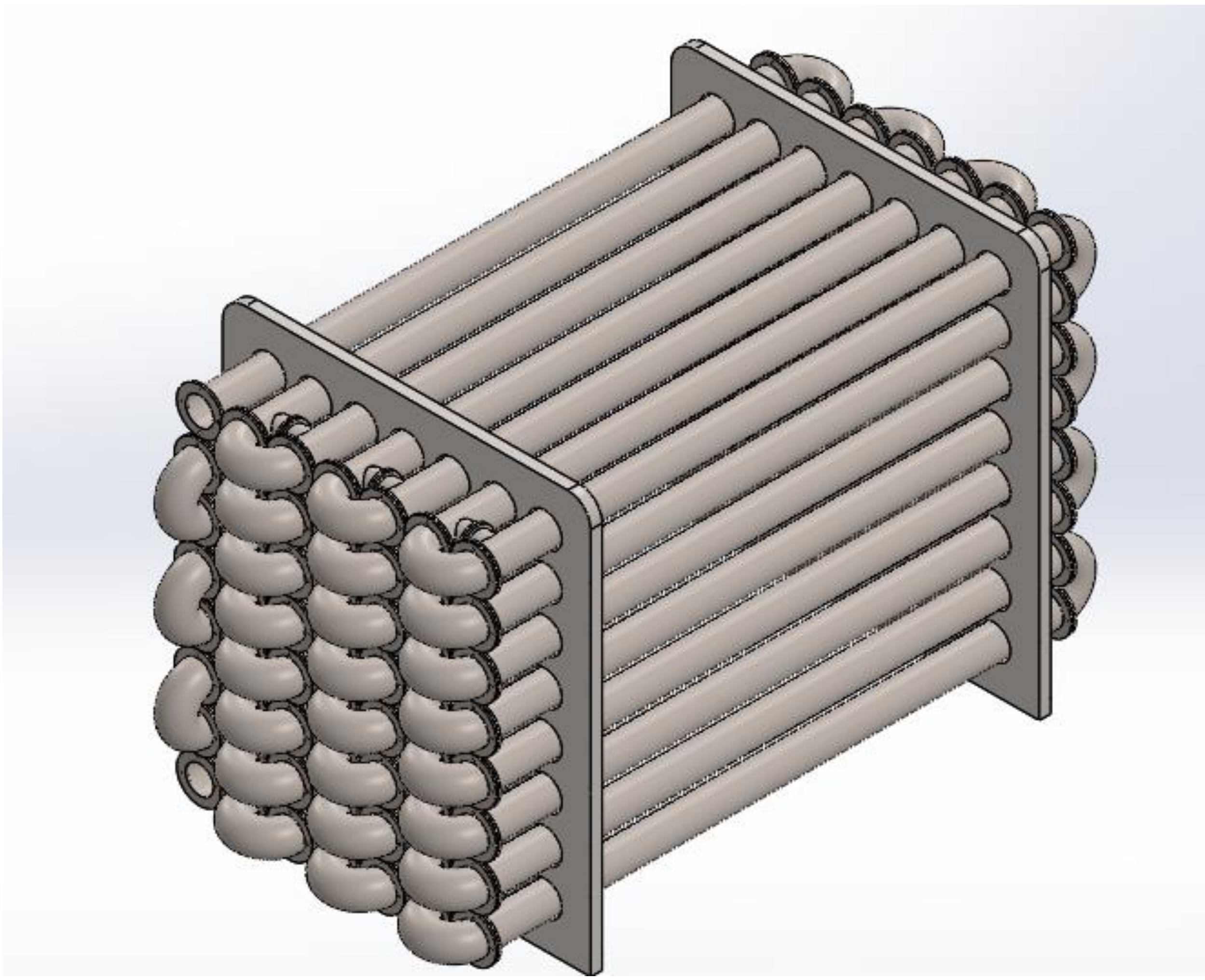
Shell h	(W/m2-K)	781.93	Actual U	(W/m2-K)	40.16
Tube h	(W/m2-K)	49.41	Required U	(W/m2-K)	39.53
Hot regime	(--)	Sens. Liquid	Duty	(MegaWatts)	0.0503
Cold regime	(--)	Sens. Liquid	Area	(m2)	35.986
EMTD	(Deg C)	35.2	Overdesign	(%)	1.61

Shell Geometry			**Baffle Geometry**		
Type	(--)	Hairpin	Baffle type	(--)	None
Shell ID	(mm)	122.047	Baffle cut	(Pct Dia.)	
Series	(--)	28	Baffle orientation	(--)	
Parallel	(--)	1	Central spacing	(mm)	5153.96
Orientation	(deg)	0.00	Crosspasses	(--)	1

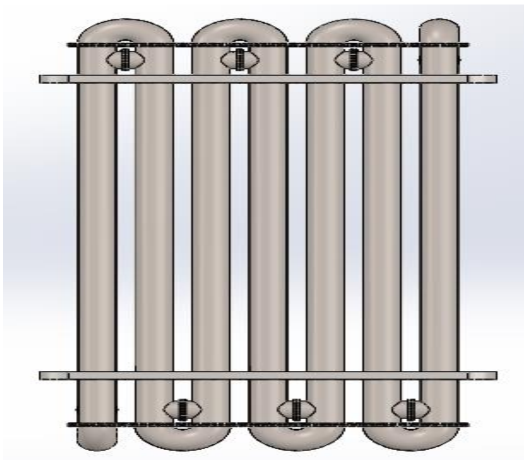
Tube Geometry			**Nozzles**		
Tube type	(--)	Plain	Shell inlet	(mm)	102.261
Tube OD	(mm)	79.375	Shell outlet	(mm)	102.261
Nominal length	(m)	2.500	Inlet height	(mm)	21.336
Pitch ratio	(--)		Outlet height	(mm)	21.336
Layout	(deg)	30	Tube inlet	(mm)	77.927
Tubecount	(--)	1	Tube outlet	(mm)	77.927

Thermal Resistance, %	**Velocities, m/s**		**Flow Fractions**	
Shell	Shellside	0.18	A	
Tube	Tubeside	0.71	B	1.000
Fouling	Crossflow		C	0.000
Metal	Window	0.18	E	
			F	

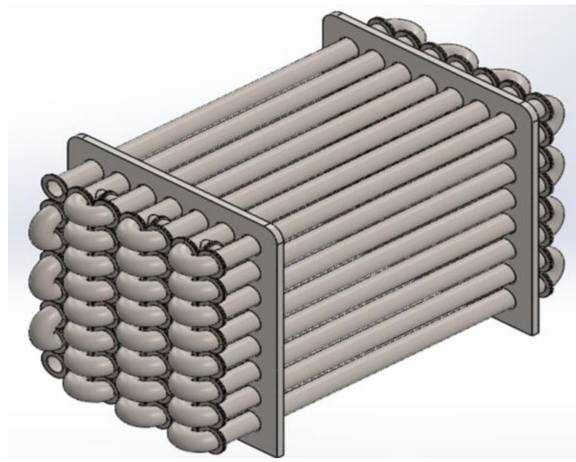
Technical Drawing:



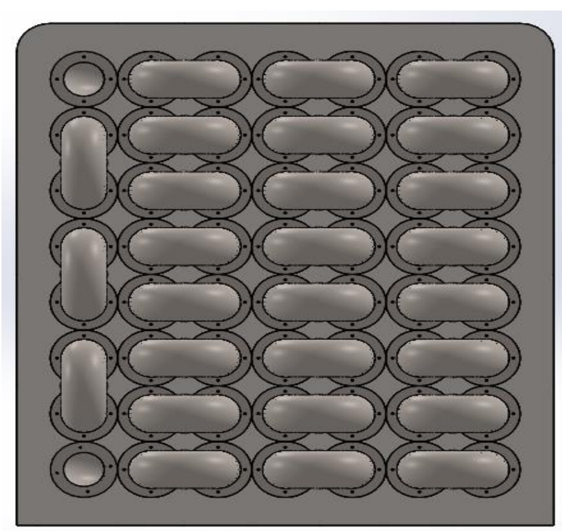
ISOMETRIC VIEW



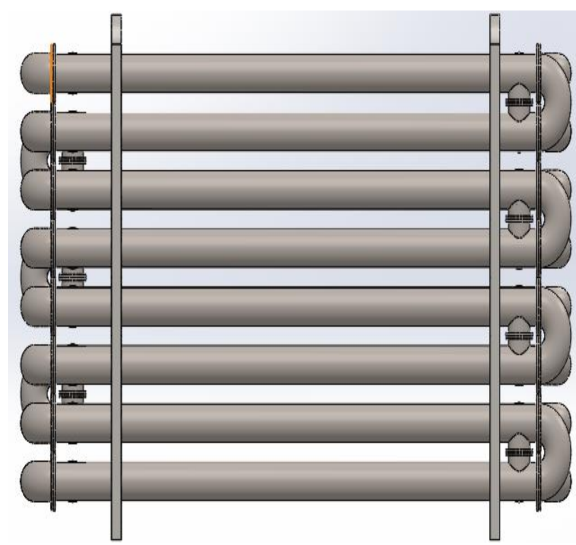
TOP VIEW



ISOMETRIC VIEW

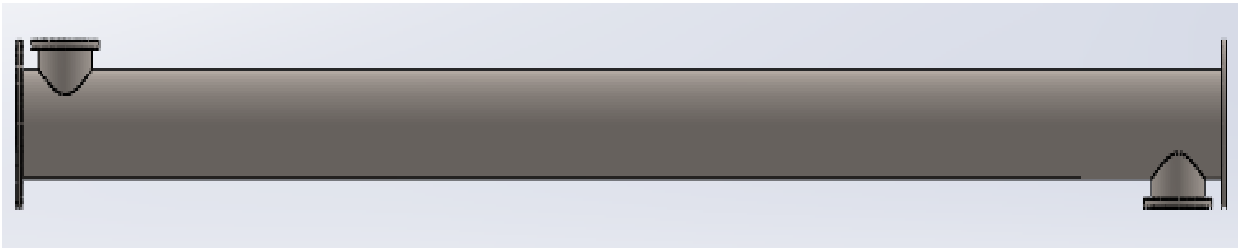


FRONT VIEW

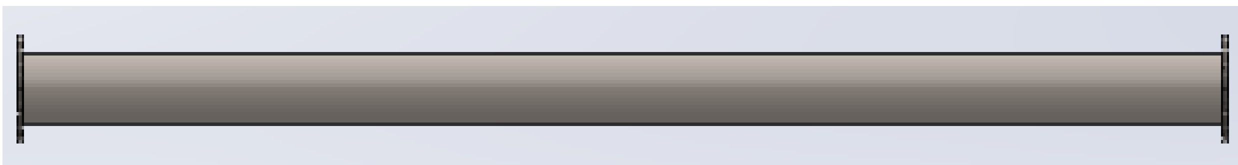


R.H.S. VIEW

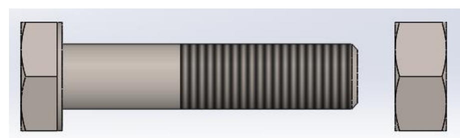
Parts:



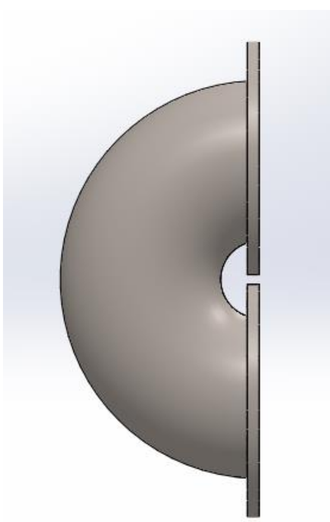
OUTER PIPE (5”)



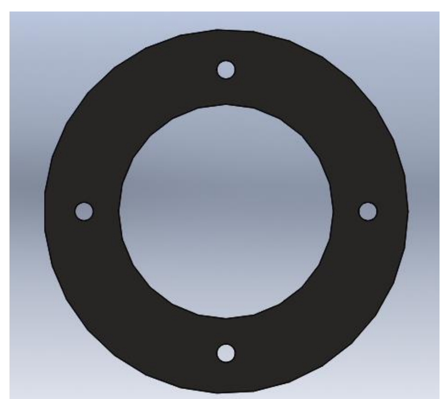
INNER PIPE (3”)



NUT and BOLT (M6)

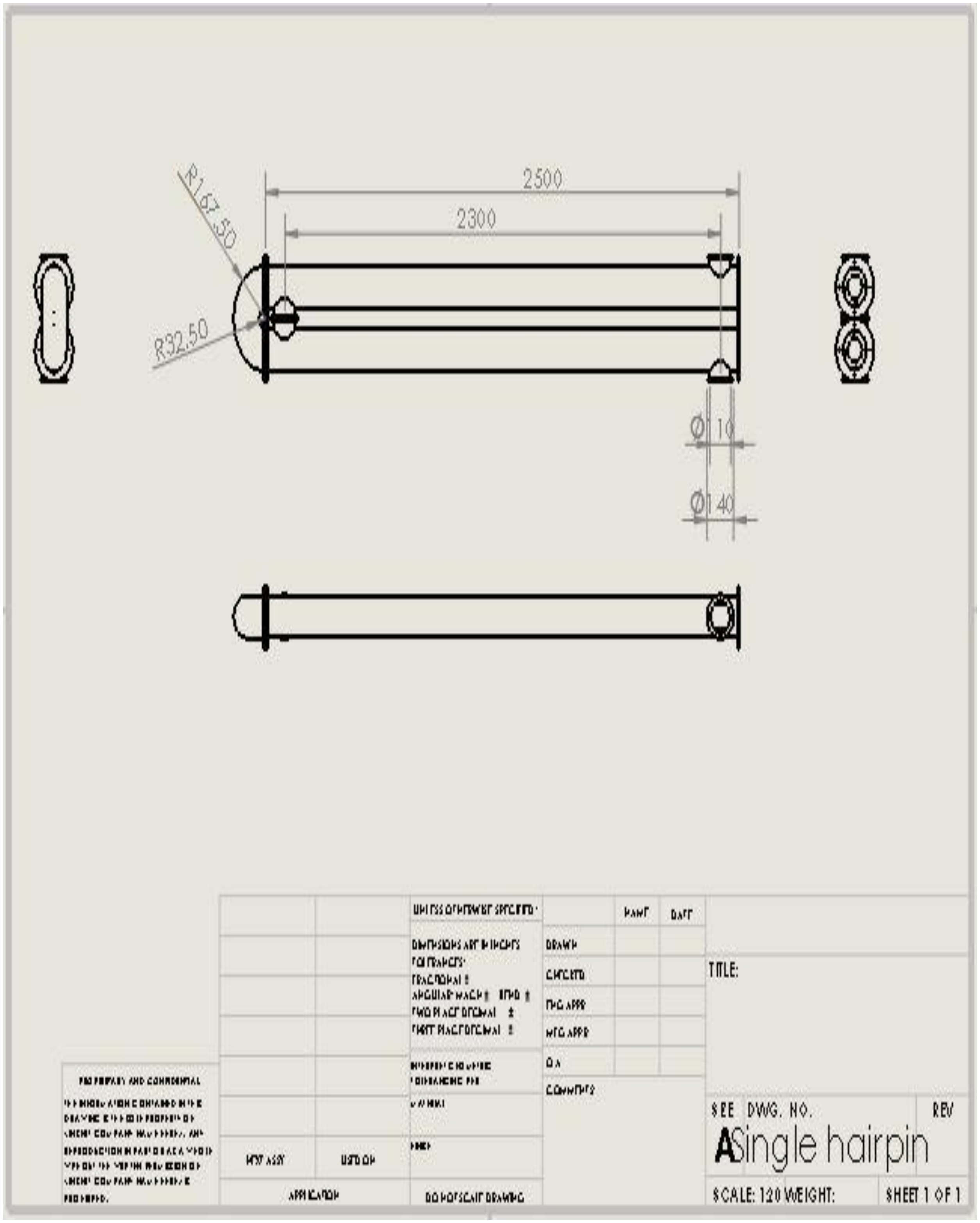


BENT



WASHER

Dimensions:



2500
2300
R167.50
R32.50
Ø110
Ø140
TITLE:
SIZE DWG. NO.
REV
A Single hairpin
SCALE: 1:20 WEIGHT:
SHEET 1 OF 1
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL

Mechanical Design:

Used material for both pipes = 316 Stainless Steel

Ultimate tensile strength = 480 MPa

Outer pipe,

Nominal diameter = 5 inch

Outer diameter = 5.125 inch = 130.175 mm

Inner diameter = 4.805 inch = 122.047 mm

Thickness = 8.128 mm

Maximum pressure(inlet) = 120kPa

Inner pipe,

Nominal diameter = 3 inch

Outer diameter = 3.125 inch = 79.375 mm

Inner diameter = 2.981 inch = 75.375 mm

Thickness = 3.658 mm

Maximum internal pressure, p_i = 230 kpa

Maximum external pressure, p_0 = 120kpa

Tangential Stress = $\frac{{r_i}^2 p_i}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right)$

Pressure(KPa)	σ_t(outer)	σ_r(outer)	σ_t(inner)	σ_r(inner)
$r = r_0$	276.1	0	2019.23	0
$r = r_i$	295.1	-19	2129.23	-110

For both pipes the resulting stresses are very low compared to the strength of the material. Pipes diameters are chosen from the available sizes in the market.

So, the selected design is mechanically okay.

Economic Analysis:

Number of hairpins = 28

Each hairpin length = 2.5m

Total length = 140 m

Total weight of the pipes with bent = 5.1 tons

Average price for both 5” and 3” pipes = 46 750 BDT [2]

So, total price of the pipe of heat exchanger = 5.1 x 46750 BDT

= 238 425 BDT

Average price for 0.5 hp pump with 80% efficiency =1600 BDT

Average price for .008 hp pump with 80% efficiency =200 BDT

Final total cost for the heat exchanger system to fulfill the requirements =238 425 BDT+1600 BDT+200 BDT

=241000 BDT (approx.)

Comparisons from HTRI Simulation:

The results are based on selected number of hairpin (28)

Inner Diameter	Outer diameter	Length	Uo	ΔP, hot	ΔP, cold	Over Design	Remarks
2"	3"	1.5	68.1	417.558	36.359	-33.31	High Pressure drop, Under design
		2	61.88	545	40	-18.64	High Pressure drop, Under design
		2.5	59.12	674.43	44	-4.93	High Pressure drop, Under design
2"	3.5"	1.5	66.7	407	17	-34.18	High Pressure drop, Under design
		2	61.7	535	19	-19.74	High Pressure drop, Under design
		2.5	58.07	636.4	20.67	-6.2	High Pressure drop, Under design
2.5"	3.5"	1.5	59.67	156.416	33.161	-30	High Pressure drop, Under design
		2	52.19	205.121	36.25	-15.68	High Pressure drop, Under design
		2.5	39	253	39	1.47	High Pressure drop
3"	4"	1.5	48.69	74	30	-28.49	Under Design
		2	44.83	97.5	33.26	-13.06	Under Design
		2.5	42.08	120	36	-1.2	Under Design
3"	5"	1.5	41	76	12	-27	Under Design
		2	43	99	13	-14	Under Design
		2.5		103.641	3.66	1.61	Accepted

Material and Cost Comparison:

316 Stainless Steel metal thermal conductivity = 15 W/m^2K

Cost = 46750 BDT/ton

Other available material Copper = 400 W/m^2K

Cost = 85000 BDT/ton

The thickness of tube is 3.658 mm which can be considered as thin tube and so the the effect of thermal conductivity is almost zero for this problem.

Rather steel is comparatively cheaper than copper. With copper the construction cost of the heat exchanger would be almost doubled.

With reference to Alibaba.com (an international manufacturing products supplier), the available heat exchanger with heat duty of 50KW has price ranged from 170 000 to 680 000 BDT.

Our design is economic according to the market price.

Final Result Summary:

Shell diameter = 5 inch

Tube diameter = 3 inch

Nominal length of hairpin = 2.5m

Effective length = 5.313 m

Heat Transfer Area = 35.986

Number of hairpin = 28

Overall Heat transfer coefficient = 40.16

Over design = 1.61%

Conclusion:

By selecting proper material and by formulating and computing variables and parameters, we have designed the cooling system efficiently by maintaining given limitations. The thermo-fluid analysis have been performed and compared for different sets of pipe diameter and length to obtain the main purpose of the cooling system. The economic analysis is performed to reduce the cost of the project. Hence this whole cooling system is both thermally and economically efficient to perform the cooling of oil by water with proper requirements. This heat exchanger can be chosen as an efficient oil cooler in an industry with the given specification in the problem statement.

References:

1. Class Materials L-3/T-2 by Prof. Dr. Zahurul Haq, Dept. ME, BUET

2. Alibaba.com (Material costing)

3. Heat Exchanger Selection, Rating and Thermal design (third edition), Sadik kakac, Hongtan Liu, Anchasa Pramuanjaroenkij