MECHANICAL MODELLING AND ANALYSIS OF HEAT EXCHANGER WITH BAFFLES TO DESIGN A COMPACT HEAT EXCHANGER

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ABSTRACT:

The materials used in the construction of the heat exchanger may be an important consideration in the selection of heat exchanger. A temperature difference of 50°C or more between the tubes and the shell will probably pose differential thermal expansion problems and the need to be considered. The case of corrosive fluids, corrosive-resistant materials such as stainless steel can be selected. The heat transfer characteristics of a mantle heat exchanger with a single pass flow arrangement are investigated under controlled indoor conditions. Measurements showed that the tank is well mixed above the mantle level of the heat exchanger. The influence of mantle inlet location, the mantle fluid flow rate and the type of mantle fluid on the flow and heat transfer coefficients in a mantle is investigated using experimental results. The mechanical modelling of the heat exchanger will give the optimum result of efficiency heat transfer. An industrial purposed heat exchanger with nearly 65⁰c practically observed to model compact heat exchanger and the efficiencies has been taken as per requirement.

***Keywords**: shell and Tube exchangers with baffles, performance of heat exchanger.Temperature distribution, analytical approach*

1.0 INTRODUCTION:

Heat exchangers are equipment that transfers heat from one medium to another. The proper design, operation and maintenance of heat exchangers will make the process energy efficient and minimize energy losses. Heat exchanger performance can deteriorate with time, off design operations and other interferences such as fouling, scaling etc. It is necessary to assess periodically the heat exchanger performance in order to maintain them at a high efficiency level. This section comprises certain proven techniques of monitoring the performance of heat exchangers, coolers and condensers from observed operating data of the equipment.

2.0 OBJECTIVES:

1. To determine the overall heat transfer coefficient for assessing the performance of the heat exchanger. Any deviation from the design heat transfer coefficient will indicate occurrence of fouling.

2.To determine the heat duty (amount of energy to be transferred), temperature changes within the exchanger, and pressure drops.

3.0 MECHANICAL DESIGN

Design Data:	Shell side	Tube side
Design pressure	12.7kg/cm^2	6kg/cm^2
Corrosion allowance	3mm	3mm

Shell Thickness: As per UG-27C (1) of ASME Section VIII DivI

Material Specification: CARBON STEEL PIPE(SA516Gr70)

The thickness of the shell is calculated using O.D based formulae,

$t = [(P_s*R)/(S*E-0.6*P_s)]+C.A$

Where

t = Thickness of the shell (mm)

P_s = Design Pressure = 12.7 kg/cm^2

R = Inside radius of the shell in consideration (mm) =(Inside dia of shell)/2

= (307)/2 = 153.5mm = 15.35 cm

S = Allowable stress (As per ASME Section II part D)

= 1400 kg/cm^2

E = Joint Efficiency = 1(since tube is seamless)

Corrosion allowance = 3.2 mm

$t = [(12.7*153.5)/((1400*1)-0.6*12.7)]+3.2 = 4.6$ mm

Provided Thickness = 8.4 mm

As per TEMA, clause RCB-3 required thickness of the shell should be 4.8mm

The required thickness is less than the provided thickness.

Hence the design is acceptable.

Thickness Of The Channel: As per UG-27C (1) ASME Section VIII Div I

Material Specification: Carbon Steel Plate (SA516GR70)

Thickness of the channel is given by

$t = [(P*R)/((S*E)-(0.6*P))]+C.A$

Where

t = Thickness of the channel (mm)

P = Design Pressure (kg/cm^2) = 6kg/cm^2

R = Inside radius of the shell (mm) = (15.75)/2 = 7.875mm

S = Allowable Stress (kg/cm^2) = 1400 kg/cm^2

E = 1(take seamless tube)

$t = [(6*7.875)/((1400*1)-(0.6*6))]+3 = 3.033$ mm

Thickness Of Tube:

Under Internal Pressure: Appendix1 of ASME Section VIII Div I

Material specification: admirality brass (SB111C44300)

The thickness of the tube is given by $t = (P*R)/[(S*E)-(0.6*P)]$

Where

t = Thickness of the tube (mm)

P = Design Pressure (kg/cm^2)

R = Inner Radius of tube (mm) =(15.75)/2 = 7.875 mm

S = Allowable Stress (kg/cm^2) = 10,000 psi =703.07 kg/cm^2

E = Joint Efficiency = 1(take seamless tube)

$t = (6*7.8785)/[(703.07*1)-(0.6*6)] = 0.06755$mm

Provided thickness=1.65mm

The required thickness is less than the provided thickness.

Hence the design is acceptable

Nozzle Wall Thickness

As per UG45,34 (1) ASME Section VIII DIV 1

t_{nw} = (P*R)/[(S*E)-(0.6*P)]+C.A

Where

t = Thickness of the nozzle wall (mm)

P = Design Pressure (kg/cm^2)

R_n= Nozzle Radius (mm) =73/2 =36.5mm

S = Allowable Stress (kg/cm^2) = 17000 psi

=17000*0.70307 kg/cm^2 =1202.2497 kg/cm^2

t_{nw} = (12.7*36.5)/[(1202.2497*1)-(0.6*12.7)]+3 =3.388mm =0.3388cm

Nozzle Neck Thickness

As per UG 45(a) ASME Section VIII DIV I

t_n = (P*R_n)/[(S*E)-(0.6*P)]

Where

t_n = Nozzle neck thickness (mm)

P = Design Pressure (kg/cm^2)

R_n = Nozzle Radius (mm) =(73/2) = 36.5 mm

S = Allowable Stress (kg/cm^2) = 1400 kg/cm^2

E = Joint Efficiency = 0.85

t_n = (12.7*36.5)/[(1400*0.85)-(0.6*12.7)]+3 =3.392mm

As per UG45 (b)1

For vessel under internal pressure

If E = Joint Efficiency = 1

then,t_n=3.3329mm

so from the above two values of t_n the larger will be considered ∴ t_n=3.392mm

Tube Sheet Thickness

FORMULA FOR SHEARAs per RCB-7(TEMA)

$$T = \frac{0.31 D_L (P/S)}{[1-(do/pitch)]}$$

T= Effective tube sheet thickness

D_L=4A/C=Equivalent diameter of tube center limit perimeter.

<u>Shear will not control when</u>

P/S<1.6[1-(do/pitch)]^2

Do=19.05mm

P_t=25.4mm

P= 6 kg/cm^2

S=703.07 kg/cm^2

∴The condition is satisfying.

FORMULA FOR BENDING:

$$T = \frac{FG}{3}\sqrt{\frac{P}{\eta S}}$$

where

$\eta=1-\frac{0.907}{(p_t/d_o)^2}$ (for triangular or rotated triangular tube patterns) $=48.98\%$

F=1 (from FIGURE RCB-7.132 SECTION-5)

G= 307+gasket central diameter$\pm$10 =342mm

$$T=\frac{1*34212.7^2*10}{3\sqrt{48.98*1400}}+2*3$$

T=23.88mm

Blank Plate

$$T=d\sqrt{\frac{CP}{SE}+\frac{1.9Wh_G}{Sed^3}}$$

C=0.3 (from UG34)

W= for gasket seating $=(A_m+A_b)*(S_a/2)$

$A_{m1}=W_m/S_b$

$A_{m2}=W_{m2}/S_a$

$W=H+H_p$ $=0.785G^2P+2b*3.14G_mP$

b= gasket width=10mm

$W_{m1}=H+H_p m=2$

$y=1600psi=112.49*10^{-2}\ kg/cm^2$

$W_{m2}=3.14bGy$ = $3.14*10*342*112.49*10^{-2}$

=12080.0761mm

$W_{m1}=(0.785*342^2*(12.7*10^{-2}))+[2(10)*3.14*342*12*10^{-2}]$

=66213.8299mm

The larger of the two is considered.

$\therefore$ W=66213.8299mm

$$h_g=\frac{C-G}{2}$$

Where

C= Bolt circle diameter =408mm

$$h_g=\frac{408-352}{2}$$

=28

$$t=352\sqrt{\frac{0.3*12.7}{1400*1}+\frac{1.9*66213.82*28}{1400*10^{-2}*352^3}}$$

t=32.4mm

Nozzle Reinforcement

Without reinforcement element

A $=dt_rF+2t_nt_rF[1-f_{r1}]$

Area required

$= d(E_1t-Ft_r)-2t_n(E_1t-Ft_r)(1-f_{r1})$

Area available in shell;

$A_1=$

use larger value

$=2(t+t_n)(E_1t-Ft_r)-2t_n(E_1t-Ft_r)(1-f_{r1})$

$=5(t_n-t_{rn})f_{r2}t$

Area available in nozzle

$A_2=$

projecting outward;

$=5(t_n-t_{rn})f_{r2}t_n$

use smaller value

$A_3 = 5tt_if_{r2}$
Area available in inward
Nozzle; use smallest value

$= 5t_it_if_{r2}$

$= 2ht_if_{r2}$

A_{41}=outward nozzle weld=$(leg)^2f_{r2}$
Area available in outward weld

A_{43}=inward nozzle weld=$(leg)^2f_{r2}$
Area available in inward weld

If $A_1+A_2+A_3+A_{41}+A_{43} \geq A$
Opening is adequately reinforced

If $A_1+A_2+A_3+A_{41}+A_{43} < A$
Opening is not adequately

reinforced so reinforcing elements must be added and/or thickness must be increased

$t_n=t_i=6.392mm$

$T_s=4.849mm$

F=1

d=73mm

$t_r=4.849mm$

$t_{rn}=3.392mm$

t=7.849mm

$E_1=1$

$f_{r1}=f_{r2}=0.8587$
$\{fr1=(S_n/S_v)=(17.1*103*0.070307)/1400=$
0.8587

$A = dt_rF+2t_nt_rF[1-f_{r1}]$
$=(73*4.849*1)+(2*6.392*4.849*1*[1-0.8587])$

$A=362.73mm^2$

$A_1= d(E_1t-Ft_r)-2t_n(E_1t-Ft_r)(1-f_{r1})$

$= 73(1*7.849-1*4.849)-2*6.392(1*7.849-1*4.849)(1-0.8587)$

$=213.5808mm^2$

$A_1=2(t+t_n)(E_1t-Ft_r)-2t_n(E_1t-Ft_r)(1-f_{r1})$

$=2(7.849+6.392)(1*7.849-1*4.849)-2*6.392(1*7.849-14.849)(1-0.8587)$

$=64.41mm^2$

There fore the larger value is considered

$A_1=213.5808mm^2$

$A_2=5(t_n-t_{rn})f_{r2}t$ $=5(6.392-3.392)0.8587*7.849$ $=101.099\ mm^2$

$A_2=5(t_n-t_{rn})f_{r2}t_n$

$=5(6.392-3.392)0.8587*6.392 =82.3321\ mm^2$

Therefore the smaller value is considered

$A_2=82.3321\ mm^2$

$A_3=5tt_if_{r2}$

$=5*7.849*6.392*0.8587$ $=215.408\ mm^2$

$A_3=5t_it_if_{r2}$ $=5*6.392*6.392*0.8587$ $=175.422\ mm^2$

$A_3=2ht_if_{r2}$ [h=2.5t=15.98mm] $=2*15.98*6.392*0.8587$

$=175.422\ mm^2$

Therefore the smallest value is considered**A_3= 175.422 mm^2**

Since excluding the weld area therefore A41 and A43 are neglected

If $A_1+A_2+A_3+A_{41}+A_{43} \geq A$ $\therefore$
$A_1+A_2+A_3+A_{41}+A_{43}=213.5808+82.3321+175.422=471.3349\ mm^2$

$A=362.73mm^2$

$471.3349 \geq 362.73$

There fore the above condition is satisfied and the opening is adequately reinforced.

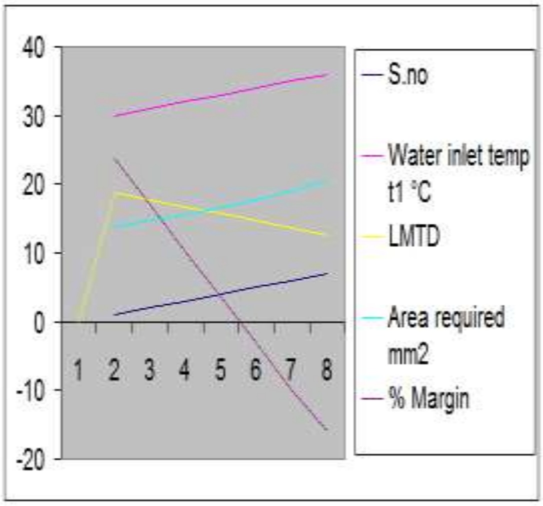
4.0 RESULTS

Table.1 Comparision of pressure drop &overall heat transfer coefficient values with HTRI

S.NO	Pressure drop, Kg/cm^2				Overall heat transfer coefficient, Kcal/hr-m^2/^{0}C		
		Theoretical	HTRI	% Error	Theoretical	HTRI	% Error
1	Shell Side	0.336	0.335	0.29	244.9	249.035	1.68
2	Tube Side	0.2629	0.229	1.4			

Table.2Variation of LMTD and surface area with water inlet temperature

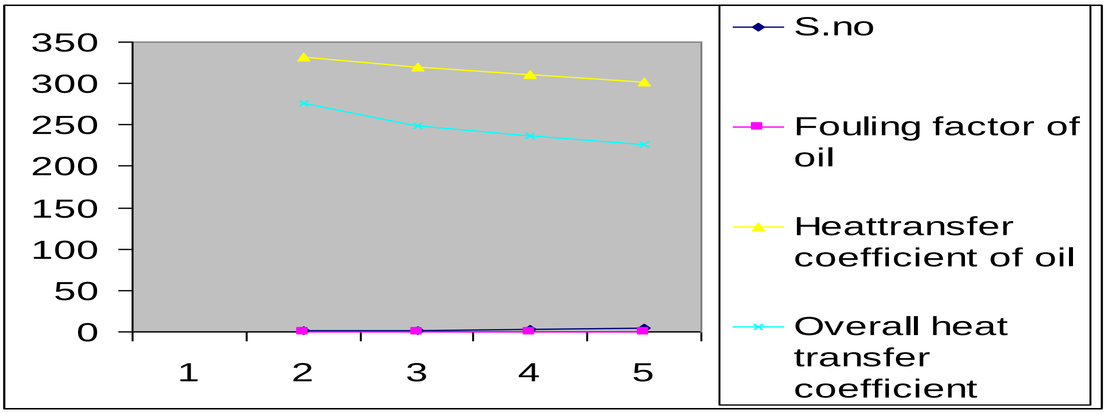
S.no	Water inlet temp t_1 °C	LMTD °C	Area required mm^2	% Margin
1.	30	18.78	13.887	23.73
2.	31	17.76	14.684	17.01
3.	32	16.74	15.57	10.32
4.	33	15.72	16.586	3.59
5.	34	14.7	17.7	-3.10
6.	35	13.67	19.07	-9.90
7.	36	12.64	20.632	-16.0



Variation of LMTD and surface area with water inlet temperature

Table shows Variation of overall heat transfer coefficient with fouling factor of oil

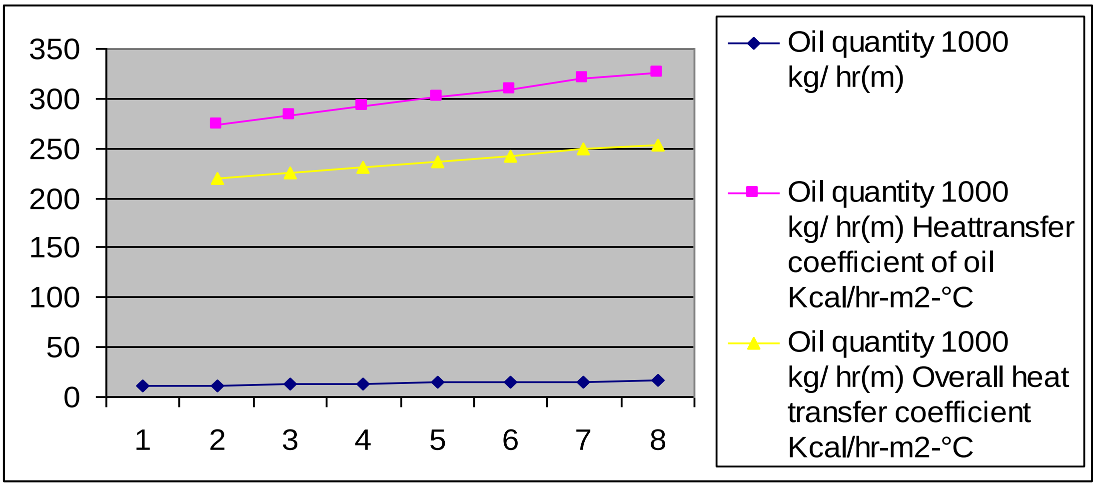
S.no	Fouling factor of oil Hr-m^2-°c/kcal	Heattransfer coefficient of oil Kcal/hr-m^2-°C	Overall heat transfer coefficient Kcal/hr-m^2-°C
1.	0.0001	331.196	276.595
2.	0.0002	320.578	249.035
3.	0.0003	310.620	237.228
4.	0.0004	301.263	226.479



Graph shows Variation of overall heat transfer coefficient with fouling factor of oil

Table shows Variation of heat load and overall heat transfer coefficient with oil quantity

S.no	Oil quantity 1000 kg/ hr(m)	Heat load Q_s Kcal/hr	Reynolds No	Heattransfer coefficient of oil Kcal/hr-m^2-°C	Overall heat transfer coefficient Kcal/hr-m^2-°C
1.	11.0655	48760.513	154.83	273.898	219.971
2.	11.8032	52011.213	165.125	283.207	225.885
3.	12.5409	55261.914	175.472	292.274	231.616
4.	13.2766	58512.615	185.79	301.048	237.092
5.	14.0163	61763.331	196.119	309.572	242.347
6.	14.754	65014	209.873	320.578	249.035
7.	15.4917	68264.71	216.757	325.944	252.267
8.	16.2294	71515.419	227.085	333.831	256.966



Graph shows Variation of heat load and overall heat transfer coefficient with oil quantity

5.0 Conclusions

It can be noticed that, mantle side heat transfer coefficient (hm) value is low (110 w/m2K) for the first 30 minutes from the beginning of the experiment for the top inlet (50 mm below the top of the mantle) and is found to be increasing continuously up to 160 w/m2K between 30 minutes and 150 minutes, where as, the hm value is observed to be almost constant 140 w/m2K for the second inlet i.e., 100 mm below the top of the mantle tank throughout the test duration. In both the cases it is observed that heatexchange is taking place effectively from 0.1 m to 0.5 m height only. The heat transfer coefficients on either side of mantle heat exchanger are noticed to be higher for 40^{0}c 65^{0}c 90^{0}c mantle fluid inlet temperature.

The difference in heat transfer coefficients for 40°C and 65°C inlet temperatures is found to be marginal on mantle side whereas it is considerable on tank side. The tank side heat transfer coefficient, ht values are greater for the second inlet than the first (top) inlet as there is a more heat transfer at the bottom of the mantle tank. It can also be observed that, at higher mantle fluid inlet temperature (65°C), more heat is transferred to the tank contents.

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