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THE DATA ANALYSIS OF COMPACT HEAT EXCHANGER REFERENCE TO SHELL SIDE WITH CONTINUOUS OF HELICAL BAFFLES

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ABSTRACT

In present day shell and tube heat ex-changer is the most common type of heat ex-changer widely used in oil refinery and in other large chemical industries, because it suits high pressure and heat transfer application. The heat ex-changer consists of 7 tubes and each tube has 600 mm length, the shell diameter is 90 mm. The helix angle of helical baffle will be varied from the length 0 to 200mm. On observing the results, it clearly shows that the pressure varies in shell side are due to different helix angle and mass flow rate. The flow pattern in the shell side of the heat ex-changer with continuous helical baffles is forced to be rotational and helical .This is due to the geometry of the continuous helical baffles, thus results in a significant increase in heat transfer coefficient per unit pressure drop in the heat ex-changer.

Key Words: *shell and tube heat ex-changer, compact heat ex-changer, pressure drop*

1.0 PROCESS HEAT TRANSFER – AN INTRODUCTION

The science of thermodynamics deals with the quantitative transitions and rearrangements of energy as heat in bodies of matter. Heat transfer is the science that deals with the rate of exchange of heat between hot and cold bodies called the source and receiver. When one Kg of water is vaporized or condensed, the energy change in either process is identical. However, the rates at which either process is different, vaporization being much more rapid than condensation. The major difference between thermodynamics and heat transfer is that the former deals with the relation between heat and other forms of energy, whereas the latter is concerned with the analysis of the rate of heat transfer. Thermodynamics deals with systems in equilibrium so it cannot be expected to predict quantitatively the rate of change in a process, which results from non-equilibrium states. Heat transfer is commonly associated with fluid dynamics and it also supplements the laws of thermodynamics by providing additional rules to establish energy transfer rates. Process heat transfer deals with the rates of heat exchange as they occur in the heat transfer equipment of the engineering process. This approach brings to better focus the importance of the temperature difference between the source and the receiver, which is, after all, the driving force whereby the transfer of heat is accomplished. A typical problem of process heat transfer is concerned with the quantities of heat to be transferred, the rates at which they may be transferred because of the natures of the bodies, the driving potential, the extent and arrangement of the surface separating the source and the receiver, and the amount of mechanical energy which may be expanded to facilitate the transfer. Since heat transfer involves an exchange in the system, the loss of heat by the one body will equal the heat absorbed by another within the confines of the same system.

1.1 Objectives

1. To calculate the overall surface area to check the optimal area for design of compact heat exchanger.
2. To examine the working parameters for further enhancement of heat transfer rate of the compact heat exchanger design.
3. To compare overall progress of compact heat exchanger design with respect to present running heat exchanger in industry.

1.2 Limitations

1. Industrial heat exchanger taken in to consideration.
2. Hot liquid i.e. oil (liquid Ethanol) in shell side and water taken as tube side cold liquid.

2.0 LITERATURE REVIEW

Heat exchangers are one of the most important devices of mechanical systems in modern society. Most industrial processes involve the transfer of heat and more often, it is required that the heat transfer process be controlled.

OkO (2008) is a device of finite volume in which heat is exchanged between two media, one being cold and the other being hot. There are different types of heat exchangers; but the type widely used in industrial application is the shell and tube. As its name implies, this type of heat exchanger consists of a shell with a bundle of tubes inside it. One fluid runs through the tubes, and another flows over the tubes to transfer heat between the two fluids. The tube bundle may consist of several types of tubes: plain, longitudinally finned, etc.

(Thirumarimurugan et al, 2008) As the two fluids in the heat exchanger that are at different temperatures, heat exchanger analysis and design therefore involve both convection and conduction. Two important problems in heat exchanger analysis are (1) rating existing heat exchangers and (2) sizing heat exchangers for a particular application. Rating involves the determination of the rate of heat transfer, the change in temperature of the two fluids and the pressure drop across the heat exchanger. Sizing involves selection of a specific heat exchanger from those currently available or determining the dimensions for the design of a new heat exchanger, given the required rate of heat transfer and allowable pressure drops.

(Holman, 2004) To ensure that the shell side fluid will flow across the tubes and thus induce higher heat transfer, baffles are installed in the shell to force the shell-side fluid to flow across the tube to enhance heat transfer and to maintain uniform spacing between the tubes.

(Ravi Kumar et al, 1988) There are design charts such as Effectiveness-Number of transfer Unit) curves and LMTD (Logarithm Mean Temperature Difference) correction factor curves for the analysis of simple types of exchangers. Similar design charts do not exist for the analysis of complex heat ex-changers with multiple entries on the shell side and complex flow arrangements.

(Kern, 1965) The thermal analysis of a shell and tube heat ex-changer involves the determination of the overall heat-transfer coefficient from the individual film coefficients; The shell-side coefficient presents the greatest difficulty due to the very complex nature of the flow in the shell. In addition, if the ex-changer employs multiple tube passes, then the LMTD correction factor must be used in calculating the mean temperature difference in the ex-changer. For the turbulent flow regime ($Re \geq 104$), the following correlation is widely used.

3.0THERMAL DESIGN DATA

The heat ex-changer which is supplied to a chemical industry and the data of heat transfer requirement taken from Nikhitha pharma industries - cherlapalley, Hyderabad ,India with the co-ordination of OM-kali engineering works for further set up of arrangements ..

Heat duty =65000 kcal/hr (Input data)

Quantity of water = 50m^3/hr (Assumed)

Quantity of oil = 14.75m^3/hr (Input data)

Water inlet temperature = 33°c (Input data)

Oil outlet temperature = 45°c (Input data)

Allowable pressure drop on water side = 0.6 kg/cm^2 (Input data)

Allowable pressure drop on oil side = 0.6 kg/cm^2 (Input data)

Fouling factor on water side = 0.0004 hr-m^2-°c/kcal

Fouling factor on oil side = 0.0002 hr-m^2-°c/kcal (Input data)

Tube material =Admiralty brass

Thermal conductivity of tube material = 66 BTU/hr-ft$^{2\ 0}$F

Number of tubes = 90

Number of tube passes = 2

Length of tube = 3300mm=3.300 m

Outside diameter of the tube = OD=19.05mm=0.01905m

Thickness of tube = 1.650mm=0.00165m

Inside diameter of tube = OD-2*Thk = 15.75mm=0.01575m

Tube type = Plain type

Tube pitch = 25.4mm=0.0254m

Ratio of outside to inside surface area=$A_o/A_i = \pi d_o L/\pi d_i L$ = 1.2095

Number of baffles = 33

Baffle cut =22%

Type of heat exchanger = Shell and tube AEW type heat exchanger (floating rear tube sheet)

Baffle thickness = 6mm=0.006m

Shell inside diameter = 307mm=0.307m

Shell outside diameter =323.8mm=0.3238m

Shell thickness = 8.4mm=0.0084m

Baffle spacing = 86mm=0.086m

4.0 HEAT EXCHANGER - PROPERTIES

μ_0 = 25.6375cp = 0.0256357 kg/m-sec=92.295 Kg/m-hr

C_{p0} = 0.4663 kcal/kg-°c

K_0 = 0.1295 kcal/m-hr-°c

ρ_0 = 851.85 kg/m^3

4.1 PROPERTIES OF WATER:

μ_w = 0.7496cp = 0.0007496 kg/m-sec=2.6985 kg/m-hr

C_{pw} = 0.9992 kcal/kg-°c

K_w = 0.54275 kcal/m-hr-°c

ρ_w = 992.945 kg/m^3

5.0 THERMAL DESIGN & CALCULATIONS

Heat Load Q: Heat Side (Shell Side)

Q= m*cp*(T_1-T_2)= (14750)*0.4663*(54.45-45)

Q=65684.18 Kcal/hr

Water Side (Tube Side)

$Q = m*cp*(t_2-t_1) = (50000)*0.9992*(34.3-33)$

Q=64948 Kcal/hr

Logarithmic Mean Temperature Difference (LMTD):

$LMTD = (\Delta T_1-\Delta T_2)/Ln\ (\Delta T_1/\Delta T_2)$

$\Delta T_1 = T_1-t_2$

$\Delta T_2 = T_2-t_1$

Where $T_1 = 54.45°c$

$T_2 = 45°c$

$t_1 = 33°c$

$t_2 = 34.30°c$

$\Delta T_1 = 20.15°c$

$\Delta T_2 = 12°c$

LMTD = (20.15-12)/ ln (20.15/12) = 15.724°c

$R= (T_1-t_2)/(t_2-t_1) = 15.5$

$S = (t_2-t_1)/(T_1-t_1)= 0.06$

Correction Factor based on R & S = 0.95

MTD = 0.95*15.724

$=14.94^0{}_C$

Tube Side Heat Transfer Coefficient Water, tube side, cold fluid:

$h_i = 0.023*(K_w/d_i)*(Re)^{0.8}*(Pr)^{0.4}$

$Re = (\rho_w*V_w*d_i)/\mu_w$;

$Pr = (\mu_w*c_p)/K_w$

Water side flow area = I.D cross sectional area of the tube*(No of tubes/No of passes)

$= (\prod/4)*(0.01575)^2*(90/2) = 0.008767\ m^2$

Velocity of water (Vw) = Raw water quantity/Water side flow area

$= 50/(0.008767*3600) = 1.595$ m/sec

Reynolds number (Re) = $(\rho_w*V_w*d_i)/\mu_w$

$=(992.945*1.595*3600*0.01575)/(0.7496*10^{-3}*3600)$

$= 33276.44$

$(Re)^{0.8} = 4146.766$

Prandtl number (Pr) = $(\mu_w*c_{pw})/K_w = (.7496*3600*0.9992*10^{-3})/0.54275$
$= 4.968$

$(Pr)^{0.4} = 1.898$

$h_i = 0.023*(k_w/d_i)*(Re)^{0.8}*(Pr)^{0.4}$

$= 0.023*(0.54275/0.01575)*4146.766*1.898$

$= 6240.66$ Kcal/hr-m^2-°c

Shell Side Heat Transfer Coefficient

Shell side: oil (hot fluid)

$h_{of}=1/((1/h_o)+f_{fo})$

$h_0 = 0.36*(K/D_{eq})*(Re)^{0.55}*(Pr)^{0.33}*(\mu_b/\mu_k)^{0.14}$

$(\mu_b/\mu_k)^{0.14}=(25.63/20.725)^{0.14}=1.03018$

μ_b=Average. Viscosity=((30.550+20.725)/2)=25.63 cP

$\mu_{k=}$ Viscosity at higher temperature=20.725cP

From the triangular pitch and 22% baffle cut

Vertical height = (0.866*P) = 0.866*25.4 = 21.99mm

Average row is at 2nd row from center; number of tubes in this row are 7.

∴ Vertical height = 2*21.99 + 18mm(partition lane clearance from center line)

= 61.98mm

= 0.06198m

Window flow area = Area of segment- Area of the tube in the window portion

Area of segment = Area of sector- Area of triangle inside

Area of sector = $(\prod*di^2*\alpha)/360$

Chord length=C=√((D/2)-h)((D/2)+h) -(no of tubes in middle row) +2(pass partition gap)

C=√((307/2)-61.98)((307/2)+61.98) -(7+(2*18))

C=97.4248mm

Free flow area = Free flow cross length*Baffle pitch

= 97.4248 *86

= 8378.48 mm^2

Velocity on cross pass = Quantity of flow/Free flow area

= $(14754*10^6)/(3600*8378.48*851.85)$

v_0 = 0.5742m/sec

v_o=60% of v_o=0.6*0.5742=0.3445m/sec

D_{eq} = $4*[(1/2*P_t*0.86*P_t)-(1/2*\prod/4*(d_0)^2)]\ /(1/2*\prod*d_0)$ = $4*[(1/2*25.4*0.86*25.4)-(1/2*3.14/4*(19.05)^2)]/(1/2*3.14*19.05)$

= (277.4-142.4)/29.90

= 18.0335mm = 0.0180335m

Reynolds Number Re = $(\rho_0*v_0*D_{eq})/\mu_0$

= $(851.85*0.3445*3600*0.0180335)/25.6375*10^{-3}$

= $209.873(Re)^{0.55}$ = 18.9267

Prandlts Number Pr = $(\mu_0*c_{p0})/K_0$

= $(25.6375*10^{-3}*0.4663)/(0.1295*3600)$

= $332.205(Pr)^{0.33}$ = 6.793

$h_0 = 0.36*(K/D_{eq})*(Re)^{0.55}*(Pr)^{0.33}$

= 0.36*(0.1295/0.0180335)*18.92677*6.793*1.03018

= 342.541 kcal/m^2-hr-°c

$h_{of}=1/((1/h_o)+f_{fo})$

=1/((1/342.541)+0.0002)

h_{of}=320.5787 kcal/m^2-hr-°c

Overall Heat Transfer Coefficient

$1/U = 1/h_{0f}+(1/h_i+$ waterside fouling factor)(A_0/A_i)+oil side fouling factor+Tube wall resistance.

Tube wall resistance=$r_w = (d_0/24K)*\ln(d_0/(do-2*t))$

Where ,

d_o =19.05mm=19.05/25.4 inches

K=66 BTU/hr-ft^2 ^{0}F (From TEMA)

= ((19.05/25.4)/(24*66))*ln ((19.05/25.4)/((15.75/25.4))= $1.844*10^{-5}$ hr-m^2-°c/k cal

/U = (1/320.5787)+[((1/6240.66)+0.0004)*(1.2095)]+0.0002+($1.844*10^{-5}$)

= 0.004015

U = 249.035 kcal/hr-m^2-°c

Surface Area Calculations

Surface Area = Heat duty/(U*MTD)

= 64948/(249.035*15.724)

= 16.586 m^2

Effective length=Actual length –[2*(tube sheet thickness)]

=3300-(2*55)

L =3190mm

Actual area provided = $\prod*d_0*L*n$

= 3.14*0.01905*3.190*90

= 17.1827

% Margin available on surface area = (Actual area-Surface Area)/Surface Area

= (17.1827-16.586)/16.586

= 3.594%

6. PRESSURE DROP DESIGN

Shell Side Pressure Drop:

$\Delta P_{s1} = (B_0*2f^1*N_r*G_s^2)/(g_{0*}\rho_0)$

Where

B_0 = Number of baffles + 1 = 33+1 = 34

X_T = Ratio of row pitch to tube pitch

= 25.4/19.05 = 1.33

$f^1 = b_0 *(D_0*G_s/\mu_f)^{-0.15}$

Where

$b_0 = 0.23+(0.11/(X_T-1)^{1.08})$ (for staggered tubes)

$= 0.23+(0.11/(1.33-1)^{1.08})$ =0 .59031

$f^1 = 0.59031*((19.05*10^{-3}*489.149981)/(2.6134*10^{-3}))^{-0.15}$

= 0.17308

N_r = Number of rows of tubes across which shell side fluid flows = 8

G_s = m/(t*free area) $=14754/(3600*8378.48*10^{-6})$ = 489.14998 kg/m^2-sec

g_s = Acceleration due to gravity = 9.81m/sec^2

ρ_0 = Density of shell side fluid = 851.85 kg/m^3

$\Delta P_{s1} = (B_0*2f^1*N_r*G_s^2)/(g_c*\rho_0)$

$= [34*2*0.17308*8*(489.14998)^2]/[9.81*851.85]$ = 2695.86 kg/m^2

= 0.2695 kg/cm^2

Nozzle Losses:

$N_1 = V^2/2g$ V = Velocity of nozzle

d_{n2} = diameter of nozzle = 73mm= 0.073m

V = Quantity of flow/Area of nozzle $= Q/(\prod/4*d_{n2}^2)$

$= 14754/((\prod/4)*851.85*3600*0.073^2)$ = 1.1494 m/sec

$N_1 = 0.5(1.1494)^2/(2*9.81)$ = 0.0336kg/cm^2

$N_2 = ½*(V^2/2g)$ = ½*0.067345 = 0.0336 kg/cm^2

Total Nozzle losses (ΔP_n) = 0.0336+0.0336 = 0.067345 kg/cm^2

Total pressure drop (ΔP_s) = $\Delta P_{s1}+\Delta P_n$ = 0.2695+0.067345 = 0.336 kg/cm^2

7.0 RESULTS

Table 7.1 shows the final results of theoretical calculations

S.NO	Type of calculation	Result
1	Free flow area	8378.48 mm^2
2	Velocity on cross pass	0.5742m/sec
3	Reynolds Number Re	18.9267
4	Overall Heat Transfer Coefficient	249.035 kcal/hr-m^2-°c
5	% Margin available on surface area	3.594%
6	Shell Side Pressure Drop	0.2695 kg/cm^2
7	Total Nozzle Losses	0.067345 kg/cm^2
8	Total pressure drop (ΔP_s)	0.336 kg/cm^2

7.1 DISCUSSIONS

By checking the results the main objective of theoretical formulation has been observed to prepare an industrial heat exchanger will occupy a larger size. This will take more space and the pressure drop is more because of its larger size and surface area.

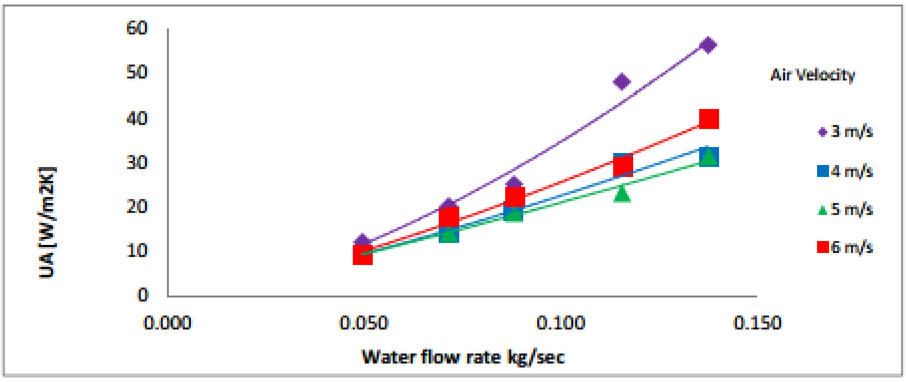

Figure 7.1 Effect of Water Flow Rate on the Overall Heat Transfer Coefficient

8.0 CONCLUSIONS

By observing the results it has been optimized that the surface area calculations are meeting the criteria. Designing compact heat exchanger with same surface area to reach the same advantages which given for heavy duty purposes for the advanced optimization techniques for further future environments and accommodate spaces in globalised industrial sectors to increase the performances at a unique situations. To optimize the pressure drop a compact exchanger have to be develop with same surface area and to minimize the losses for better enhancement. The present data taken from the industry gives an idea to make a compact heat exchanger with cross flow and fins with design optimization.

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