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Design and Mathematical Analysis of Heat Exchanger Subjected to Parallel Flow

RajivRanjan

(Department of Mechanical Engineering, Government polytechnic katihar, Bihar, India)

ABSTRACT

Heat Exchanger is an equipment which transfer the energy from a hot fluid to a cold fluid, with maximum rate and minimum investment and running costs. Heat Exchanger reduces "the waste of thermal energy" which is very important. So, in that respect it helps us saving the fossil fuel which is depleting at a very high rate. The sustainability of the industrial growth the modern society that is dependent to a great extent on the heat exchangers.

In thisdesigning process flow turbulence control, acoustic noise control during operation and external heat in-leak is negligible consider.Design of heat exchangers components includes its dimensions such as length, diameters, thickness, material types, flow configuration, number of heats exchanging units, etc. This study help designer to understand each parameter effect on heat transfer inparallel flow process of heat exchangers.

INTRODUCTION

A heat exchanger is a device used to transfer heat between one or more fluids. Heat exchangers have a wide range of industry applications. They are widely used in space heating, refrigeration, power plants, petrochemical plants, petroleum refineries and sewage treatment.

There are many types of heat exchanger designs for various applications.

1. **Nature of heat exchange process: -**
   a) **Direct contact heat exchangers** – In this type of heat exchanger the exchange of heat takes place by direct mixing of hot and cold fluids.
   b) **Indirect contact heat exchangers** – In this type of heat exchanger the heat transfer between two fluids could be carried out by transmission through wall which separates the two fluids. This type includes regenerators and recuperators or surface exchangers.

2. **Relative direction of fluid motion: -**
   a) Parallel flow or unidirectional flow -In parallel flow, both the hot and cold fluids enter the heat exchanger at the same end and move in the same direction
   b) Counter -flow - In counter-flow, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite direction.
   c) Cross- flow- In cross-flow, the hot and cold fluid streams move perpendicular to each other.

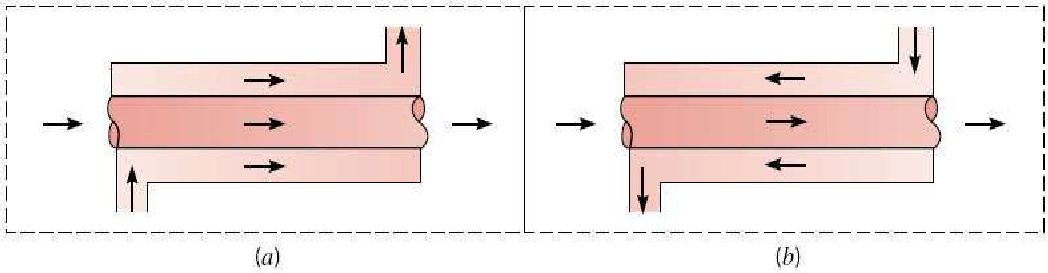

Fig- 1 (a) Parallel flow (b) Counter flow

3. **Design and construction features: -**
   a) Concentric tubes- In this types two concentric tube are used each carrying one of the fluids. The direction of flow may be parallel or counter as desired.
   b) Shell and tubes- In this type of heat exchanger on of the fluids flow through a bundle of tubes enclosed by a shell. The other fluid is forced through the shell and it flows over the outside surface of the tubes. Shell and tube heat exchangers contain a large number of tubes packed in a shell with their axes parallel to that of the shell
   c) Multiple shell and tube passes –In this type of exchangers the fluid is flow back and forth across the tubes by baffles. This heat exchangers enhances the overall heat transfer.

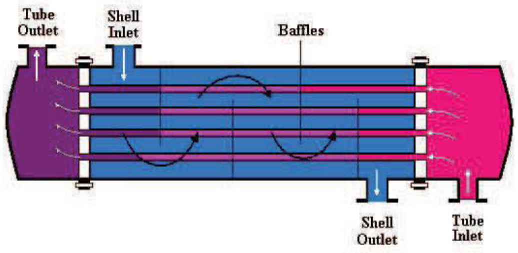

Fig 2- A Shell and Tube heat exchanger

Fig 3- Tube bundle and insertion

DESIGN CONSIDERATIONS AND ANALYSIS OF HEAT EXCHANGER

The flow rates of both hot and cold streams, their terminal temperatures and fluid properties are the primary inputs of design of heat exchangers

1) **Design considerations** - In design of a shell and tube heat exchanger typically includes the determination of heat transfer area, number of tubes, tube length and diameter, tube layout, number of shell and tube passes, type of heat exchanger (fixed tube sheet, removable tube bundle etc), tube pitch, number of baffles, its type and size, shell and tube side pressure drop etc.
2) **Shell -** Shell is the container for the shell fluid and the tube bundle is placed inside the shell. Shell diameter should be selected in such a way to give a close fit of the tube bundle. The clearance between the tube bundle and inner shell wall depends on the type of exchanger. Shells are usually fabricated from standard steel pipe with satisfactory corrosion allowance.
3) **Tube**- The most efficient condition for heat transfer is to have the maximum number of tubes in the shell to increase turbulence. The tube thickness should be enough to withstand the internal pressure along with the adequate corrosion allowance. The tube thickness is expressed in terms of BWG (Birmingham Wire Gauge) and true outside diameter (OD). The tube length of 6, 8, 12, 16, 20 and 24 ft are preferably used. Longer tube reduces shell diameter at the expense of higher shell pressure drop. Finned tubes are also used when fluid with low heat transfer coefficient flows in the shell side. Aluminium tubematerials are considered for analysis purpose.
4) **Tube pitch, tube-layout and tube-count** - Tube pitch is the shortest centre to centre distance between the adjacent tubes. The tubes are generally placed in square or triangular patterns (pitch) . The number of tubes that can be accommodated in a given shell ID is called tube count. The tube count depends on the factors like shell ID, OD of tube, tube pitch, tube layout, number of tube passes, type of heat exchanger and design pressure.
5) **Tube passes**- The number of passes is chosen to get the required tube side fluid velocity to obtain greater heat transfer co-efficient and also to reduce scale formation. The tube passes vary from 1 to 16.
6) **Tube sheet -** The tubes are fixed with tube sheet that form the barrier between the tube and shell fluids. The tube sheet thickness should be greater than the tube outside diameter to make a good seal. The recommended standards (IS:4503) should be followed to select the minimum tube sheet thickness.
7) **Baffles -** Baffles are used to increase the fluid velocity by diverting the flow across the tube bundle to obtain higher transfer co-efficient. The distance between adjacent baffles is called baffle-spacing. The baffle spacing of 0.2 to 1 times of the inside shell diameter is commonly used.
8) **Fouling Considerations -** The most of the process fluids in the exchanger foul the heat transfer surface. The material deposited reduces the effective heat transfer rate due to relatively low thermal conductivity. Therefore, net heat transfer with clean surface should be higher to compensate the reduction in performance during operation. Fouling of exchanger increases the cost of (i) construction due to oversizing, (ii) additional energy due to poor exchanger performance and (iii) cleaning to remove deposited materials. Fouling is considered uniform.
9) **Selection of fluids for tube and the shell side**- The routing of the shell side and tube side fluids has considerable effects on the heat exchanger design.

DESIGN PROBLEM

Related input data for the design of heat exchanger, which is used for multiple distillation unit is given below-
Input steam pressure = 4 bar gauge
Standard atmospheric pressure = 1.013 bar
Absolute pressure = Atmospheric pressure + gauge Pressure
Therefore Input steam pressure = 1.013 + 4 = 5.013 bar
Inlet Temperature of Steam, Thi = 152.7 $^{\circ}C$ at 5.013 bar
Outlet Temperature of Steam, Tho = 117 $^{\circ}C$
Inlet Temperature of Water, Tci = 110 $^{\circ}C$

Specific Heat for Steam, Cph = 2.5 kJ/kg K
Specific Heat for Water, Cpc = 4.18 kJ/kg K
Mass –flow rate of Steam, mh = 45 kg/h = 0.0125 kg/s
Mass –flow rate of Water, mc = 250 kg/h = 0.0694 kg/s

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ANALYSIS OF PROBLEM

Assumptions - The heat transfer coefficients and the fouling factors are constant and uniform

1. Energy Balance Equation-

mc * Cph * (Tco – Tci) = mh * Cph * (Thi – Tho)

0.0694 * 4.18 * (Tco – 110) = 0.0125 * 2.5 * (152.7 – 117)

0.290092 * Tco – 31.91012 = 1.1156

0.290092 * Tco = 33.025745

Tco = 113.85$\circ_C$

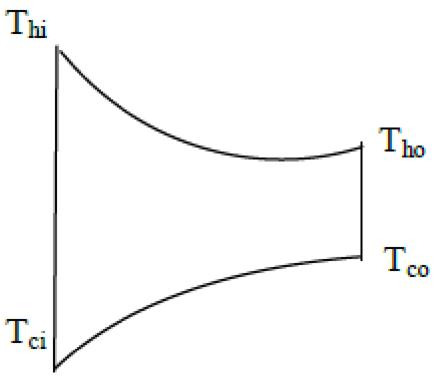
Outlet Temperature of Water, Tco = 113.85$\circ_C$

2. Heat Transfer Rate,

Q = mc*Cph* (Tco – Tci)

= 0.0694 * 4.18 * (113.85 – 110) = 1.17 kW

3. Log Mean Temperature Difference,
LMTD for Parallel Flow, ΔT1 = Thi - Tci
= 152.7 – 110 = 42.7$\circ_C$
ΔT2 = Tho - Tco = 117 – 113.85 = 3.15$\circ_C$



LMTD, ΔTm = (ΔT1− ΔT2)/ ln(ΔT1/ΔT2) = 15.17$\circ_C$

4. Temperature Correction Factor,

The selection of parameters **R** and **P** should be such that the value of correction factor, Ft is more than 0.75

Capacity Ratio, (**R**)=(Tci− Tco) / (Tho − Thi)

= (110 – 113.85) / (117 – 152.7)

R = 0.12

Temperature Ratio,(**P**) = (Tho− Thi)/ (Tci− Thi)

= (117 – 152.7) / (110 – 152.7)

P = 0.84

$\boldsymbol{F_t} = \ (\sqrt{(R^2}+1))/(R-1)*\times\ (ln(1-P)/(1-P*R)/ln(\ 2-P(R+1-\ \sqrt{(R^2}+1)/(\ 2-P(R+1+\sqrt{(R^2}+1))$

$\boldsymbol{F_t}$ = 0.904

5. Mean Temperature Difference,

DTm = $\boldsymbol{F_t}$ * **LMTD**

= 13.71$\circ_C$

6. Overall Heat Transfer Co-efficient

The range of overall heat transfer co-efficient for water is 800 – 1500 w/m^2 $\circ_C$. For the given problem overall heat transfer is assumed as U = 945 w/m^2 $\circ_C$.

7.Provisional Area,

A = (Q/ U∗ ΔT)

= 0.090 m^2

8.Tube Outer Diameter,

Case I- Number of Tubes, Nt = 7, Length of Tubes, L = 100 cm is considered

do = 900 / (3.14*7*100) = 0.41 cm

Case II- Number of Tubes, Nt = 7, Length of Tubes, L = 50 cm is considered

do = 3095.25 / (3.14*7*50) = 2.816 cm

We get best result in case II, therefore outer diameter of tube is 2.816 cm, And the configuration of the tube is-

do = 2.816 cm,Nt = 7& L = 50 cm

9. Tube Pitch,

Pt = 1.25 * do

= 1.25 * 2.816

= 35.2 mm

10. Bundle Diameter,

$\boldsymbol{D_b}$ = do [Nt / K1]$^{1/n}$

Square pitch, $p_t = 1.25d_o$

No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

Table 1: - Relation between constant $K_1$and n_1

For Square pitch,

Pt = 1.25 * do

K_1 = 0.0366

n_1 = 2.63

D_b = 28.16 *[7 / 0.0366]$^{1/2.63}$

= 0.207 m

11. Bundle diameter clearance**,**

For fixed floating head, **BDC = 10 mm**is considered.

12. Shell Diameter,

$D_S = D_b$ + BDC = 207.57 + 10

= 217.57 mm

13. Baffle Spacing,

B_S = 0.4 *D_S

= 0.4 * 217.57

= 87.03 mm

14.Area for cross – flow,

$$As = \frac{(Pt - do) * D_S * B_S}{Pt}$$

= 3550.33 mm^2

15. Shell – side mass velocity,

$$Gs = \frac{\text{Shell Side flow rate [kg/s]}}{As}$$

Shell – side flow rate = 0.0694 kg/s is considered

Gs = (0.0694 /3.55∗ 10−3) = 19.55 kg/m^2−s

16. Shell equivalent diameter for a square pitch arrangement,

d_e = 1.27 ∗ [$Pt^2 - 0.785 * {d_o}^2$]/d_o

= 27.81 mm = 0.02781 m

17. Shell – side Reynolds number,

Following data is takenfor water at 110$\circ_c$ temperature

Density,ρ = 951 kg/m^3

Kinematic viscosity, υ = 0.273 * $10^{-6} m^2$/s

Fluid thermal Conductivity, K_f= 0.62 W/m-K

Specific heat, Cp = 4233 J/kg-K

Reynold no. (Re) = (Gs ∗d_e)/ρυ

= 2094.1

Re > 2000 therefore flow inside shell side is Transition and Turbulent

18. Prandtle number,

Pr =(μ ∗ Cp)/ kf

= (ρ∗ υ ∗Cp)/ kf

= 1.77

19.Nusselt number,

Nu = 0.023 *$(Re)^{0.8}$ * $(Pr)^n$

n = 0.4 for heating………

= 0.3 for cooling………

Nu = 0.023 * $(Re)^{0.8}$ * $(Pr)^{0.4}$ = 0.023 *$(2094.1)^{0.8}$ * $(1.77)^{0.4}$

= 13.11

20. Heat transfer coefficient,

ho = (Nu ∗ kf)/ d_e = (13.11 ∗ 0.62)/0.02781

= 292.35 W/m^2 -K

21.<u>Tube inside diameter</u>,

d_i = d_o - t

Thickness of tube metal = 6 mm is considered

d_i = 28.16 – 6

= 22.16 mm

22.<u>Tube – side Reynolds number</u>,

Following data is takenfor steam at 152.7$\circ_c$ temp.

Dynamic viscosity, µ = 1.408 $* 10^{-5}$N $- s/m^2$

Fluid thermal Conductivity, kf = 0.0311 W/m-K

Specific heat, Cp = 2335.2 J/kg-K

Re = (Gs∗d_i) / µ

= (19.55 ∗ 0.02216) / 1.4085$*10^{-5}$

= 100000

Re > 2000 therefore flow inside tube is Transition and Turbulent.

23. <u>Prandtle number</u>,

Pr = (µ ∗ Cp)/ kf

= 1.4085 $*10^{-5}$∗ 2335.2) / 0.0311

= 1.057

24. <u>Nusselt number</u>,

Nu = 0.023 * $(Re)^{0.8}$ * $(Pr)^{0.3}$

= 0.023 *$(100000)^{0.8}$ * $(1.057)^{0.3}$

= 233.86

25.<u>Heat transfer coefficient</u>,

ho = (Nu ∗ kf) /d_i

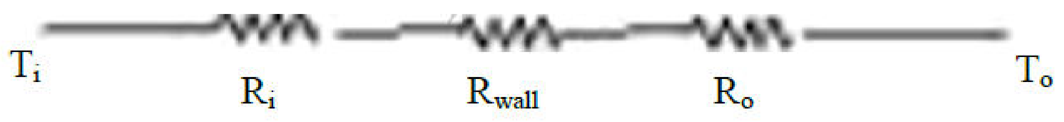
= (233.86 ∗ 0.0311)/ 0.02216

= 328.21 W/m^2– K

26. <u>Overall heat transfer coefficient in Shell and tube heat exchanger</u>

Surface Area of inner tube, A_i= 2πr_iL

Surface Area of outer tube, A_O= 2πr_oL



Total Thermal resistance,

$\sum R = R_i + R_{wall} + R_o$

$\sum R = \frac{1}{hiAi} + \frac{\log(ro/ri)}{2\pi LK} + \frac{1}{hoAo}$

$$Q = \frac{Ti - To}{\sum R}$$

$$= UA\Delta T = UiAi\Delta T = UoAo\Delta T$$

Overall heat transfer coefficient based on outside surface area of tube can be expressed as:

$$Uo = \frac{1}{\sum R\,Ao} = \frac{1}{\frac{Ao}{hiAi} + \frac{Ao\log(ro/ri)}{2\pi LK} + \frac{1}{ho}}$$

$= 136.89\ W/m^2$-K

Overall heat transfer coefficient based on inner surface area of tube can be expressed as:

$$Ui = \frac{1}{\sum R\,Ai} = \frac{1}{\frac{1}{hi} + \frac{Ai\log(ro/ri)}{2\pi LK} + \frac{Ai}{hoAo}}$$

$= 136.89\ W/m^2$-K

After comparing the overall heat transfer coefficient, I obtained from previous step with that I assumed in step 6. It is smaller to what I assumed, then I have a valid assumption, that tabulate my results such as total surface area of tubes, number of tubes, exchanger length and diameter and other design specification.

CONCLUSION

Following conclusion can be made after analysis:

1. Above design of heat exchanger was safe, because our calculated dimensions are verified very accurately.

2. Due to agronomic consideration of design, the tube of very large diameter is not selected.

3. An appropriate heat exchanger is designed for multiple effect distillation unit to condense 45 Kg/hr steam and Material selected isAluminium

Dimension of heat exchangerobtained is given below-

Sr.No.	Parts	Dimension
1	Number of tubes, Nt	7
2	Length of tube, L	500 mm
3	Outer diameter of tube, do	28.16 mm
4	Thickness of the tube, t	6 mm
5	Inner diameter of tube, di	22.16 mm
6	Overall heat transfer coefficient U	136.89 W/m^2K
7	Tube pitch, pt	35.2 mm
8	Bundle diameter, Db	207.57 mm
9	Bundle diameter clearance, BDC	10 mm
10	Shell Diameter, Ds	217.57 mm
11	Baffle spacing, Bs	87.03 mm
12	Area for cross – flow, As	3550.33 mm^2

This study help designer to understand each parameter effect on heat transfer in parallel flow process of heat exchangers. So, we can say heat exchanger one way it is helping us to shape the thermal energy to convert the thermal energy and directly and indirectly both ways it is also helping in protecting the environment. Thus, it is great interest in engineering fields for study of heat exchange. Hence the design engineer needs an efficient strategy for design of heat exchange and utilize the resources optimally to meet the desired purpose.

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