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Thermal and Mechanical Design of Heat Exchanger

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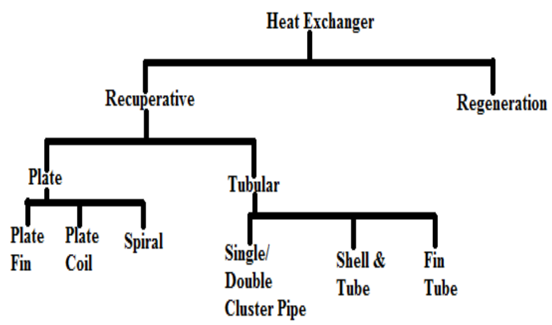
Abstract:
Shell and Tube heat exchangers are having special importance in boilers, oil coolers, condensers, pre-heaters. They are also widely used in process applications as well as the refrigeration and air conditioning industry. The robustness and medium weighted shape of Shell and Tube heat exchangers make them well suited for high pressure operations. The basic configuration of shell and tube heat exchangers, the thermal analysis and design of such exchangers form an included part of the mechanical, thermal, chemical engineering scholars for their curriculum and research activity.
This report presents the application for the optimal design of shell-and-tube heat exchangers. The main objective in any heat exchanger design is the estimation of the minimum heat transfer area required for a given heat duty, as it governs the overall cost of the heat exchanger. Many configurations are possible with various design variables such as outer diameter, pitch, and length of the tubes; tube passes; baffle spacing; baffle cut etc. Hence the design engineer needs to design thermal and mechanical aspects of heat exchanger.

Keywords: Heat exchanger, Thermal design, Mechanical design.

HEAT EXCHANGERS

Introduction:
A heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. The media may be separated by a solid wall to prevent mixing or they may be in direct contact..They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air.

Classification of heat exchangers:



Direct contact heat exchangers:

In this the exchange of heat takes place by direct mixing of hot and cold fluids and transfer of heat and mass takes place simultaneously.The use of such units is made under conditions where mixing of two fluids is either harmless or desirable.

Examples: 1.Cooling towers
2.Condensers

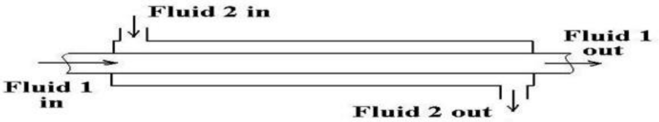
Indirect contact heat exchangers:

In this type of heat exchanger,the heat transfer between the two fluids could be carried by transmission through the wall which separates the two fluids.It includes

1) Regenerators b) Recuperators

Parallel flow heat exchanger:

In a parallel flow heat exchanger,as the name suggets, the fluid streams (hot and cold) travel in the same direction.The two streams enter at one end and leave at the other end.



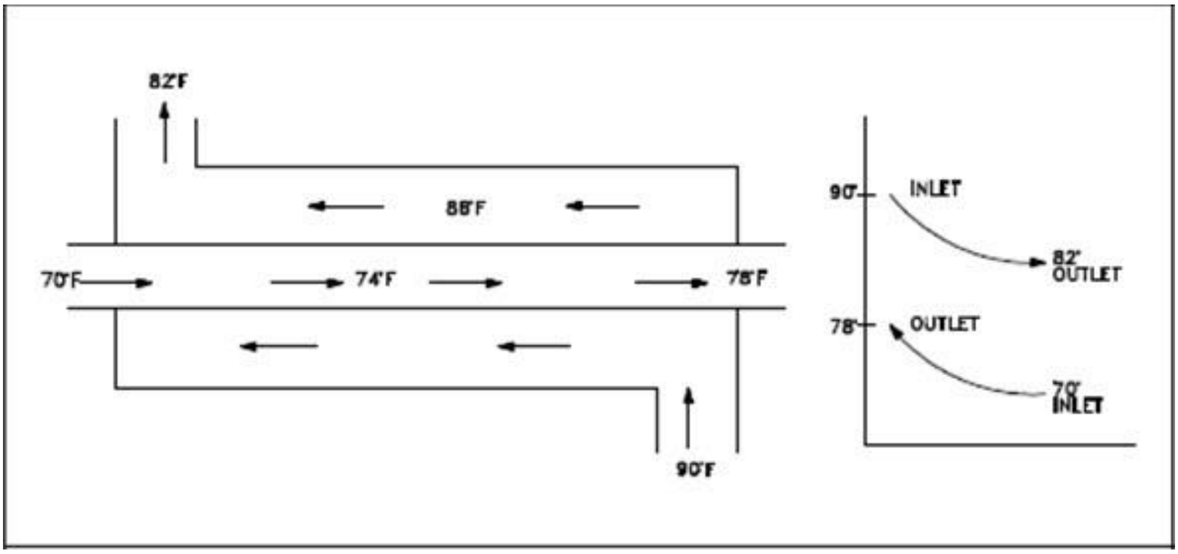
Double Pipe Heat Exchanger
Parallel Flow

Examples: Oil coolers,oil heaters,water heaters

As the two fluids are seperated by wall,this type of heat exchanger may be called parallel flow recuperator or surface heat exchanger.

Counter flow heat exchanger:

In a counter flow heat exchanger,the two fluids flow in opposite directions.The hot and cold fluids enter at the opposite ends.the temperature difference between the two fluids remains more or less nearly constant.



Cross Flow Heat Exchanger:

In a cross flow heat exchanger,the hot and cold fluids Cross one another in space,usually at right angles.

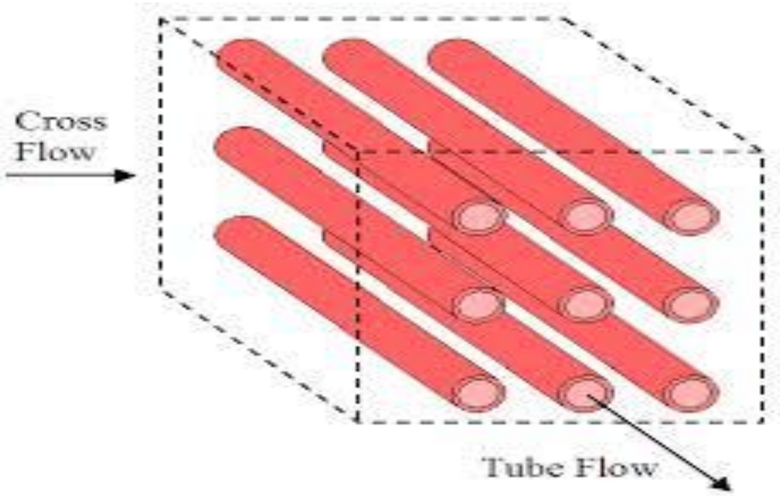


Plate heat exchanger:

Another type of heat exchanger is the plate heat exchanger. One is composed of multiple, thin, slightly separated plates that have very large surface areas and fluid flow passages for heat transfer. This stacked-plate arrangement can be more effective, in a given space, than the shell and tube heat exchanger. Advances in gasket and brazing technology have made the plate-type heat exchanger increasingly practical. In HVAC applications, large heat exchangers of this type are called *plate-andframe*; when used in open loops, these heat exchangers are normally of the gasket type to allow periodic disassembly, cleaning, and inspection.

There are many types of permanently bonded plate heat exchangers, such as dip-brazed, vacuum-brazed, and welded plate varieties, and they are often specified for closed-loop applications such as refrigeration. Plate heat exchangers also differ in the types of plates that are used, and in the configurations of those plates. Some plates may be stamped with "chevron", dimpled, or other patterns, where others may have machined fins and/or grooves.

Plate fin heat exchanger:

This type of heat exchanger uses "sandwiched" passages containing fins to increase the effectiveness of the unit. The designs include crossflow and counterflow coupled with various fin configurations such as straight fins, offset fins and wavy fins. Plate and fin heat exchangers are usually made of aluminium alloys, which provide high heat transfer efficiency. The material enables the system to operate at a lower temperature and reduce the weight of the equipment.

Plate and fin heat exchangers are mostly used for low temperature services such as natural gas, helium and oxygen liquefaction plants, air separation plants and transport industries such as motor and aircraft engines.

LITERATURE REVIEW

As per literature and industrial survey at [A_1 & A_2] the design is carried out using in-house developed software for design and drafting. This dedicated software enables qualified engineers to accomplish complex design calculations complying strictly with the requisite international codes and standards. the dimensions of various components. Also an experienced team of design engineers undertakes thermal and mechanical design of complex heat exchangers and generate fabrication drawings to scale along with weights and estimates based on customer's specifications. These designs are optimized to arrive at an optimal size. After carrying out the design, the final output is in an AutoCAD drawing format (DWG) WILFRIED ROETZEL, CHAKKRIT NA RANONG., [1] calculated the axial temperature profiles in a shell and tube heat exchanger by numerically for given maldistributions on the tube side. For comparison the same maldistributions are handled with the parabolic and hyperbolic dispersion model with fitted values for the axial dispersion coefficient and third sound wave velocity. The analytical results clearly demonstrate that the hyperbolic model is better suited to describe the steady state axial temperature profiles.

SAHOO, R.K., AND WILFRIED ROETZEL.,[2] derived The fundamental equations of hyperbolic model and its boundary conditions in terms of cross-sectional mean temperature from the basic equations of heat exchanger The traditional parabolic model and the proposed hyperbolic model which includes the parabolic model as a special case can be used for dispersive flux formulation. Instead of using the heuristic approach of parabolic or hyperbolic formulation, these models can be quantitatively derived from the axial temperature profiles of heat exchangers. In that paper both the models are derived for a shell-and-tube heat exchanger with pure maldistribution (without back mixing) in tube side flow and the plug
flow on the shell side. The Mach number and the boundary condition which plays a key role in the hyperbolic dispersion have been derived and compared with previous investigation. It is observed that the hyperbolic model is the best suited one as it compares well with the actual calculations. This establishes the hyperbolic model and its boundary conditions.

WILFRIED ROETZEL, AND CHAKKRIT NA RANONG.,[3] tested and compared the newer
hyperbolic dispersion model and parabolic model considering the processes with pure maldistribution (without back mixing) on the tube side of a shell and tube heat exchanger and plug flow on shell side . The boundary conditions of the model equations are discussed in detail for the steady state and equations of the axial temperature profiles are provided in the programmable form. For the hyperbolic model simple relationships between the model parameters are derived.Considering the transient adiabatic processes in the tube bundle a concept for the experimental determination of the model parameter M, the third sound Mach number, is developed. Authors concluded that for an overall consideration of a heat exchanger with maldistribution the parabolic model is satisfactory. The parameter Pepar depends on both NTUs of the heat exchanger which makes the model difficult to handle. The advantage of the parabolic model is that only the only one parameter is needed. The hyperbolic model is superior to the parabolic model because it predicts the axial temperature profiles correctly, especially temperature jumps and (positive) slopes.

YIMINXUAN, WILFRIEDROETZEL.,[4] Applied the dispersion model is to the description of the effects of shell and tube side flow maldistribution. By means of this model, an efficient and versatile method of predicting transient response of multi pass shell and tube heat exchangers is developed. The method allows for effect of maldistribution on transient process, influence of heat capacities of fluids and solid components, arbitrary inlet temperature variations and step disturbances of flow rates. General forms of initial conditions and two different flow arrangements are considered. A general form of the solution for steady-state and dynamic simulation is derived. Temperature profiles are determined with numerical inversion of the Laplace transform. Some examples are calculated and the effect of maldistribution is discussed.Flow mal distribution hinders transient responses to any

inlet changes and decreases thermal effectiveness of heat exchangers. Its effect becomes more remarkable with increasing NTC’. The Peclet
number has been used to quantitatively describe this kind of effect. The calculation has shown that the dispersion model should be applied instead of the plug-flow model if Pe< 55.

ANALYSIS OF HEAT EXCHANGERS

Introduction:

The goal of heat exchanger design is to relate the inlet and outlet temperatures ,overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing. We will limit ourselves to the design of recuperators only. That is, the design of a two fluid heat exchanger used for the purposes of recovering waste heat.

Enthalpy balance on either fluid stream to give:

$$Q_c = \dot{m}_c(h_c2 - h_c1)$$

And $$Q_h = \dot{m}_h(h_h1 - h_h2)$$

For constant specific heats with no change of phase, we may also write

$Q_c= (\dot{m}\ cp)c(Tc2 - Tc1)$

And $Qh = (\dot{m}\ cp)h(Th1 - Th2)$

Now from energy conservation we know that $Qc = Qh = Q$, and that we may relate the heat transfer rate Q and the overall heat transfer coefficient U, to the some mean temperature difference Tm by means of

$Q = UATm$

where A is the total surface area for heat exchange that U is based upon. Later we shall show thats

$Tm = f(Th1, Th2, Tc1, Tc2)$

It is now clear that the problem of heat exchanger design comes down to obtaining an expression for the mean temperature difference. Expressions for many flow con-figurations, i.e. parallel flow, counter flow, and cross flow, have been obtained in the heat transfer field. We will examine these basic expressions later. Two approaches to heat exchanger design that will be discussed are the LMTD method and the effectiveness - NTU method. Each of these methods has particular advantages dependingupon the nature of the problem specification.

Overall Heat Transfer Coefficient:

A heat exchanger analysis always begins with the determination of the overall heat $_I$ transfer coefficient. The overall heat transfer coefficient may be defined in terms of individual thermal resistances of the system. Combining each of these resistances in series gives:

$1/UA = 1/(hA)i + 1/R_o + 1/R_i + 1/(hA)o$

where $\eta 0$ is the surface efficiency of inner and outer surfaces, h is the heat transfer coefficients for the inner and outer surfaces, R_o is outside fouling factor

And Ri is inside fouling factor.

The surface efficiency accounts for the effects of any extended surface which is present on either side of the parting wall. It is related to the fin efficiency of an extended surface in the following manner:

The thermal resistances include the inner and outer film resistances, inner and outer extended surface efficiencies, and conduction through a dividing wall which keeps the two fluid streams from mixing. The shape factors for a number of useful wall configurations are given below in Table 1. Additional results will be presented for some complex doubly connected regions.

The effects of fouling on heat exchanger performance is discussed in a later section. Finally, we should note that

$UA = UoAo = UiAi$

Table 1 - Shape Factors

Geometry		S
1.	Plane Wall	A/t
2.	Cylindrical Wall	2πL/ln(ro/ri)
3.	Spherical Wall	4πriro/(ro – ri)

Table 2 - Order of Magnitude of h

Fluid		h [W/m²K]
1.	Gases (natural convection)	5-25
2.	Gases (forced convection)	10-250
3.	Liquids (non-metal)	100-10,000
4.	Liquid Metals	5000-250,000
5.	Boiling	1000-250,000
6.	Condensation	1000-25,000

LMTD Method:

Logarithmic mean tempe rature difference is defined as that temperature difference which,if constant,would give the same rate of heat transfer as actually occurs under variable conditions of temperature difference.

In order to derive expression for LMTD for various types of heat exchangers ,the following assumptions are made:

1. The overall heat transfer coefficient U is constant
2. The flow conditions are steady
3. The specific heats and mass flow rates of both fluids are constant
4. There is no loss of heat to the surroundings due to the heat exchanger being perfectly insulated
5. There is no change of phase either of the fluid during the heat transfer
6. The changes in potential and kinetic energy are negligible
7. Axial conduction along the tubes of the heat exchanger is negligible

The log mean temperature difference (LMTD) is derived in all basic heat transfer texts. It may be written for a parallel flow or counterflow arrangement. The LMTD has the form:

LMTD =(T2 −T1)/lnT2/T1)

where T1 and T2 represent the temperature difference at each end of the heat exchanger, whether parallel flow or counterflow. The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required.

The LMTD method is also applicable to crossflow arrangements when used with the crossflow correction factor. The heat transfer rate for a crossflow heat exchanger may be written as:

Q = FUATLMTD

where the factor F is a correction factor, and the log mean temperature difference is based upon the counterflow heat exchanger arrangement.

The LMTD method assumes that both inlet and outlet temperatures are known. When this is not the case, the solution to a heat exchanger problem becomes some-what tedious. An alternate method based upon heat exchanger effectiveness is more appropriate for this type of analysis. If T1 = T2 = T, then the expression for the LMTD reduces simply to T.

NTU Method:

The effectiveness / number of transfer units (NTU) method was developed to simplify a number of heat exchanger design problems. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate if there were infinite surface area. The heat exchanger effectiveness depends upon whether the hot fluid or cold fluid is a minimum fluid. That is the fluid which has the smaller capacity coefficient $C = m^{\cdot} C_p$. If the cold fluid is the minimum fluid then the effectiveness is defined as:

$$\mathbf{Q} = \frac{\mathbf{Cmax(TH{,}in - TH{,}out)}}{\mathbf{Cmin(TH{,}in - TC{,}in)}}$$

otherwise, if the hot fluid is the minimum fluid, then the effectiveness is defined as:

$$\mathbf{Q} = \frac{\mathbf{Cmax\ (TC{,}out - TC{,}in)}}{\mathbf{Cmin(TH{,}in - TC{,}in)}}$$

The heat tansfer rate

$$\mathbf{Q\ max = Cmin(TH{,}in - TC{,}in)}$$

It is now possible to develop expressions which relate the heat exchanger effectiveness to another parameter referred to as the nmber of transfer units (NTU). The value of NTU is defined as:

NTU =UA/Cmin

It is now a simple matter to solve a heat exchanger problem when

Q = f(NTU,Cr)

where Cr = Cmin/Cmax.

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The handout summarizes a number of these solutions and the special cases which may be derived from them. For convenience the Q–NTU relationships are given for a simple double pipe heat exchanger for parallel flow and counter flow:

Parallel Flow

$$Q = \frac{1 - \exp[-NTU(1 + Cr)]}{1 + Cr}$$

Counter Flow

$$Q = \frac{\mathbf{1 - exp[-NTU(1 - Cr)]}}{\mathbf{1 - Cr\ exp[-NTU(1 - Cr)]}}$$

$$\varepsilon = \frac{NTU}{1+NTU}, \quad Cr = 1$$

For other configurations, the student is referred to the Heat Transfer course text, or the handout. Often manufacturer's choose to present heat exchanger performance in terms of the inlet temperature difference

$ITD = (T_{h,i} - T_{c,i})$. This is usually achieved by plotting the normalized parameter $Q/ITD = Q/(T_{h,i} - T_{c,i})$. This is a direct consequence of the Q − NTU method.

Heat Exchanger Pressure Drop

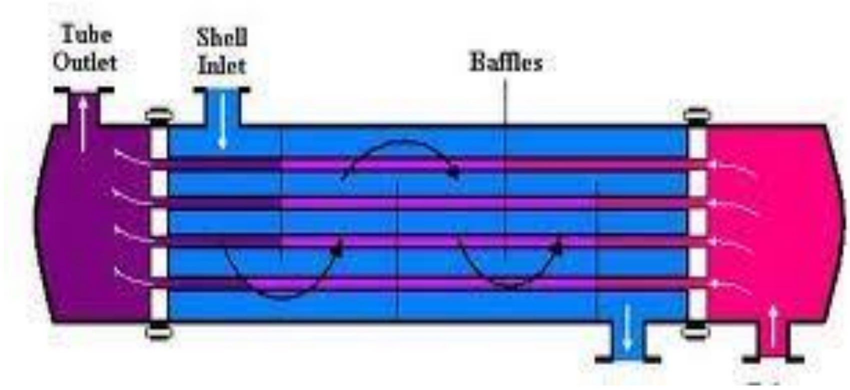
Pressure drop in heat exchangers is an important consideration during the design stage. Since fluid circulation requires some form of pump or fan, additional costs are incurred as a result of poor design. Pressure drop calculations are required for both fluid streams, and in most cases flow consists of either two internal streams or an internal and external streams.

CONSTRUCTIONAL DETAILS OF SHELL AND TUBE HEAT EXCHANGER

Introduction:

The shell-and-tube heat exchanger is named for its two major components – round tubes mounted inside a cylindrical shell. The shell cylinder can be fabricated from rolled plate or from piping (up to 24 inch diameters). The tubes are thin-walled tubing produced specifically for use in heat exchangers.

Other components include: the channels (heads), tube sheets, baffles, tie rods & spacers, pass partition plates and expansion joint (when required). Shell & tube heat exchanger designs and constructions are governed by the TEMA and ASME codes.



Shell and tube heat exchanger

Tubes:

Tubing may be seamless or welded. Seamless tubing is produced in an extrusion process; weldedtubisssssng is produced by rolling a strip into a cylinder and welding the seam. Welded tubing is usually more economical.Normal tube diameters are 5/8 inch, 3/4 inch and 1 inch. Tubes of smaller diameter can be used but they are more difficult to clean mechanically. Tubes of larger diameter are sometimes used either to facilitate mechanical cleaning or to achieve lower pressure drop. The normal tube wall thickness ranges from 12 to 16 BWG (from 0.109 inches to 0.065 inches thick).Tubes with thinner walls (18 to 20 BWG) are used when the tubing material is relatively expensive
such as titanium.

Tubing may be finned to provide more heat transfer surface; finning is more common on the outside of the tubes, but is also available on the inside of the tubes. High flux tubes are tubing with special surface to enhance heat transfer on either or both sides of the tube wall. Inserts such as twisted tapes can be installed inside tubes to improve heat transfer especially when handling viscous fluids in
laminar flow conditions. Twisted tubes are also available. These tubes can provide enhanced heat transfer in certain applications.

Tubesheets:

Tubesheets are plates or forgings drilled to provide holes through which tubes are inserted. Tubes are appropriately secured to the tubesheet so that the fluid on the shell side is prevented from mixing with the fluid on the tube side. Holes are drilled in the tubesheet normally in either of two patterns,triangular or square.

Fig 5.3: Tubesheet

The distance between the centers of the tube hole is called the tube pitch; normally the tube pitch is 1.25 times the outside diameter of the tubes. Other tube pitches are frequently used to reduce the shell side pressure drop and to control the velocity of the shell side fluid as it flows across the tube bundle. Triangular pitch is most often applied because of higher heat transfer and compactness it
provides. Square pitch facilitates mechanical cleaning of the outside of the tubes.
Two tubesheets are required except for U-tube bundles. The tubes are inserted through the holes in the tubesheets and are held firmly in place either by welding or by mechanical or hydraulic expansion.

A rolled joint is the common term for a tube-to-tube sheet joint resulting from a mechanical expansion of the tube against the tubesheet. This joint is most often achieved using roller expanders; hence the term rolled joint. Less frequently, tubes are expanded by hydraulic processes to affect a mechanical bond. Tubes can also be welded to the front or inboard face of the tubesheet. Strength welding
designates that the mechanical strength of the joint is provided primarily by the welding procedure and the tubes are only lightly expanded against the tubesheet to eliminate the crevice that would otherwise exist. Seal welding designate that the mechanical strength of the joint is provided primarily by the tube expansion with the tubes welded to the tubesheet for better leak protection. The cost of sealwelded joints is commonly justified by increased reliability, reduced maintenance costs, and fewer process leaks. Seal-welded joints are required when clad tubesheets are used, when tubes with wall thickness less than 16 BWG (0.065 inch) are used, and for some metals that cannot be adequately expanded to achieve an acceptable mechanical bond (titanium and Alloy 2205 for instance).

Baffles:

Baffles serve three functions:
1) support the tube
2) maintain the tube spacing
3) direct the flow of fluid in the desired pattern through the shell side.

A segment, called the baffle cut, is cut away to permit the fluid to flow parallel to the tube axis as it flows from one baffle space to another. Segmental cuts with the height of the segment approximately 25 percent of the shell diameter are normally the optimum. Baffle cuts larger or smaller than the optimum typically result in poorly distributed shell side flow with large eddies, dead zones behind the baffles and pressure drops higher than expected. The spacing between segmental baffles is called the baffle pitch. The baffle pitch and the baffle cut determine the cross flow velocity and hence the rate of heat transfer and the pressure drop. The baffle pitch and baffle cut are selected during the heat exchanger design to yield the highest fluid velocity and heat transfer rate while respecting the allowable pressure drop. The orientation of the baffle cut is important for heat exchanger installed horizontally. When the shell

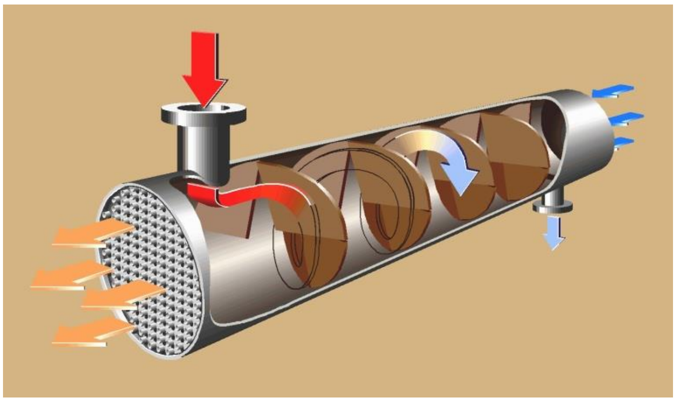
side heat transfer is sensible heating or cooling with no phase change, the baffle cut should be horizontal. This causes the fluid to follow an up-and-down path and prevents stratification with warmer fluid at the top of the shell and cooler fluid at the bottom of the shell. For shell side condensation, the baffle cut for segmental baffles is vertical to allow the condensate to flow towards the outlet without significant liquid holdup by the baffle. For shell side boiling, the baffle cut may be

either vertical or horizontal depending on the service.

Other types of baffles are sometimes used such as: double segmental, triple segmental, helical baffle, EM baffle and ROD baffle. Most of these types of baffles are designed to provide fluid flow paths other than cross flow.

Arrangement of baffles

Longitudinal baffles are sometimes provided to divide the shell creating multiple passes on the shell side. This type of heat exchangers is sometimes useful in heat recovery applications when several shell side passes allow to achieve a severe temperature cross.



Tie Rods and Spacers:

Tie rods and spacers are used for two reasons:

1) Hold the baffle assembly together
2) Maintain the selected baffle spacing.

The tie rods are secured at one end to the tubesheet and at the other end to the last baffle. They hold the baffle assembly together. The spacers are placed over the tie rods between each baffle to maintain the selected baffle pitch. The minimum number of tie rod and spacers depends on the diameter of the shell and the size of the tie rod and spacers.

Shell Assembly:

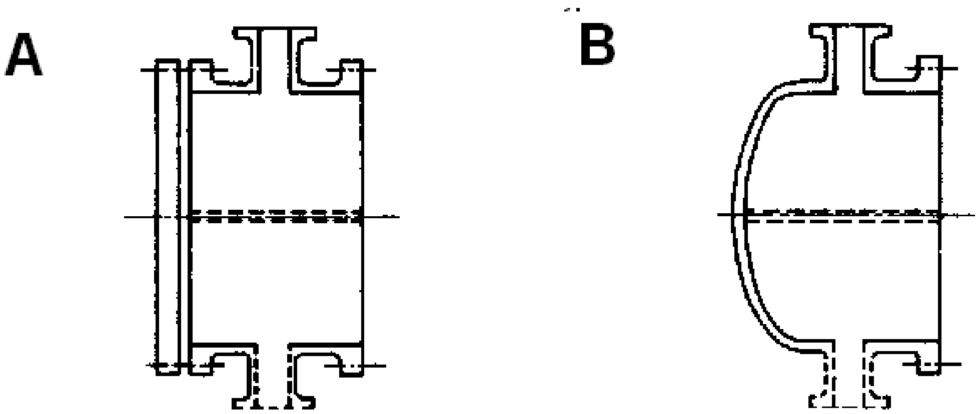
The shell is constructed either from pipe or rolled plate metal. Foreconomic reasons, steel is the most commonly used material, and when applications involving extreme temperatures and corrosion resistance, others metals or alloys are specified. Using off-the-shelf pope reduces manufacturing costs and lead time to deliver to the end customer. A consistent inner shell diameter or'roundness' is need to minimize the baffle spacing on the outside edge, excessive space reduces performance as the fluid tends to channel and bypasses the core. Roundness is increased typically by using a mandrel and expanding the shell around it, or by double rolling the shell after welding the longitudinal seam.

In some cases, although extreme, the shell is cast and then bored out until the correct inner diameter is achieved.When fluid velocity at the nozzle is high, an 'impingement' plate is specified to distribute fluid evenly in the tubes, thereby preventing fluid-induced erosion,vibration and cavitation. Impingement plates effectively eliminate the need to configure a full tube bundle, which would otherwise provide less available surface. An impingement plate can also be installed above the shell thereby allowing a full tube count and therefore maximizing shell space .

Bonnets and End Channels:

Bonnets / end channels regulate the flow of fluid in the tube-side circuit, they are typically fabricated or cast. They are mounted against the tube sheet with a bolt and gasket assembly; many designs include a 'machine grooved' channel in the tube sheet sealing the joint. If one or more passes are intended, the head may include pass ribs that direct flow through the tube bundle (figure C). Pass ribs are aligned on either end to provide effective fluid velocities through an equal number of tubes at a time ensuring a constant, even fluid velocity and pressure drop throughout the bundle.

Shell and tube configurations with up to (4) passes are the most common, however specialty designs do allow 20 or more crossings. The tube sheet configuration in a multi-pass shell and tube design must have provisions for the pass ribs, requiring either removal of tubes to allow a low cost straight pass rib
or alternately a pass rib with curves around the tubes adding cost to the manufacture process. When a full bundle count is needed for the thermal requirement, machine pass ribs usually prevent the need to 'upsize' to the next larger shell diameter. The material used in the cast bonnets / heads used in smaller diameters (ie 15" or less) are typically, poured from iron, steel, bronze, nickel plated, or stainless steel. Pipe connections are normally NPT, others including SAE, tri-clamp, ASME flanged, BSPP, and others types are available.



Channel and removable cover

Bonnet (integral cover)

Types of Bonnet

<u>THERMAL DESIGN</u>

Operating Parameters:

Sno	Parameters	Shell side	Tube side
1.	Fluid	Light Naphtha	Kero - cr
2.	Inlet temperature	160° C	225° C
3.	Outlet temperature	169° C	181° C
4.	Inlet pressure	9.161 kg/cm^2	12.433 kg/cm^2
5.	Outlet pressure	0.9 kg/cm^2	3 kg/cm^2
6.	Design pressure	14 kg/cm^2	14 kg/cm^2

7.	Pressure drop	0.142 kg/cm^2	2.59 kg/cm^2
8.	No. of passes	4	4

- Tube size = 5/8 " O.D
- Oil flow rate = 357994 Kg/hr
- Tube material = Carbon Steel
- Specific heat of transformer oil, C = 0.6280 J/kg-K
- Density of oil, ρ= 767.5 kg/m^3
- Length of the tube = 6000 mm

CALCULATIONS:

- Making heat balance between hot and cold fluid

$m_h c_h (T_1 - T_2) = m_c c_c (t_2 - t_1)$

$742.05 \times 2382(225 - 181) = mc \times 4186(169 - 160)$

$14.1405 \times 10^6 = m_c \times 20930$

$m_c = 675.61$ kg/min

- True temperature difference:

$$R = \frac{T_1 - T_2}{t_2 - t_1} \qquad S = \frac{t_2 - t_1}{T_1 - t_1}$$

$$R = \frac{225 - 181}{169 - 160} \qquad S = \frac{169 - 160}{225 - 160}$$

$$R = 4.8 \qquad S = 0.13$$

- $\theta_1 = 225 - 160 = 65°C$

- $\theta_2 = 181 - 169 = 12°C$

- $\mathrm{LMTD} = \frac{\theta_1 - \theta_2}{\ln\frac{\theta_1}{\theta_2}} = \frac{65-12}{\ln\frac{65}{12}} = 31.54°C$

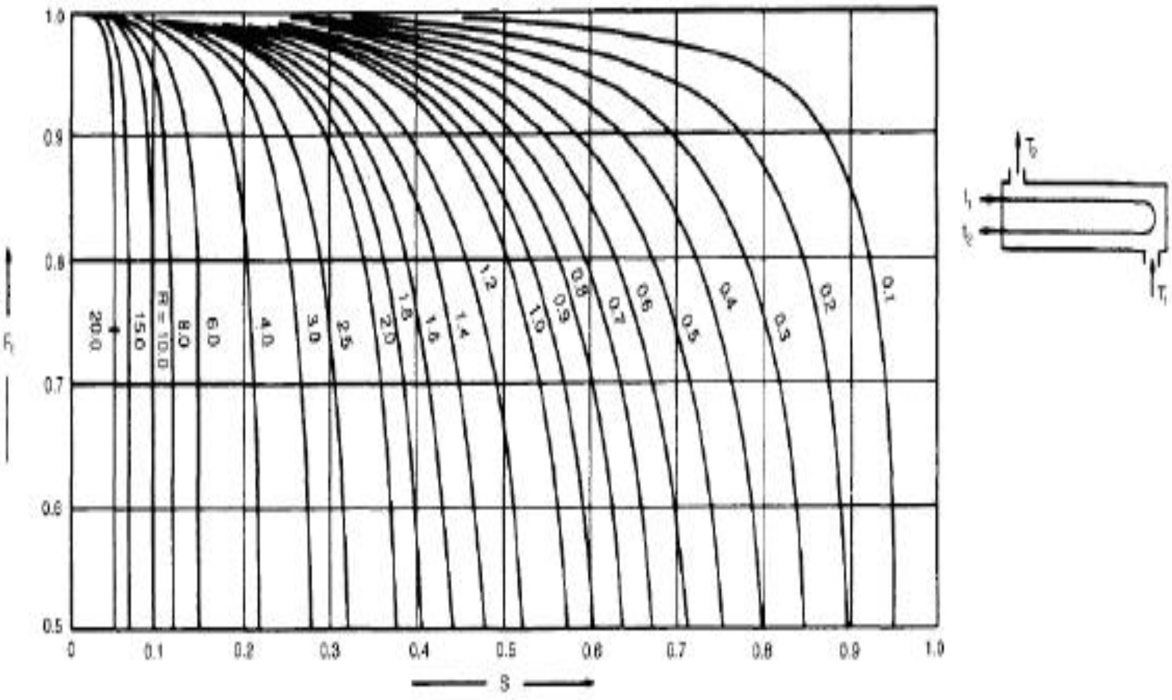
- From graph (Kern):

Temperature difference factor, $F_T = 0.95$

$$\Delta_t = \mathrm{LMTD} \times F_T$$

$$= 31.54 \times 0.95$$

$$\Delta_t = 29.96°C$$



➢ Caloric temperature:

$$T_C = \frac{T_1 + T_2}{2} \qquad t_C = \frac{t_1 + t_2}{2}$$

$$T_C = \frac{225 + 181}{2} \qquad t_C = \frac{160 + 169}{2}$$

$$T_c = 203℃ \qquad t_c = 164.5℃$$

- Assuming square pitch as tube pattern.

➢ Tube diameter, $d_o = 20$ mm

Tube pitch, P_T $= 1.25 \times d_o$

$= 1.25 \times 20$

$= 25$ mm

➢ Clearance between tubes, c' $= P_T - d_o$

$= 25 - 20$

$= 5$ mm

➢ Shell inner diameter D_s $=$ 21 times of tube ID

$= 21 \times 20$

$= 420$ mm

➢ Minimum baffle spacing, B $= \frac{1}{5} \times$ shell ID

$$= \frac{1}{5} \times 420$$

$= 84$ mm

Calculation of Shell side heat transfer coefficient:

- Flow area, $a_s = ID \times \frac{c' \times B}{P_T}$

$$= 420 \times 5 \times \frac{84}{25}$$
$$= 705 \text{ mm}^2$$

- Mass velocity, $G_s = \frac{m_c}{a_s}$

$$= \frac{5966}{7056 \times 10^{-6} \times 60}$$
$$= 14092.02 \frac{\text{kg}}{\text{m}^2\text{sec}}$$

- Equivalent diameter, $D_e = \frac{4 \times \text{free area}}{\text{wetted perimeter}}$

$$= \frac{4 \times \left[{P_T}^2 - \frac{\pi {d_o}^2}{4}\right]}{\pi d_o}$$
$$= \frac{4\left[25^2 - \frac{\pi}{4} \times 20^2\right]}{\pi \times 20}$$
$$= 19.78 \text{ mm}$$

- From heat transfer data book, at t_c = 164.5°C

$\mu_s = 0.02476 \text{ N}\frac{\text{S}}{\text{m}^2}$

$\nu_s = 3.21388 \times 10^{-5} \text{ N}\frac{\text{S}}{\text{kg}}$

$\rho_s = 770$ kg/m^3

K_s = 0.06978 W/m-K

C_s = 2868 J/kg-K

- Reynolds number, $R_{es} = \frac{G_s \times D_e}{\mu_s}$

$$= \frac{14092.02 \times 19.78 \times 10^{-3}}{2.476 \times 10^{-4}}$$
$$= 11257.67$$

- j_H = 140 from graph (Kern)

$h_o = j_H \times \frac{K}{D_e} \times (\frac{C_s \times \mu_s}{K_s})^{1/3} \times \emptyset_s$

$$\frac{h_o}{\emptyset_s} = 140 \times \frac{0.06978}{19.78 \times 10^{-3}} \times \left[\frac{2868 \times 0.02476}{0.06978}\right]^{1/3}$$
$$\frac{h_o}{\emptyset_s} = 4962.48 \text{ W/m}^2 J$$

Calculation of Tube side Heat Transfer Coefficient :

- Assuming number of tubes = 1000
- Tube length = 6000 m

- Tensile strength of carbon steel = 130 MPa
- Yield strength of carbon steel = 105 MPa
- Factor of safety = 3

- Allowable stress $= \frac{\text{tensile strength}}{\text{factor of safety}} = \frac{95}{3}$

$= 31.67$ MPa

- As per TEMA:
- For BWG of 20, standard thickness = 2 mm
- Internal area, $a_t' = 1.5606\ cm^2$
- Tube, I.D = 14.10 mm, O.D = 20 mm
- $\frac{O.D}{I.D} = 1.418$
- Flow area , $a_t = \frac{N_t \times a_t'}{n}$

$$= \frac{100 \times 156.06}{2}$$

$$= 7806\ mm^2$$

- Mass velocity, $G_t = \frac{m_h}{a_t}$

$$= \frac{4635}{7806 \times 10^{-6} \times 60} = 9896.23 \frac{kg}{m^2 sec}$$

- From heat transfer data book, at $T_c = 203°C$

$\mu_t = 0.037\ N\frac{S}{m^2}$

$\rho_t = 780\ kg/m^3$
$K_t = 0.1035$ W/m-K
$C_t = 2.6293$ kJ/kg-K

- Reynolds number, $R_{et} = \frac{G_t \times D}{\mu_t}$

$$= \frac{9896.23 \times 14.1 \times 10^{-3}}{0.037}$$

$$= 3771.26$$

$$\frac{L}{d_i} = \frac{6000}{14.10} = 425.53$$

- $j_H = 10.6$ from graph (Kern)

- $$h_i = j_H \times \frac{K}{D} \times \left(\frac{C_t \times \mu_t}{K_t}\right)^{1/3} \times \phi_t$$

$$\frac{h_i}{\phi_t} = 10.6 \times \frac{0.1035}{14.1 \times 10^{-3}} \times \left[\frac{2692.3 \times 0.037}{0.1035}\right]^{1/3}$$

$$\frac{h_i}{\phi_t} = 768.15\ W/m^2k$$

$$\frac{h_{io}}{\phi_t} = \frac{h_i}{\phi_t} \times \frac{ID}{OD} = 768.15 \times \frac{1}{1.418}$$

$$= 541.7\ W/m^2K$$

- Tube wall temperature:

$$t_w = t_c + \frac{h_o/\phi_s}{\frac{h_{io}}{\phi_t} + \frac{h_o}{\phi_s}}(T_c - t_c)$$

$$= 164.5 + \frac{4962.48}{541.7 + 4962.48}(203 - 164.5)$$
$$= 164.5 + 34.71$$
$$t_w = 199.2℃$$

Shell side:

- At t_w = 199.2℃, from heat transfer data book

$\nu_s = 0.0278 \frac{N\,S}{m^2}$

$\rho_s = 775\ kg/m^3$

- $\mu_w = \nu_s \times \rho_s = 7.68 \times 10^{-4} \frac{N\,S}{m^2}$
- Viscosity correction factor, $\emptyset_s = \left(\frac{\mu_s}{\mu_w}\right)^{0.14} = \left(\frac{0.02476}{0.0238}\right)^{0.14}$

$$= 1.005$$

- Corrected coefficient, $h_o = \frac{h_o}{\emptyset_s} \times \emptyset_s$

$$= 4962.48 \times 1.005$$
$$= 4990.02 \frac{w}{m^2k}$$

Tube side:

- At t_w = 199.2℃, from heat transfer data book

$\mu_w = 0.042 \frac{N\,S}{m^2}$ $\mu_t = 0.037 \frac{N\,S}{m^2}$

$$\emptyset_t = \left(\frac{\mu_t}{\mu_w}\right)^{0.14} = \left(\frac{0.037}{0.042}\right)^{0.14}$$
$$\emptyset_t = 0.982$$

- Corrected coefficient, $h_i = \frac{h_{io}}{\emptyset_t} \times \emptyset_t$

$$= 541.7 \times 0.982$$
$$= 532.17 \frac{w}{m^2k}$$

- Overall heat transfer coefficient:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + R_{fi}\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) + \frac{r_o}{K}\ln\left(\frac{r_o}{r_i}\right)$$

$$= \frac{1}{4990.02} + 0.0002 + 0.0003\left(\frac{10}{7.05}\right)^2 + \frac{1}{532.17}\left(\frac{10}{7.05}\right)^2 + \frac{10\times10^{-3}}{112.4} ln\left(\frac{10}{7.05}\right)$$

$\frac{1}{U_o} = 4.185 \times 10^{-3}$

$U_o = 207.68\ W/m^2 K$

- Heat transfer rate

$Q = U_o\, A\, \Delta t$

$8936.99 \times 10^3 = 207.68 \times A \times 29.96$

$A = 1450.85\ m^2$

- Area of one tube, $a = \pi d_o l$

$a = \pi \times 0.02 \times 6$

$a = 0.376\ m^2$

- No. of tubes required, $N_t = \frac{\text{total area}}{\text{area of one tube}}$

$$= \frac{1450.85}{0.376}$$

$$N_t = 3848.49$$

No. of tubes in one pass $= \frac{N}{2} = \frac{3848}{2} = 1924$

Iteration:

- Taking no. of tubes as 1924
- Flow area $a_t = \frac{N_t \times a_t'}{n} = \frac{1924 \times 156.06}{2}$

$$= 150129.72\ mm^2$$

- Mass velocity, $G_t = \frac{m_h}{a_t}$

$$= \frac{77.25}{150129.72 \times 10^{-6}}$$

$$= 514.55 \frac{kg}{m^2 sec}$$

- Reynolds number, $Re_t = \frac{diG_t}{\mu}$

$$= \frac{14.1 \times 10^{-3} \times 514.55}{0.037}$$

$$= 196$$

$$\frac{L}{d_i} = \frac{6000}{14.1} = 425.53$$

- $j_H = 90$

$$h_i = j_H \times \frac{K_t}{D_t} \times \left(\frac{C_t \times \mu_t}{K_t}\right)^{1/3} \times \emptyset_t$$

$$\frac{h_i}{\emptyset_t} = 90 \times \frac{0.1035}{14.1 \times 10^{-3}} \times \left[\frac{2692.3 \times 0.037}{0.1035}\right]^{1/3}$$

$$\frac{h_i}{\emptyset_t} = 6374.62$$

$$\frac{H_{io}}{\emptyset_t} = \frac{h_i}{\emptyset_t} \times \frac{ID}{OD} = 6374.62 \times \frac{1}{1.418}$$

$$= 4495.5.\ W/m2\ K$$

- Tube wall temperature:

$$t_w = t_c + \frac{h_o/\emptyset_s}{\frac{h_{io}}{\emptyset_t} + \frac{h_o}{\emptyset_s}} (T_c - t_c)_s$$

$$= 164.5 + \frac{4962.48}{4962.48 + 4495.5}(203 - 164)$$

$$t_w = 184.5\ ℃$$

Shell side:

- At t_w = 184.5 °C, from data book

$$\nu_s = 0.775 \times 10^{-6}\ m^2/sec$$

$$\rho_s = 996.7\ kg/m^3$$

$\mu_w = \mathrm{v_s} \times \rho_s = 0.0236\ \frac{\mathrm{N\,S}}{\mathrm{m^2}}$

- Viscosity correction factor, $\emptyset_s = \left(\frac{\mu_s}{\mu_w}\right)^{0.14} = \left(\frac{0.02476}{0.0236}\right)^{0.14}$

$$= 1.006$$

- Corrected coefficient, $h_o \quad = \frac{h_o}{\emptyset_s} \times \emptyset_s$

$$= 4962.48 \times 1.006$$
$$= 4992.25\, \frac{\mathrm{w}}{\mathrm{m^2k}}$$

Tube side:

- At t_w = 184.5°C, from data book

$\mu_w = 0.043 \frac{\mathrm{N\,S}}{\mathrm{m^2}}$

$$\emptyset_t = \left(\frac{\mu_t}{\mu_w}\right)^{0.14} = \left(\frac{0.037}{0.043}\right)^{0.14}$$
$$= 0.979$$

- Corrected coefficient, $h_{io} \quad = \frac{h_{io}}{\emptyset_t} \times \emptyset_t$

$$= 4495.5 \times 0.979$$
$$= 4401.09\ \frac{\mathrm{w}}{\mathrm{m^2k}}$$

- Overall heat transfer coefficient:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + R_{fi}\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) + \frac{r_o}{K}\ln\left(\frac{r_o}{r_i}\right)$$

$$= \frac{1}{4992.25} + 0.0002 + 0.0003\left(\frac{10}{7.05}\right)^2 + \frac{1}{4401.9}\left(\frac{10}{7.05}\right)^2 + \frac{10\times10^{-3}}{112.4}\, ln\left(\frac{10}{7.05}\right)$$

$\frac{1}{U_o} = 0.00149168$

$U_o = 670.385\ \mathrm{W/m^2\,K}$

- Heat transfer rate , $Q = U_o\, A\, \Delta t$

$$8936.99 \times 10^3 = 670.385 \times A \times 29.66$$
$$A = 449.46\ \mathrm{m^2}$$

- Area of one tube, $a = \pi d_o l$

$$a = \pi \times 20 \times 10^{-3} \times 6000$$
$$a = 0.376\ \mathrm{m^2}$$

- No. of tubes required, $N_t = \frac{\text{total area}}{\text{area of one tube}}$

$$= \frac{449.46}{0.376}$$
$$N_t = 1195.37$$

- No. of tubes in one pass $= \frac{N}{2} = \frac{1195.37}{2} = 598$
- The obtained number of tubes is less than the assumed value of iteration 1.
- Hence the number of tubes 598 is fixed.

- Taking into account additional area as 9%

Area $= 449.46\times 1.09 = 489.92\ m^2$

No. of tubes $= \frac{489.92}{0.376} = 1302.95$

No. of tubes in one pass = 651.4

Pressure Drop Calculations:

Shell side:

- For R_{es} = 11257.67, friction factor f = 0.000354 ft^2/in^2
- Specific gravity of cold fluid s = 0.5462
- No. of crosses, N+1 = L_t /B

$= \frac{6000}{84} = 71.42$

- Pressure drop, $\Delta P_s = \frac{f G_s^2 D_s\ N+1}{5.22*10^{10} D S \emptyset_s}$ psi

$= \frac{0.000354\times(14092.02)^2\times 420\times 71.42}{5.22\times 10^{10}\times 19.78\times 1.005}$

= 2.033 psi

= 0.142 kg/cm^2

Tube side:

- For $Re_t =$ 3771.26, friction factor ,f = 0.0169 ft^2/in^2
- Specific gravity of hot fluid s = 0.6573
- Pressure drop in tube, $\Delta P_t = \frac{f G_t^2 L n}{5.22*10^{10} D S \emptyset_t}$ psi

$= \frac{0.0169\times(9896.23)^2\times 6\times 2}{5.22\times 10^{10}\times 0.01978\times 0.873\times 0.982}$ psi

= 2.24 psi

= 0.156 kg/cm^2

- Pressure drop in return flow of hot fluid

$\Delta P_r = \frac{4nv^2}{s\times 2g}$ psi

$= \frac{4\times 2\times 3.80}{0.873}$

= 2.44 kg/cm^2

- Total pressure drop in tube side= $\Delta P_t + \Delta P_t$

= 0.156+2.44

= 2.596 kg/cm^2

- Allowable pressure on shell and tube side = 14 kg/cm^2

Hence the heat exchanger is satisfactory for the service

MECHANICAL DESIGN

Components are designed for strength taking into account of design pressure, material, effect of corrosion, pressures acting on components.

1. Design of shell:

- Shell material = carbon steel
- Allowable stress of

  Shell S_{allow} = 95N/mm^2
- Shell I.D, Ds = 550 mm
- Radius of shell = 275mm
- Corrosion allowance = 3.175
- Design is done in corroded condition:
- E = joint efficiency (spot radiography) = 0.85
- Thickness of shell, t_{shell} = $\frac{P_S \times R}{2SE - 0.6P}$

$$= \frac{1.373 \times 550}{2 \times 95 \times 0.85 - 0.6 \times 1.373}$$

= 4.69mm

- Adding corrosion allowance = 4.69+3.175

  = 7.865 mm
- As per TEMA for shell inner dia= 550 mm
- Minimum shell thickness = 13.17 mm = 14 mm
- Shell thickness to be provided = max(7.865 ,14) =14 mm

2. Design of flange:

- Shell outer diameter, B = shell I.D + 2(thickness)

  B = 550 + 2(14)

  B = 578 mm
- Pitch circle diameter, C = O.D + 2(fillet size) + 2(radius)

  = 578 + 2(10) + 2(30)

  C = 658 mm
- Gasket O.D = PCD – 23 –(2×3) – (2× 1.5)

  = 658 – 23 -6 – 3

  = 626 mm
- Gasket effective diameter, G = gasket OD – 2b

  = 626 – 2(5.635)

  = 614.73 mm

b = effective gasket width

$$b = 2.52\sqrt{b_o}$$

where $b_o = \frac{\text{gasket width}}{2}$

$= \frac{10}{2}$ $= 5$ mm

b = $2.52\sqrt{5}$ = 5.635 mm

Considering Iron Jacketted Asbestos fillet gasket

m = 3.75 (gasket factor)

y = 7600 psi (seating stress)

y = $534.3 \frac{kg}{cm^2}$

- Minimum load for seating the gasket

$H_G = \pi bGy$

$= \pi \times 0.5635 \times 61.473 \times 534.3$

$= 58145.1$ kgf

- Hydrostatic force acting over the effective gasket diameter

$H = \frac{\pi}{4} G^2 P$

$= \frac{\pi}{4} \times 614.73^2 \times 14 \times 10^{-2}$

$= 41551.50$ kgf

- Hydrostatic force acts on gasket width

$H_p = 2b\pi GmP$

$= 2\times \frac{5.63}{10} \times \pi \times \frac{614.73}{10} \times 3.75 \times 14$

$= 11416.47$ kgf

Bolts:

Specification: SA 193 B7

- Allowable stress, S = 25000 psi = $1757.68 \frac{kg}{cm^2}$
- Bolts area required will be greater of

$A_m = \left[\frac{H_G}{S}, \frac{H+H_P}{S}\right]$

$= \left[\frac{58145.1}{1757.68}, \frac{41551.50+11416.47}{1757.68}\right]$

$=[33.08, 30.13]$

$A_m = 33.8 \text{ cm}^2$

- For M20 × 2.5*P*

- Core diameter, d_c = 16.93 mm
- Area of one bolt, a_b = 225 mm^2
- No. of bolts required = $\frac{A}{a_b}$

$= \frac{33.08\times 100}{225}$

$= 14.70$

$\approx$ 16 (Bolts to be multiple of 4)

- Flange O.D , A = P.C.D + 2E

= 658 + 2(25)

= 708 mm

- Load in gasket seating condition

H_G = $b\pi GY$ = 58145.1 kgf
= 57.02 KN

- Load in operating condition

H_P = $2b\pi Gmp$ = 11416.47 kgf = 111.957 KN

H = $\pi G^2 p/4$ = 41551.50 kgf. = 407.48 KN

$H_P + H$ = W_{m1}

W_{m1} = 111.95 + 407.48

W_{m1} = 519.42 KN

Flange load:

- Hydrostatic load on inside diameter

H_D = $\frac{\pi}{4} b^2 p$

= $\frac{\pi}{4} \times 578^2 \times \frac{1}{100} \times 14$

= 36734 kgf

H_D = 360.23 KN

H_G = $W_{m1} - H$ = 519.42 – 407.48

H_G = 111.94 KN

H_T = $H - H_D$ = 407.48 – 360.23 = 47.25

- Total load under operating condition

H_T = 47.25 KN

Lever arm:

- h_D = 0.5(C -B) = 0.5(658 - 578) = 40 mm
- h_G = 0.5(C -G) = 0.5(658 -615) = 21.5 mm
- h_T = $0.5(h_D + h_G)$ = 0.5(40+21.5) = 30.75 mm

Flange moments:

- M_D = $H_D \times h_D$ = 360.23×40 = 14.4 KN-mm
- M_G = $H_G \times h_G$ = 570.02×21.5 = 12.2 KN-mm
- M_T = $H_T \times h_T$ = 47.25×30.75 = 1452.93 KN-mm
- M_o = $M_D + M_G + M_T$

= 28117.56 N-mm

Bolting up condition:

- H_G = W = $0.5(A_m + A_b)S_{all}$

= 0.5×1757.68(33.08 + 16(2.25))

= 60710.26 kgf

= 595.36 KN

M_a = $H_G \times h_G$ = 12.25 KN-m

M_{max} = M_o = 28.117 K N-m

M = M_{max}/B

= 28117×1000/578

= 48645.32 N

K = $\frac{A}{B}$

$$= \frac{708}{578}$$

$$K = 1.22$$

$$Y = \frac{1}{K-1}\left[0.66845 + \frac{5.7169K^2\log_{10}K}{K^2-1}\right]$$

$$= \frac{1}{1.22-1}\left[0.66845 + \frac{5.7169\times1.22^2\log_{10}\times1.22}{1.22^2-1}\right]$$

$$Y = 9.87$$

- Thickness of flange, t $= \sqrt{\frac{MY}{SF_o}}$

$$= \sqrt{\frac{48.64\times10^3\times9.87}{120.65}}$$

$$= 63 \text{ mm}$$

- Adding corrosion allowance and step thickness

$$t = 63 + 3.175 + 5 = 71.175 \text{ mm}$$

3. Tube sheet design:

For square tube pattern

- Thickness of tube sheet, $T = \frac{FG}{3}\sqrt{\frac{P_T}{\eta S_t}}$

where G = Gasket effective diameter,cm

η = ligment efficiency

P_T = Design pressure on tube side,kg/cm^2

- Ligment efficiency, $\eta = 1 - \frac{0.785}{\left(\frac{pitch}{tube\ OD}\right)^2}$

$$\eta = 1 - \frac{0.785}{\left(\frac{1.25\times20}{20}\right)^2}$$

$$\eta = 0.49$$

- For admiralty brass:

$$S_t = 968.73 \text{ kgf/cm}^2$$

$$T = \frac{1.25\times614.73}{3.175}\sqrt{\frac{14}{0.49\times968.73}}$$

$$T = 34.22 \text{ mm}$$

- Provided Thickness = T + (corrosion allowance)$_{p.p}$

$= 34.22 + 3.175$

$= 37.175$ mm

4. Nozzle design:

Channel side:

- Mass flow rate of hot fluid m_h = 77.25 kg/sec

$$m_h = \rho av$$

$$77.25 = 780\times\frac{\pi}{4}d^2{}_n\times1.95$$

I.D, d_n = 254 mm

O.D = 300 mm

Thickness of channel side nozzle, t_c = $\frac{46}{2}$ = 23 mm

Shell side:

- Mass flow rate of water m_c = 99.43 kg/sec

m_c = ρav

99.43 = $770\times\frac{\pi}{4}d^2\times1.36$

Inner diameter, d = 347 mm

Outer diameter, D = 400 mm

Thickness of Shell side nozzle, t = (53)/2

= 26.5 mm

5. BONNET DESIGN:

Bonnet thickness , t = $\frac{PLM}{2SE-0.2P}$

where,P = Design pressure on shell side, kg/cm^2

L = Crown radius , mm

S = Allowable stress, kg/cm^2

E = Joint efficiency

- Crown radius, L = 0.8D = 0.8×550

= 440 mm

- Knuckle radius ,r = 0.15D

= 0.15 × 550

= 82.5 mm

M = $\frac{3+\sqrt{(\frac{L}{r})}}{4}$

M = $\frac{3+\sqrt{(440/82.5)}}{4}$

= 1.66

- Required thickness,t = $\frac{PLM}{2SE-0.2P}$

= $\frac{14\times440\times1.66}{2\times968.73\times0.85-0.2\times14}$

= 6.22 mm

- Adding pressing allowance = 10%

t = 1.1×6.22

= 6.842 mm

- Provided thickness, t = 7 mm

CONCLUSION

- Thermal design of a shell and tube heat exchanger which uses a light naptha & kerosene is done according to TEMA standards.

- Using square tube pattern layout results in effective working and cleaning of heat exchanger.

- Mechanical design which includes design of shell, tube sheet , nozzle, flange, bonnet is done using ASME standards.

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