A

Project Report

on

DESIGNING OF SHELL AND TUBE HEAT EXCHANGER

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"DESIGNING OF SHELL AND TUBE HEAT EXCHANGER"

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CERTIFICATE

This is to certify that the work contained in this project titled "Designing of Shell and Tube Heat Exchanger" has been carried out by Mudit Baurai (R040307027), Mohit Patwal (R040307025) and Jibran Abrar (R040307016) under my supervision and has not been submitted elsewhere for a degree.

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Abstract

Shell and tube heat exchangers represent the most efficient and widely used equipment for the transfer of heat between the two process streams in industrial process applications. A shell and tube heat exchanger consists of a bundle of tubes enclosed in a high pressure vessel (shell). The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers. Shell and tube heat exchangers have the ability to transfer large amounts of heat in relatively low cost, serviceable designs. They can provide large amounts of effective tube surface while minimizing the requirements of floor space, liquid volume and weight.

In this project we have studied the design aspects for shell and tube heat exchanger for a crude oil and kerosene as fluid streams, and discussed the various problems associated with the heat exchanger in this case which are fouling, thermal expansion, mechanical stress and vibration problems. We have calculated the various design parameters such as shell outer diameter, tube inner diameter, tube pitch, pressure drop on both shell and tube side, required heat transfer area, heat transfer coefficient and velocity of fluids on both shell and tube sides through which efficient heat transfer can be obtained by following Kern's method and Tubular Exchange Manufacturers Association (TEMA) standards.

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1. INTRODUCTION

1.1 HEAT EXCHANGE PROCESS:

HEAT EXCHANGE PROCESS:-Heat exchange is a natural process of transfer of heat from a body, either solid or fluid at higher temperature to another body, either solid or fluid at lower temperature. Heat exchange between the two bodies takes place through three basic phenomena: Conduction, Convection and Radiation.

The basic laws governing the heat exchange between the two bodies are:-

1. FOURIER LAW OF HEAT CONDUCTION: The law of Heat Conduction, also known as Fourier's law, states that the rate of heat transfer per unit area is directly proportional to its temperature gradient.

$$Q = -k\,A\,(dT/dx)$$

Where k is the thermal conductivity and is the ability of the material to conduct heat.

Unit: SI : W/ mt^{0}C

FPS: Btu/ft hr^o F

The negative sign indicates that temperature decreases with the increase in direction of x.

Rate of heat conduction depends upon:

- Thickness
- Material
- Geometry
- Temperature difference

Area is measured perpendicular to direction of heat transfer

2. NEWTON'S LAW OF COOLING: The heat transfer from the solid surface to the fluid can be described by Newton's law of cooling. It states that the heat transfer, **dQ/dt**, from a solid surface of area **A**, at a temperature **Tw**, to a fluid of temperature **T**, is:

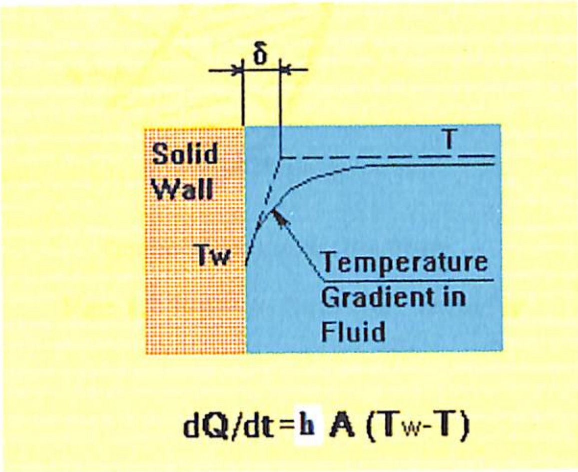


Fig: 1.1 Conduction through a wall

Where h is the heat transfer coefficient and is not the property of the material. It depends on:

- Nature of fluid (laminar or turbulent)
- Surface geometry

Units: $W/m^2\,^oK$

Heat transfer coefficient is more in forced convection than in natural convection

The general equation for heat transfer across the wall separating two fluids is given by:

$$Q=UA\Delta T_m$$

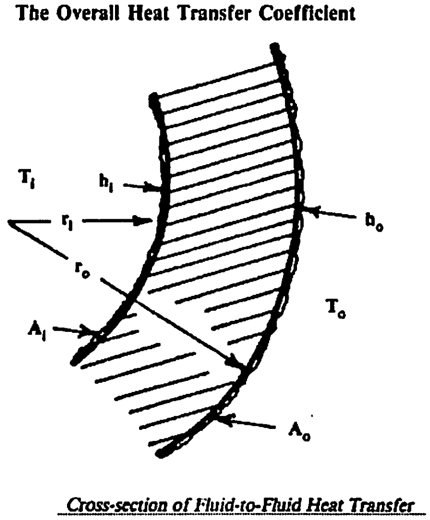


Fig: 1.2 Fluid to fluid heat transfer

Where,

Q = heat transferred per unit time, *W,*

U = the overall heat transfer coefficient, W/m2°C,

A — heat-transfer area, m2,

ΔTm = the mean temperature difference, the temperature driving force, °C.

The overall coefficient is the reciprocal of the overall resistance to heat transfer, which is the sum of several individual resistances. For heat exchange across a typical heat exchanger tube the relationship between the overall coefficient and the individual coefficients, which are the reciprocals of the individual resistances, is given by:

$$Uo = \frac{1}{\frac{1}{ho} + \frac{1}{hod} + \frac{do\ln(\frac{do}{di})}{2kw} + \frac{do}{di} \times \frac{1}{hid} + \frac{do}{di} \times \frac{1}{hid}}$$

Where,

U_0 = the overall coefficient based on the outside area of the tube, W/m2°C,

H_0 = outside fluid film coefficient, W/m2°C,

h_i = inside fluid film coefficient, W/m2°C,

h_{od} = outside dirt coefficient (fouling factor), W/m2°C,

h_{id} = inside dirt coefficient, W/m2°C,

kw = thermal conductivity of the tube wall material, W/m°C,

d_i = tube inside diameter, m,

d_i= tube outside diameter, m.

The magnitude of the individual coefficients will depend on the nature of the heat transfer process (conduction, convection, condensation, boiling or radiation), on the physical properties of the fluids, on the fluid flow-rates, and on the physical arrangement of the heat-transfer surface. As the physical layout of the exchanger cannot be determined until the area is known the design of an exchanger includes a trial and error procedure.

Mean temperature difference (Heat exchange driving force):

For counter-current flow, the logarithmic mean temperature is given by:

$$\Delta Tlm = \frac{\Delta T1 - \Delta T2}{ln\left(\frac{\Delta T1}{\Delta T2}\right)}$$

Where ΔT_{lm} = log mean temperature difference,

T_1 = inlet shell-side fluid temperature,

T_2 = outlet shell-side fluid temperature,

t1 = inlet tube-side temperature,

t2 = outlet tube-side temperature.

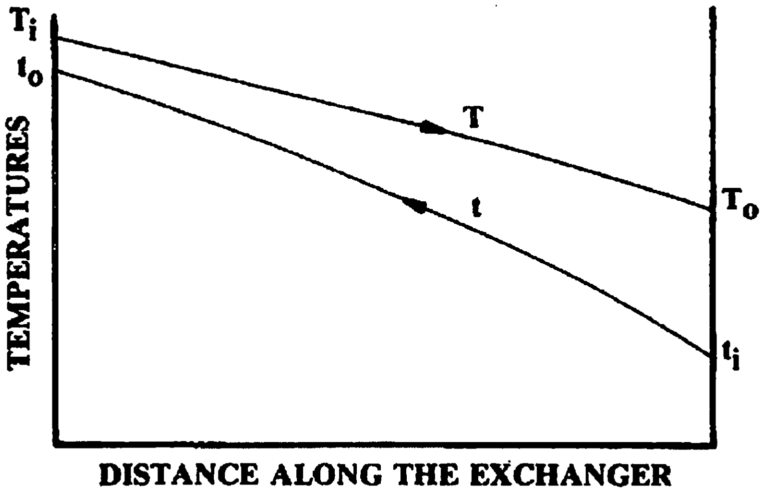


Fig: 1.3 LMTD for counter current

$$\Delta T_m = Ft \Delta T_{lm}$$

Where ΔTm — true temperature difference, the mean temperature difference for use in the design.

F_t = the temperature correction factor.

After distillation heat transfer is the most important operation in the refineries and other process plants, which is done with the help of heat exchangers. The process of heat transfer makes the existing operation more economic and cost effective by recovering and utilizing the waste heat in a very useful manner.

1.2 HEAT EXCHANGERS

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid surface and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single or multi-component fluid streams. In other applications, the objective may be to recover or reject heat, or sterilize, pasteurize, distil, fractionate, concentrate, crystallize, or control a process fluid.

In a few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, heat transfer between the two fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak. Such exchangers are referred to as direct transfer type, or simply recuperators. In contrast, exchangers in which there is intermittent heat exchange between the hot fluid and cold fluids via thermal energy storage and release through the exchanger surface or matrix are referred to as indirect transfer type, or simply regenerators. Such exchangers usually have fluid leakage from one fluid stream to the other, due to pressure differences and matrix rotation/valve switching. Common examples of heat exchangers are shell-and tube exchangers, condensers automobile radiators, evaporators, air pre-heaters, and cooling towers. If no phase change occurs in any one of the fluids in the exchanger, it is sometimes referred to as a sensible heat exchanger. There could be internal thermal energy sources in the exchangers, such as in electric heaters and nuclear fuel elements.

Chemical reaction and combustion may take place within the exchanger, such as in boilers, fired heaters, and fluidized-bed exchangers. Mechanical devices may be used in some exchangers such as in scraped surface exchangers, agitated vessels, and stirred tank reactors. Heat transfer in the separating wall of recuperators (direct transfer type) generally takes place by conduction. However, in a heat pipe heat exchanger, the heat pipe not only acts as a separating wall, but also facilitates the transfer of heat by condensation, evaporation, and conduction of the working fluid inside the heat pipe. In general, if the fluids are immiscible, the separating wall may be eliminated, and the interface between the fluids replaces a heat transfer surface, as in a direct-contact heat exchanger.

A heat exchanger consists of heat transfer elements such as a matrix or core containing the heat transfer surface, and fluid distribution elements such as headers, manifolds, tanks, inlet and outlet nozzles or pipes, or seals. Usually, there are no moving parts in a heat exchanger; however, there are exceptions, such as a rotary regenerative exchanger (in which the matrix is mechanically driven to rotate at some design speed) or a scraped surface heat exchanger.

The heat transfer surface is a surface of the heat exchanger that is in direct contact with fluids and through which heat is transferred by conduction. The portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as primary or direct surface. To increase the heat transfer area, appendages may be intimately connected to the primary surface to provide an extended, secondary, or indirect surface. These extended surface elements are referred to as fins. Thus, heat gets conducted through the fin and convected or radiated from the fin (through the surface area) to the surrounding fluid, or vice versa, depending on whether the fin is being heated or cooled. Therefore the addition of the fins to the primary surface decreases the side's thermal resistance and results in increases of the total heat transfer from the surface for the same temperature difference. Fins may form flow passages for the individual fluids but they do not separate the two (or more) fluids of the exchanger. These secondary surfaces or fins may also be introduced primarily for structural strength purposes or to provide thorough mixing of the highly viscous liquid.

Heat exchangers are not only often used in the process, power, cryogenic, petroleum, transportation, air-conditioning, refrigeration, heat recovery, alternative fuel, and manufacturing industries, they also serve as key components of many industrial products available in the marketplace.

1.3 Applications of heat exchanger

Heat transfer equipment such heat exchangers found a lot of applications in the following operations.

- **Petroleum Industry:**
  - Refinery - Brine cooling, crude oil/water interchanger, treated crude oil / untreated crude oil heat interchanger.
  - MTBE - Product heating, cooling & interchanging, jacket water cooling, condensing.
  - Alkylation - Hearing of corrosive fluids, Isobutene condenser and reactor interchanger, cooling.
  - Oil & Gas - Sea water coolers and crude oil heat treatment.
  - Dehydration/Desolving - Crude oil interchanger, water/crude oil interchanger, water interchanger, crude oil cooling.
  - Desulphurization - Lean/rich fluid interchanger & cooling, acid gas condenser.

- **Hydrocarbon Processing:**
  - Methanol - Preheating, water cooling, liquid product cooling.

- Propylene Oxide - Sodium hydroxide cooling, reaction mixture cooling, heat recovery from reactor bottom liquid.
- Ethylene Glycol - Product cooling, reactor feed interchanger and heater.
- Ethylene Oxide - Lean/rich cycle water interchanger, lean cycle water cooler.
- Formaldehyde - Methanol preheating, formalin cooling, water cooling.

- **Polymer:**
  - Acrylic Fibres - Heating & cooling of solvents, reactor cooling.
  - Nylon - Water cooling & heating, nylon salt cooling, condensers and interchangers.
  - Polyester - Glycol cooling, solvent heating.
  - Polyethylene - Pelleter water & cooling water systems.
  - Polyol - Water heating & cooling, reactor feed heating & cooling, polyol product cooling, polyol mix heating.
  - Polypropylene - Circulating, pellet, refrigerated and feed matercooling, condensers and interchangers.
  - Polyurethane - Heating & cooling of polyol and isocyanate.
  - PVC - Reactor jacket water cooling.
  - Viscose - Heating & Cooling of NaOH, viscose and different acids, heating of spin bath solution.
- **Pharmaceutical** - Product heating & cooling, cooling water systems, hot water systems, condensers & interchangers.

- **Mining** - Plating heaters & coolers, analyzing heaters & coolers, strike solution cooling, quench oil coolers, sulfuric acid, hydrochloric acid, hydrogen peroxide, titanium dioxide, chloride alkaline, soda ash, steel.
- **Automotive**
  - Pickling - Sulfuric and hydrochloric acid heating and cooling.
  - Rinsing - Heating rinsing water.
  - Phosphatizing - Phosphoric acid solution heating.
  - Passivating - Heat and maintain passicating bath temperature. Precutting Cooling of electrolytic bath solution (5-15% pain and water).
  - Priming & Painting - Heat and maintain paint temperature.

- **Pulp and Paper** - Blowdown liquor coolers, caustic soda coolers, black liquor heating, and boiler blowdown heat recovery.
- **Textile Applications** - Heat recovery, caustic solution heating & cooling washers.
- **Vegetable Oil** - Product heating, product cooling, product economizing, cooling water systems, hot water systems, waste water treatment.
- **Sugar** - Water, juice, syrup and molasses heating. Juice demineraliztion and evaporation.

o **HVAC** - Cooling tower isolation, free cooling, heat pump systems, sea water isolation, thermal storage systems, pressure Interceptor.

- **Marine** - Seawater isolation exchanger, central cooling, jacket fresh water cooling, lube oil cooling, camshaft lube, oil cooling.

1.4 CLASSIFICATION OF HEAT EXCHANGERS:

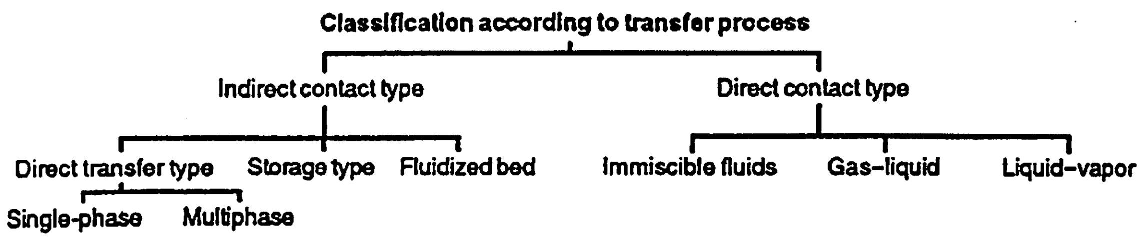
The heat exchangers can be classified according to transfer processes, number of fluids, surface compactness, construction, flow arrangement and heat transfer mechanisms.

1.4.1 CLASSIFICATION ACCORDING TO TRANSFER PROCESS:-

Heat exchangers are classified according to transfer process into indirect and direct contact types.

Indirect-contact heat exchangers:- In an indirect-contact heat exchanger, the fluid streams remain separate and the heat transfers continuously through an impervious dividing wall or into and out of a wall in a transient manner. Thus, ideally, there is no direct contact between thermally interacting fluids. This type of heat exchanger also referred to as a surface heat exchanger, can be further classified into direct-transfer type, storage type, and fluidized-bed exchangers.

Direct-contact heat exchangers:- In a direct-contact exchanger, two fluid streams come into direct contact, exchange heat, and are then separated. Common applications of a direct-contact exchanger involve mass transfer in addition to heat transfer, such as in evaporative cooling and rectification; applications involving only sensible heat transfer are rare. The enthalpy of phase change in such an exchanger generally represents a significant portion of the total energy transfer. The phase change generally enhances the heat transfer rate. Compared to indirectcontact recuperators and regenerators, in direct-contact heat exchangers, (1) very high heat transfer rates are achievable, (2) the exchanger construction is relatively inexpensive, and (3) the fouling problem is generally nonexistent, due to the absence of a heat transfer surface (wall) between the two fluids. However, the applications are limited to those cases where a direct contact of two fluid streams is permissible. These can be further classified into immiscible fluid exchangers, gas-liquid exchangers and liquid-vapor exchangers.



1.4.2 CLASSIFICATION ON ACCORDING TO NUMBER OF FLUIDS:- Most processes of heating, cooling, heat recovery, and heat rejection involve transfer of heat between two fluids. Hence, two-fluid heat exchangers are the most common. Three fluid heat exchangers are widely used in cryogenics and some chemical processes (e.g., air separation systems, a helium–air separation unit, purification and liquefaction of hydrogen, ammonia gas synthesis). Heat exchangers with as many as 12 fluid streams have been used in some chemical process applications. The design theory of three- and multifluid heat exchangers is algebraically very complex.

Classification according to number of fluids

- Two-fluid
- Three-fluid
- N-fluid ($N > 3$)

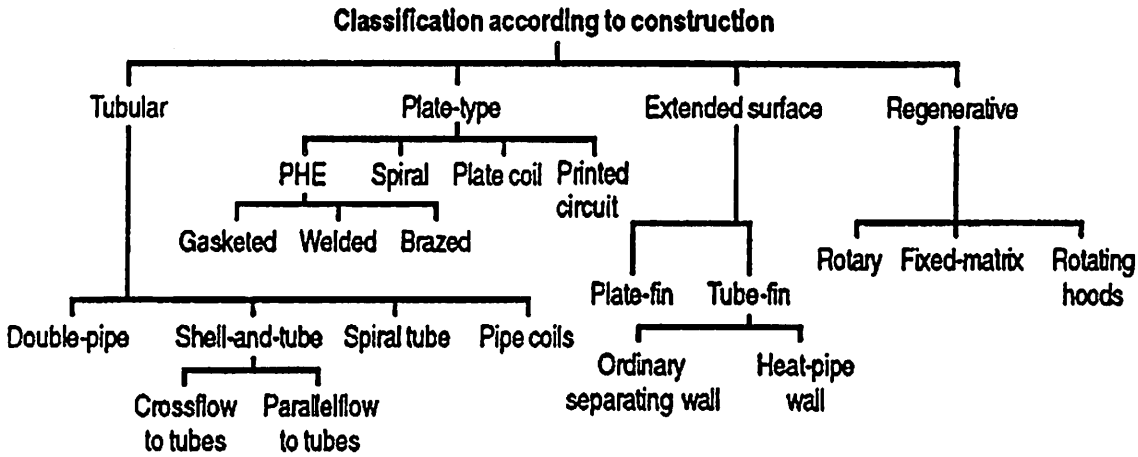
1.4.3 CLASSIFICATION ACCORDING TO SURFACE COMPACTNESS:- Compared to shell-and-tube exchangers, compact heat exchangers are characterized by a large heat transfer surface area per unit volume of the exchanger, resulting in reduced space, weight, support structure and footprint, energy requirements and cost, as well as improved process design and plant layout and processing conditions, together with low fluid inventory. These exchangers are categorised according to their surface area density (β) as: Gas-to-fluid exchanger, liquid to liquid phase.

Classification according to surface compactness

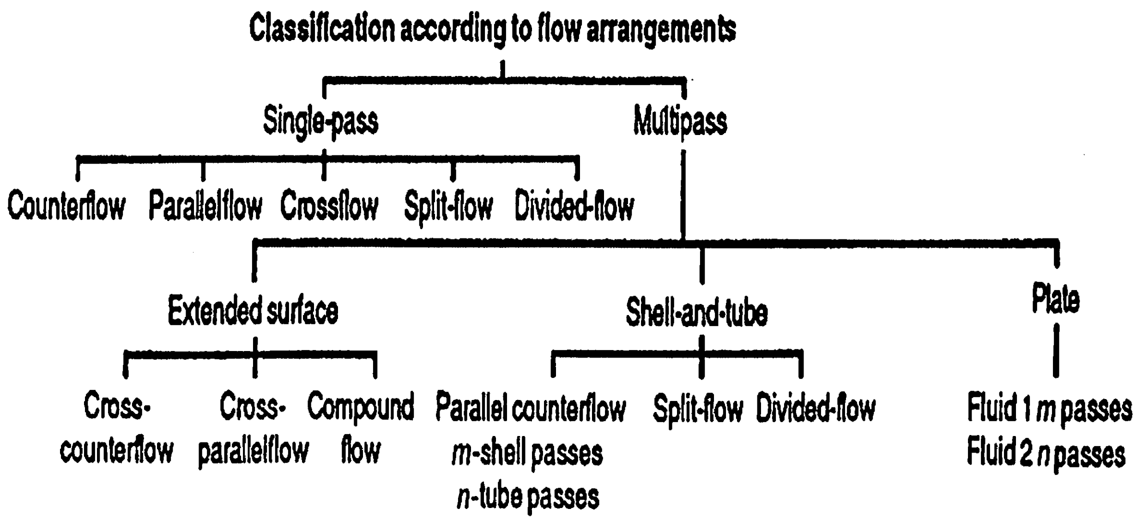
- Gas-to-fluid
  - Compact ($\beta \geqslant 700\ m^2/m^3$)
  - Noncompact ($\beta < 700\ m^2/m^3$)
- Liquid-to-liquid and phase-change
  - Compact ($\beta \geqslant 400\ m^2/m^3$)
  - Noncompact ($\beta < 400\ m^2/m^3$)

1.4.4 CLASSIFICATION ACCORDING TO CONSTRUCTION:- Heat exchangers are frequently characterized by construction features. Four major construction types are tubular, plate-type, extended surface, and regenerative exchangers.

Heat exchangers with other constructions are also available, such as scraped surface exchanger, tank heater, cooler cartridge exchanger, and others. Some of these may be classified as tubular exchangers, but they have some unique features compared to conventional tubular exchangers. Since the applications of these exchangers are specialized, we concentrate here only on the four major construction types noted above. Although the "-NTU and MTD methods are identical for tubular, plate-type, and extended-surface exchangers, the influence of the following factors must be taken into account in exchanger design: corrections due to leakage and bypass streams in a shell-and-tube exchanger, effects due to a few plates in a plate exchanger, and fin efficiency in an extended-surface exchanger.

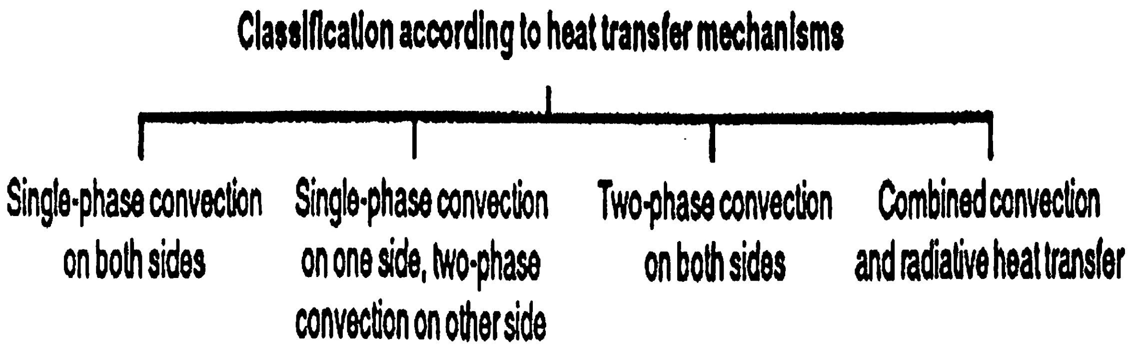


1.4.5 CLASSIFICATION ACCRODING TO FLOW ARRANGEMENT:- A number of flow arrangement are possible in a heat exchanger. The choice of a particular flow arrangement is dependent on the required exchanger effectiveness, available pressure drops, minimum and maximum velocities allowed, fluid flow paths, packaging envelope, allowable thermal stresses, temperature levels, piping and plumbing considerations, and other design criteria.



1.4.6 CLASSIFICATION ACCORDING TO HEAT TRANSFER MECHANISM:-
The basic heat transfer mechanisms employed for transfer of thermal energy from the

fluid on one side of the exchanger to the wall (separating the fluid on the other side) are single-phase convection (forced or free), two-phase convection (condensation or evaporation, by forced or free convection), and combined convection and radiation heat transfer. Any of these mechanisms individually or in combination could be active on each fluid side of the exchanger. Some examples of each classification type are as follows. Single-phase convection occurs on both sides of the following two-fluid exchangers: automotive radiators and passenger space heaters, regenerators, intercoolers, economizers, and so on. Single-phase convection on one side and two-phase convection on the other side (with or without desuperheating or superheating, and subcooling, and with or without noncondensables) occur in the following two-fluid exchangers: steam power plant condensers, automotive and process/power plant air-cooled condensers, gas or liquid heated evaporators, steam generators, humidifiers, dehumidifiers, and so on. Two-phase convection could occur on each side of a two-fluid heat exchanger, such as condensation on one side and evaporation on the other side, as in an air-conditioning evaporator. Multicomponent two-phase convection occurs in condensation of mixed vapors in distillation of hydrocarbons. Radiant heat transfer combined with convective heat transfer plays a role in liquid metal heat exchangers and high-temperature waste heat recovery exchangers. Radiation heat transfer is a primary mode in fossil-fuel power plant boilers, steam generators, coal gasification plant exchangers, incinerators, and other fired fired heat exchangers.



1.5 SHELL AND TUBE HEAT EXCHANGER:

A **shell and tube heat exchanger** is a class of heat exchanger designs. It is the most common type of heat exchanger used in oil refineries and other large chemical processes, and is suited for higher-pressure applications. As its name implies, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes, and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids. The set of tubes is called a tube bundle, and may be composed by several types of tubes: plain, longitudinally finned, etc.

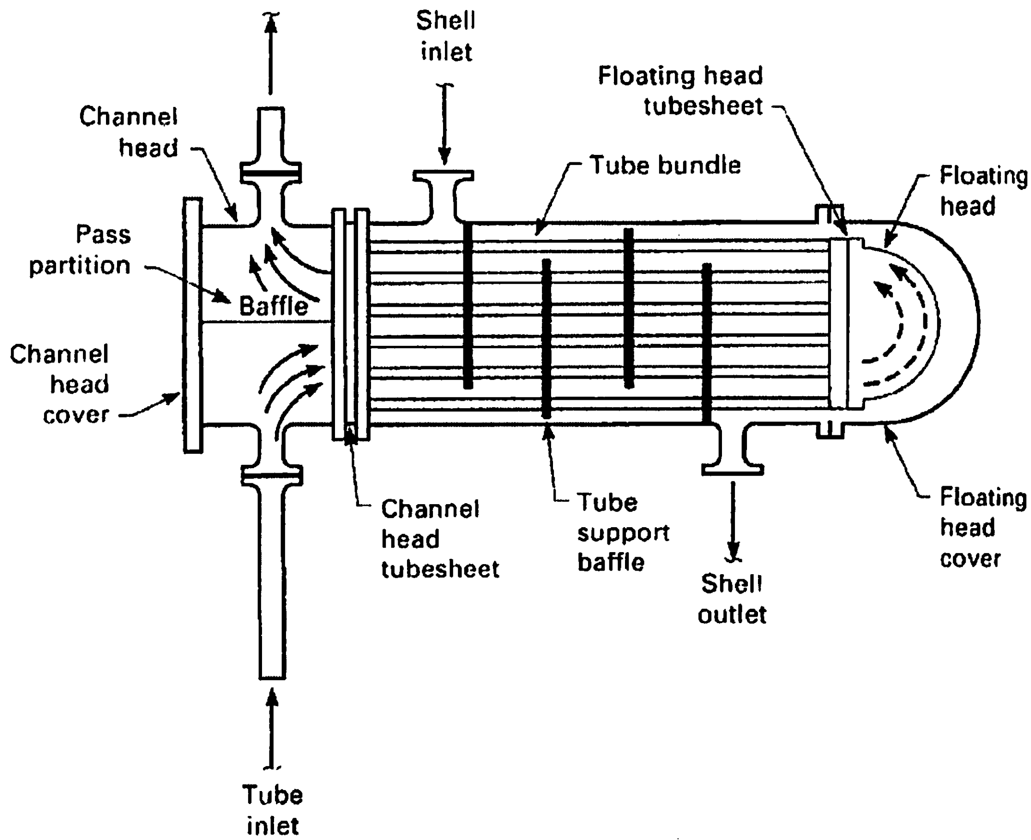


Fig: 1.4 Shell and tube heat exchanger

1.5.1 Specification of Shell and Tube Heat Exchanger:-

Shell and Tube heat exchangers are most widely used heat transfer equipments in chemical industry. Sizes of various parts of shell and tube heat exchangers like shell, tubes, tierods, are standardized. Shell and tube heat exchangers are generally designed according to ASME (American Society of Mechanical Engineers) and TEMA (Tubular Exchanger Manufacturers Association) codes and standards which not only specifies the design method but also the manufacturing methods for each part. TEMA or IS 4503 specify standard sizes of shell, tube, tierods, etc. and also maximum allowable baffle spacing, minimum tube sheet thickness, baffle thickness, no. of tierods required, etc.

In TEMA standard, shell and tube heat exchangers are classified in three categories:

- Class R: Covers heat exchangers which are used for severe duties in petroleum and related industries.
- Class B: Covers heat exchangers which are used in chemical process industries not involving severe duties, and
- Class C: Covers heat exchangers which are used in commercial and in less important process applications.

1.5.2 TYPES OF SHELL AND TUBE HEAT EXCHANGER:

A shell and tube heat exchanger can be classified according to their construction:

Fixed Tubesheet: A fixed-tubesheet heat exchanger has straight tubes that are secured at both ends to tubesheets welded to the shell. The construction may have removable channel covers, bonnet-type channel covers, or integral tubesheets. The principal advantage of the fixed tubesheet construction is its low cost because of its simple construction. In fact, the fixed tubesheet is the least expensive construction type, as long as no expansion joint is required. Other advantages are that the tubes can be cleaned mechanically after removal of the channel cover or bonnet, and that leakage of the shellside fluid is minimized since there are no flanged joints.

A disadvantage of this design is that since the bundle is fixed to the shell and cannot be removed, the outsides of the tubes cannot be cleaned mechanically. Thus, its application is limited to clean services on the shell side. However, if a satisfactory chemical cleaning program can be employed, fixed-tubesheet construction may be selected for fouling services on the shell side. In the event of a large differential temperature between the tubes and the shell, the tubesheets will be unable to absorb the differential stress, thereby making it necessary to incorporate an expansion joint. This takes away the advantage of low cost to a significant extent.

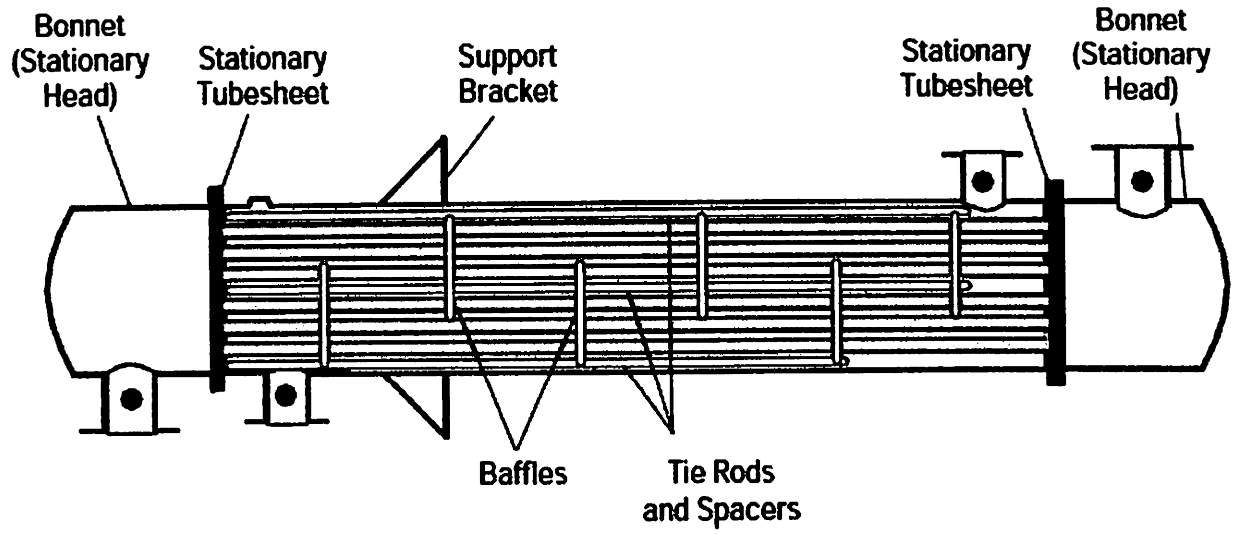


Fig 1.5 Fixed tube sheet

U-Tube: As the name implies, the tubes of a U-tube heat exchanger are bent in the shape of a U. There is only one tubesheet in a U-tube heat exchanger. However, the lower cost for the single tube sheet is offset by the additional costs incurred for the bending of the tubes and the somewhat larger shell diameter (due to the minimum U-bend radius), making the cost of a U-tube heat exchanger comparable to that of a fixedtubesheet exchanger. The advantage of a U-tube heat exchanger is that because one end is free, the bundle can expand or contract in response to stress differentials. In addition, the outsides of the tubes can be cleaned, as the tube bundle can be removed. The disadvantage of the U-tube construction is that the insides of the tubes cannot be cleaned effectively, since the U-bends would require flexible- end drill shafts for cleaning. Thus, U-tube heat exchangers should not be used for services with a dirty fluid inside tubes.

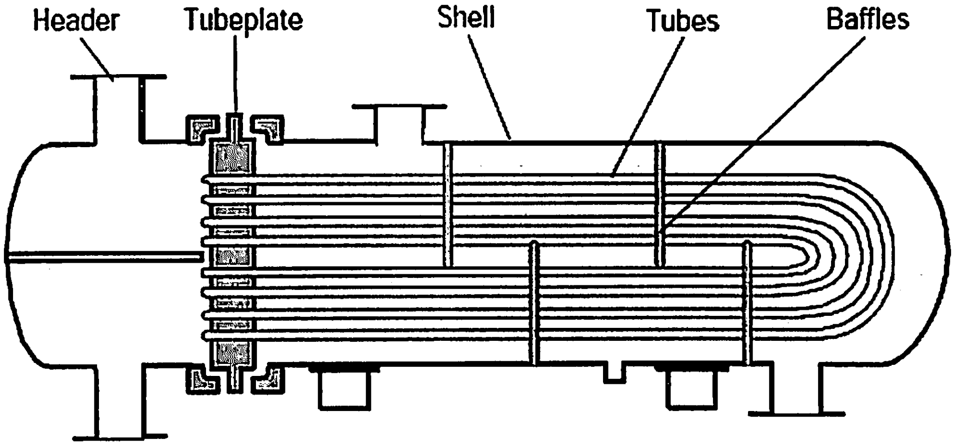


Fig: 1.6 U-tube

. **Floating head**: The floating-head heat exchanger is the most versatile type of Shell and tube heat exchanger, and also the costliest. In this design, one tube sheet is fixed relative to the shell, and the other is free to "float" within the shell. This permits free expansion of the tube bundle, as well as cleaning of both the insides and outsides of the tubes. Thus, floating-head Shell and tube heat exchangers can be used for services where both the shell side and the tube side fluids are dirty — making this the standard construction type used in dirty services, such as in petroleum refineries.

There are various types of floating- head construction. The two most common are the pull-through with backing device (TEMA S) and pull through (TEMA T) designs. The TEMA S design is the most common configuration in the chemical process industries (CPI). The floating-head cover is secured against the floating tube sheet by bolting it to an ingenious split backing ring. This floating-head closure is located beyond the end of the shell and contained by a shell cover of a larger diameter. To dismantle the heat exchanger, the shell cover is removed first, then the split backing ring, and then the floating-head cover, after which the tube bundle can be removed from the stationary end. In the TEMA T construction, the entire tube bundle, including the floating-head assembly, can be removed from the stationary end, since the shell diameter is larger than the floating-head flange. The floating head cover is bolted directly to the floating tube sheet so that a split backing ring is not required. The advantage of this construction is that the tube bundle may be removed from the shell without removing either the shell or the floating head cover, thus reducing maintenance time. This design is particularly suited to kettle reboilers having a dirty heating medium where U-tubes cannot be employed. Due to the enlarged shell, this construction has the highest cost of all exchanger types.

Floating head heat exchangers are further classified into two types:

Split-ring floating head heat exchanger: Split-ring floating head heat exchanger is widely used in the chemical process industries. In this configuration split backing ring is provided. Compared to pull thorough floating head heat exchanger, this requires less diameter for shell to accommodate the same number of tubes. More maintenance time is required. In maintenance, shell cover is removed first, then the split backing ring, then floating head cover and finally tube bundle is removed from the stationary end.

Pull through floating head heat exchanger: Pull through floating head heat exchanger is the costliest type of shell and tube heat exchanger because it requires largest shell diameter for the given numbers of tubes. In this type of heat exchanger it is possible to remove the entire tube bundle including floating head assembly from the stationary end.

TEMA designations for shell and tube heat exchangers:

According to TEMA designation a shell and tube heat exchanger is divided into three parts: the front head, the shell, and the rear head. These parts are designated by three different letters. First letter indicate type of front head, second letter indicates type of shell and third indicates type of rear head.

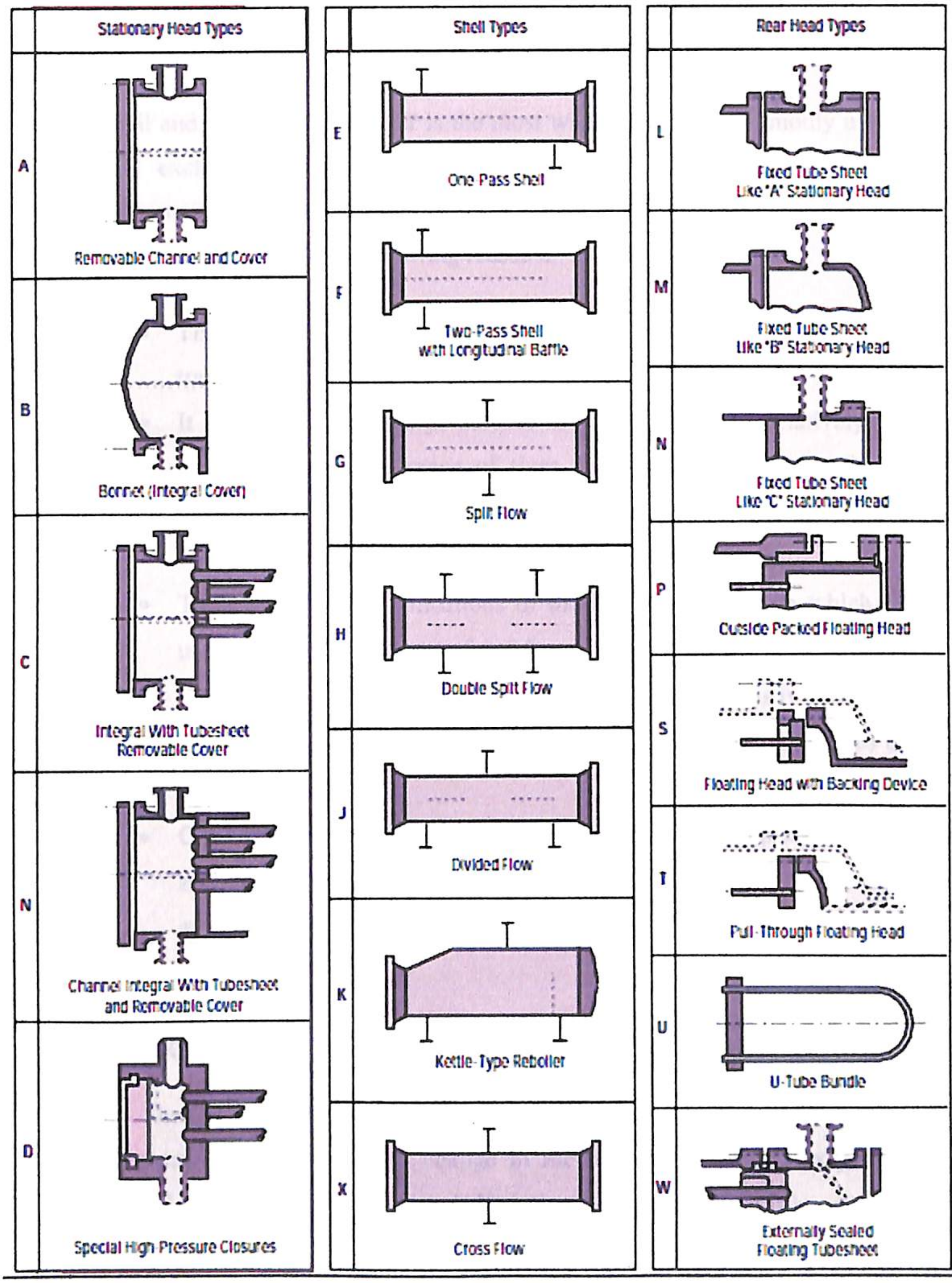


Fig: 1.7 TEMA Designation for shell and tube heat exchanger

1.5.3 Why Shell and tube is preferred –advantages over others – industrial application (economic.)

Shell and tube heat exchanger is the most widespread and commonly used basic heat exchanger configuration in the process industry. Shell and tube heat exchanger is generally preferred over other heat exchangers for most of the industrial process due to following reasons:-

- The Shell and tube heat exchanger provides a comparatively ratio of heat transfer area to volume and weight.
- It provide heat exchange surface in a form which is relatively easy to construct in a wide range of sizes and which is mechanically rugged enough to withstand normal shop fabrication stresses, shipping and field erection stresses, and normal operating conditions.
- There are many modifications of the basic configuration which can be used to solve various sort of problems.
- The Shell and tube heat exchanger can be reasonably easily cleaned.
- The components most subjected to failure like gaskets and tubes can be easily replaced.
- Good design methods exist and the expertise and shop facilities for the successful design and construction of Shell and tube heat exchanger are available throughout the world.

1.5.4 MAIN COMPONENTS OF SHELL AND TUBE HEAT EXCHANGER:-

SHELL: - Shell is the costliest part of the heat exchanger. Cost of Shell and tube heat exchanger sensitively changes with change in the diameter of shell. As per TEMA standard, shell size ranges from 6 inches (152mm) to 60 inches (1520mm).

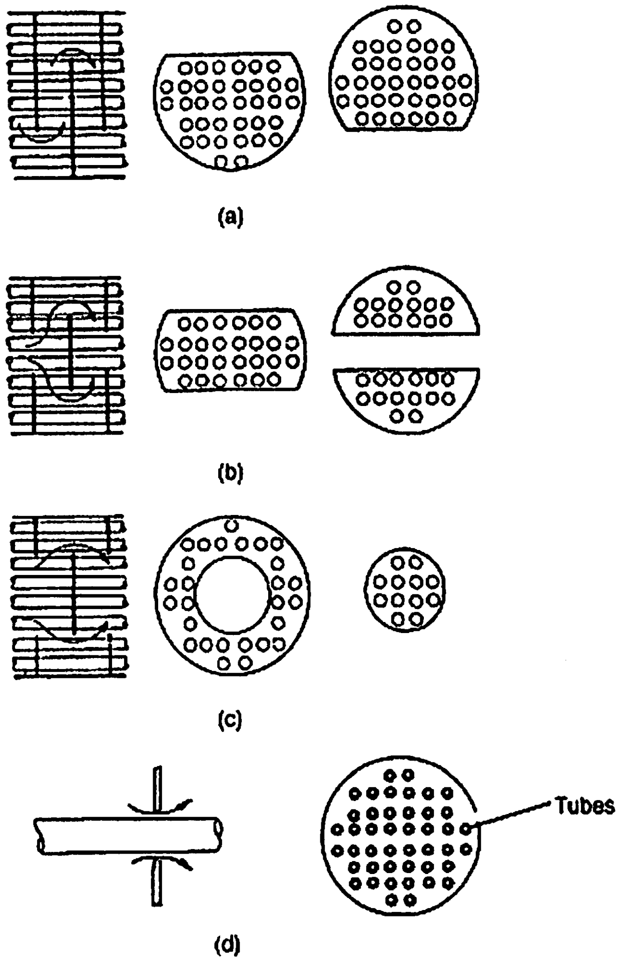
Shell diameter depends on tube bundle diameter. For fixed tube sheet Shell and tube heat exchanger, the gap between shell and tube bundle is minimum ranging from 10 to 20 mm. For pull through floating head heat exchanger, it is maximum, ranging from 90-100 mm.

BAFFLE: - Baffles are the integral part of shell and tube exchanger Baffle serves two important functions:

- Baffles are used in shell to direct the fluid stream across the tubes to increase the velocity of shell side fluid and thereby to improve shell side heat transfer coefficient. In other words baffles are used in shell to increase the turbulence in shell side fluid. This function is used in no phase change case in shell side fluid.
- Baffles indirectly support the tubes and thereby reduce the vibration in tubes. If shell side liquid velocity is higher; say more than 3 m/s, vibrational analysis calculation should be carried out to check whether baffle spacing is sufficient or not.

Different types of baffles used in shell and tube heat exchanger;

1. Segmental baffle
2. Nest baffle
3. Segmental an stripped baffle
4. Disc and doughnut baffle
5. Orifice baffle
6. Damped baffle



Types of baffle used in shell and tube heat exchangers. (*a*) Segmental (*b*) Segmental and strip (*c*) Disc and doughnut (*d*) Orifice

Fig: 1.8 Types of baffles

Most widely used baffle is segmental baffle. Other types of baffles like nest baffle, segmental and stripped baffle and disc and doughnut baffle provide less pressure drop for the same baffle spacing but provide lower heat transfer coefficient as compared to segmental baffle.

Tube side pass partition plate: - Tube side passes are provided to decrease the tube side flow area and to increases tube side fluid velocity thereby to improve the tube side heat transfer coefficient at the expense of pressure drop. More number of tube side passes are recommended only if there is no change in the phase of tube side fluid.

Tie rods: - Baffles are supported by tie rods. Tie rods are made from solid metal bar. Normally four or more tie rods are required to support the baffles. Diameter of tie rod is less than the diameter of tube. Diameter and number of tie rods required for given shell diameter are specified by TEMA.

Spacer: - Spacers are used to maintain space between baffles. Spacers are the pieces of pipes or in the most of the cases they are the pieces of extra available tubes. Spacers are passed over the tie rods and because of them baffles do not slide over tie rods under the effect of the force of fluid. Hence, spacers fix the location of baffles and maintain the space between them. Length of spacer is equal to space between the baffles.

Tube sheet: - Tubes and one end of tie rods are attached to tube sheet.One surface of tube sheet is exposed to tube side fluid and other surface is exposed to shell side fluid. This point is very important in the selection of material for tube sheet and.also in determining tube sheet thickness.

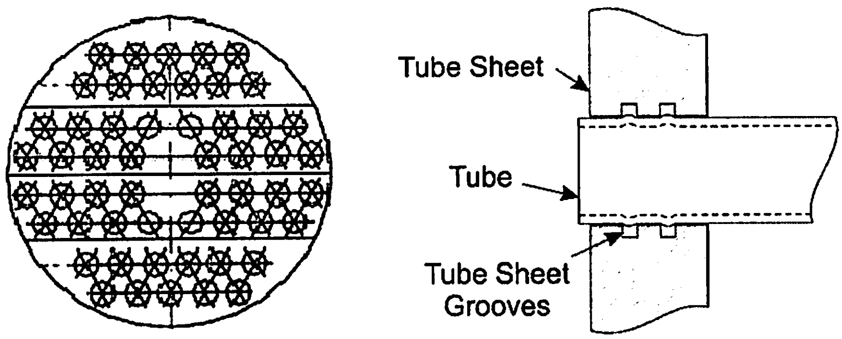


Fig: 1.9 Tube sheet

Sealing strip: - Tube bundles have pairs of metal strips set around the edge of tube bundle. These metal strips are typically ¼ thick and 4in wide. They extend down the length of tubes. There are two functions of sealing strips. Sealing strips reduce the amount of bypass stream of shell side fluid flowing through the clearance between shell inside diameter and tube bundle diameter and thereby improve the shell side heat transfer coefficient. Sealing strips also make the removal of tube bundle from the shell easy. Hence they are also known as sliding strips.

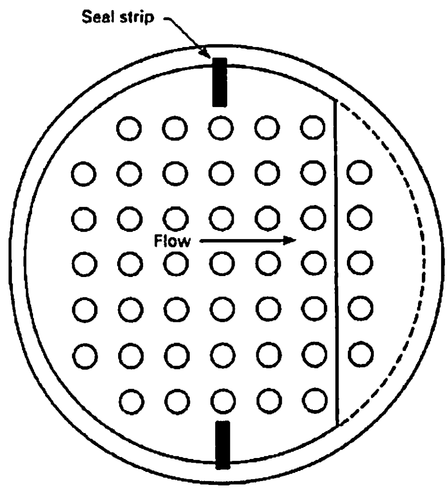


Fig: 1.10 Sealing strip

Tube:- The tubes are the basic components of the Shell and tube heat exchanger, providing the heat transfer surface between one fluid flowing inside the tube and the other fluid flowing across the outside of the tube. Tubes may be seamless or welded and most commonly made of copper or steel alloys. Other alloys of nickel, titanium, or aluminium may also be required for specific application.

Tube side pass partition plate:- Tube side passes are provided to decrease the tube side flow area and to increase tube side fluid velocity thereby to improve the tube side heat transfer coefficient at the expense of pressure drop. This is true only if there is no phase change on tube side. Hence, more number of tube side passes is recommended only if there is no phase change in the phase of tube side fluid.

Tube side passes are very common and are advantageous use for improving tube side heat transfer coefficient. These passes can be achieved in many ways by locating partition plates in channel covers.

1.6 Basic Operation and Terminology:-

1.6.1 Flow pattern in shell side: Flow pattern in the shell side is given by the Tinker flow model. On the shell side, there is not just one stream, but a main cross-flow stream and four leakage or bypass streams, as illustrated in Figure Tinker *(4)* proposed calling these streams the main cross-flow stream(B), a tube-to-baffle-hole leakage stream (A), a bundle bypass stream (C), a pass-partition bypass stream (F), and a baffle-to-shell leakage stream (E). While the B (main cross-flow) stream is highly effective for heat transfer, the other streams are not as effective. The A stream is fairly efficient, because the shell side fluid is in contact with the tubes. Similarly, the C stream is in contact with the peripheral tubes around the bundle, and the F stream is in contact with the tubes along the pass-partition lanes. Consequently, these streams also experience heat transfer, although at a lower efficiency than the B stream. However, since the E stream flows along the shell wall, where there are no tubes, it encounters no heat transfer at all. The fractions of the total flow represented by these five streams can be determined for a particular set of exchanger geometry and shell side flow conditions by any sophisticated heat exchanger thermal design software. Essentially, the five streams are in parallel and flow along paths of varying hydraulic resistances. Thus, the flow fractions will be such that the pressure drop of each stream is identical, since all the streams begin and end at the inlet and outlet nozzles. Subsequently, based upon the efficiency of each of these streams, the overall shell side stream efficiency and thus the shell side heat-transfer coefficient is established.

Since the flow fractions depend strongly upon the path resistances, varying any of the following construction parameters will affect stream analysis and thereby the shell side performance of an exchanger:

- Baffle spacing and baffle cut
- Tube layout angle and tube pitch
- Number of lanes in the flow direction and lane width.
- Clearance between the tube and baffle hole.
- Clearance between the shell I.D. and the baffle.
- Location of sealing strips and sealing rods.

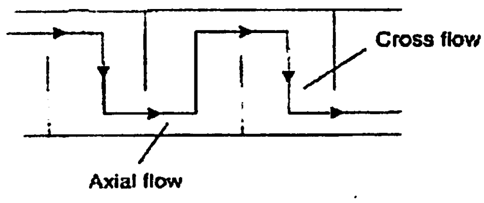
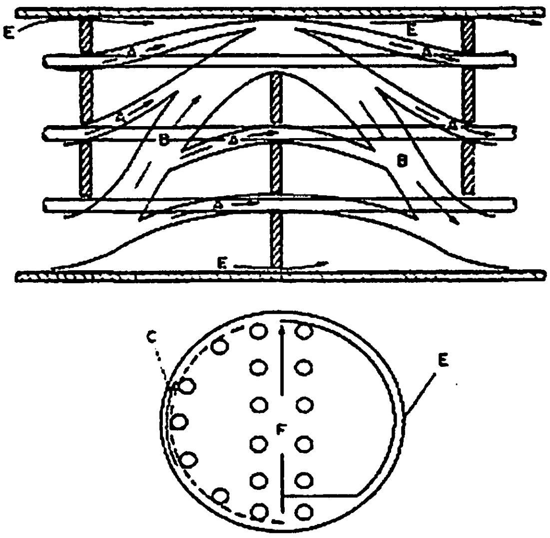


Figure 12.25. Idealised main stream flow



Shell-side leakage and by-pass paths

Fig 1.11 Tinker flow

1.6.2 Vortex shedding: Vortex shedding is a flow phenomenon, which is related to the concept of cross-flow velocity of the fluid in the heat exchanger. When a fluid flows perpendicularly across a tube, vortices, are created. The resulting turbulence that forms behind a tube will cause the tube to vibrate. But at the same time the turbulence that forms behind the tube in a heat exchanger promotes good heat transfer.

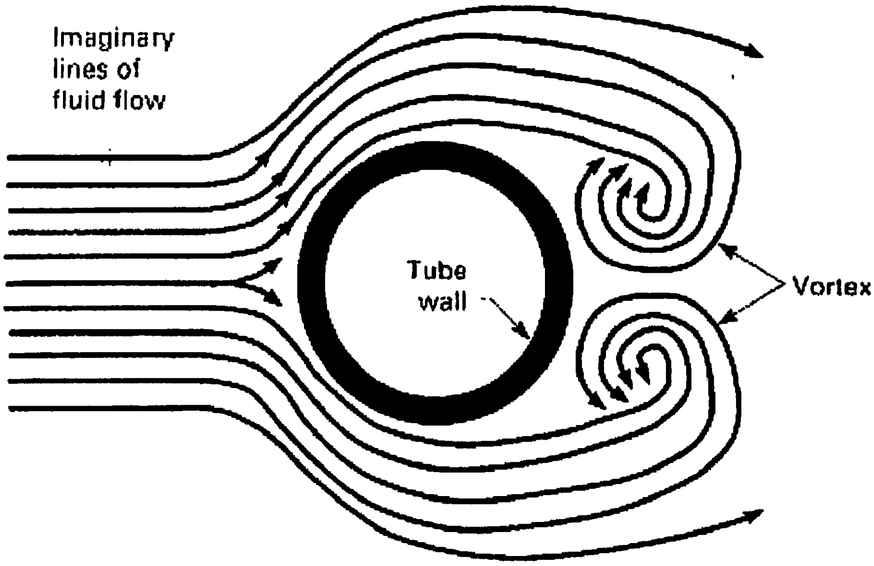


Fig: 1.12 Vortex shedding

The turbulence always encourages good sensible-heat transfer therefore it should be increased in the shell side by increasing the cross-flow velocity.

1.6.3 Tube Pitch: The tube pitch is the centre to centre distance of the two adjacent tubes.

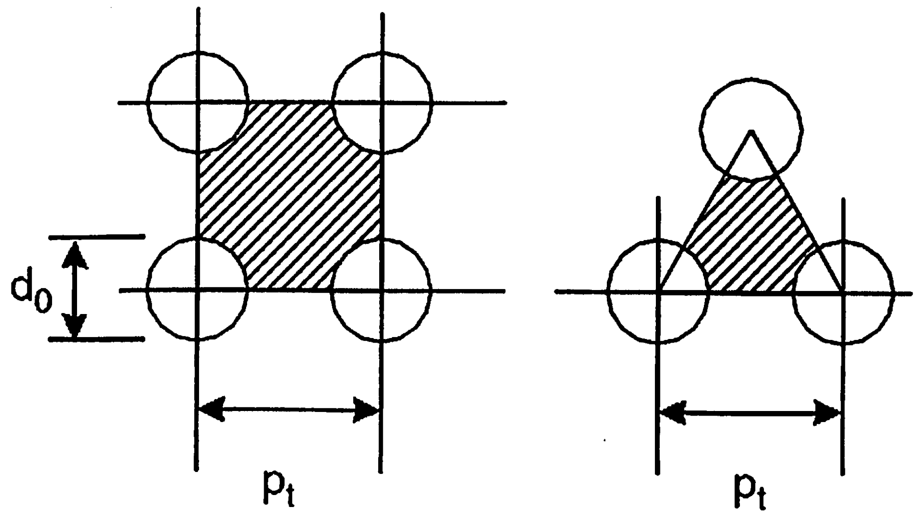


Fig 1.13 Tube pitch

Where, P_t is the tube pitch and D_0 is the outside diameter of the tube

1.6.4 Fouling factor: The most common way to account for the fouling effects in a tubular heat exchanger is the fouling factor. The fouling factor is a predetermined number that represents the amount of fouling that a particular heat exchanger transferring a particular fluid will sustain. In the heat transfer equation the fouling factor is added to the other thermal resistances to calculate the total thermal resistance which is the reciprocal of U_{clean}. There are no direct calculations to determine the appropriate fouling factor to use for a given fluid in a particular application, however guidelines do exist to help determine an appropriate fouling factor. The most common compilation of fouling factors, to be used for a variety of fluid in various applications, is supplied by Tubular Exchanger Manufacturers Association (TEMA). The below table is a list of general fouling factors used for shell and tube heat exchangers and common fluids and applications.

Table 1.1 Fouling factors for different fluids

Fluid	Fouling Resistance(ft^2-^oF-hr/BTU)
Transformer Oil	0.001
Steam	0.0005
Compressed Air	0.001
Hydraulic Fluid	0.001
Glycol Solutions	0.002
Refined Lube Oil	0.001
Sea Water	0.0005 (up to 125°F) 0.001 (over 125°F)
Cooling Tower Water	0.001 (up to 125°F) 0.002 (over 125°F)
River Water (minimum) (tube velocity≤3 fps)	0.002 (up to 125°F) 0.003 (over 125°F)
River Water (minimum) (tube velocity>3 fps)	0.001 (up to 125°F) 0.002 (over 125°F)
River Water (average) (tube velocity≤3 fps)	0.002 (up to 125°F) 0.003 (over 125°F)
River Water (average) (tube velocity>3 fps)	0.002 (up to 125°F) 0.003 (over 125°F)
River Water (muddy or silty) (tube velocity≤3 fps)	0.003 (up to 125°F) 0.004 (over 125°F)
River Water (muddy or silty) (tube velocity>3 fps)	0.002 (up to 125°F) 0.003 (over 125°F)

2. DESIGN PROCEDURE FOR SHELL AND TUBE EXCHANGER:

2.1 Data required for designing:

For designing of shell and tube heat exchanger the following data must be known:

1. Flow rates of both streams.
2. Inlet and outlet temperatures of both streams.
3. Operating pressure of both streams. This is required for gases, especially if the gas density is not known. It is not really necessary for liquids, as their properties do not vary with pressure.
4. Allowable pressure drop for both streams.
5. Fouling resistance for both streams.
6. Physical properties of both streams. These include viscosity, thermal conductivity, density, and specific heat, preferably at both inlet and outlet temperatures. Viscosity data must be supplied at inlet and outlet temperatures, especially for liquids, since the variation with temperature may be considerable.
7. Heat duty. The duty specified should be consistent for both the shell side and the tube side.
8. Type of heat exchanger. If not furnished, the designer can choose this based upon the characteristics of the various types of construction described earlier.
10. Preferred tube size. Tube size is designated as O.D. ′ thickness ′ length.
11. Maximum shell diameter. This is based upon tube-bundle removal requirements and is limited by crane capacities.
12. Materials of construction. If the tubes and shell are made of identical materials, all components should be of this material. Thus, only the shell and tube materials of construction need to be specified.
13. Special considerations. These include cycling, upset conditions, alternative operating scenarios, and whether operation is continuous or intermittent.

The phase change of the fluid in the shell and tube heat exchanger is one of the major aspects which must be taken into account. Here the design is considered for no phase change only.

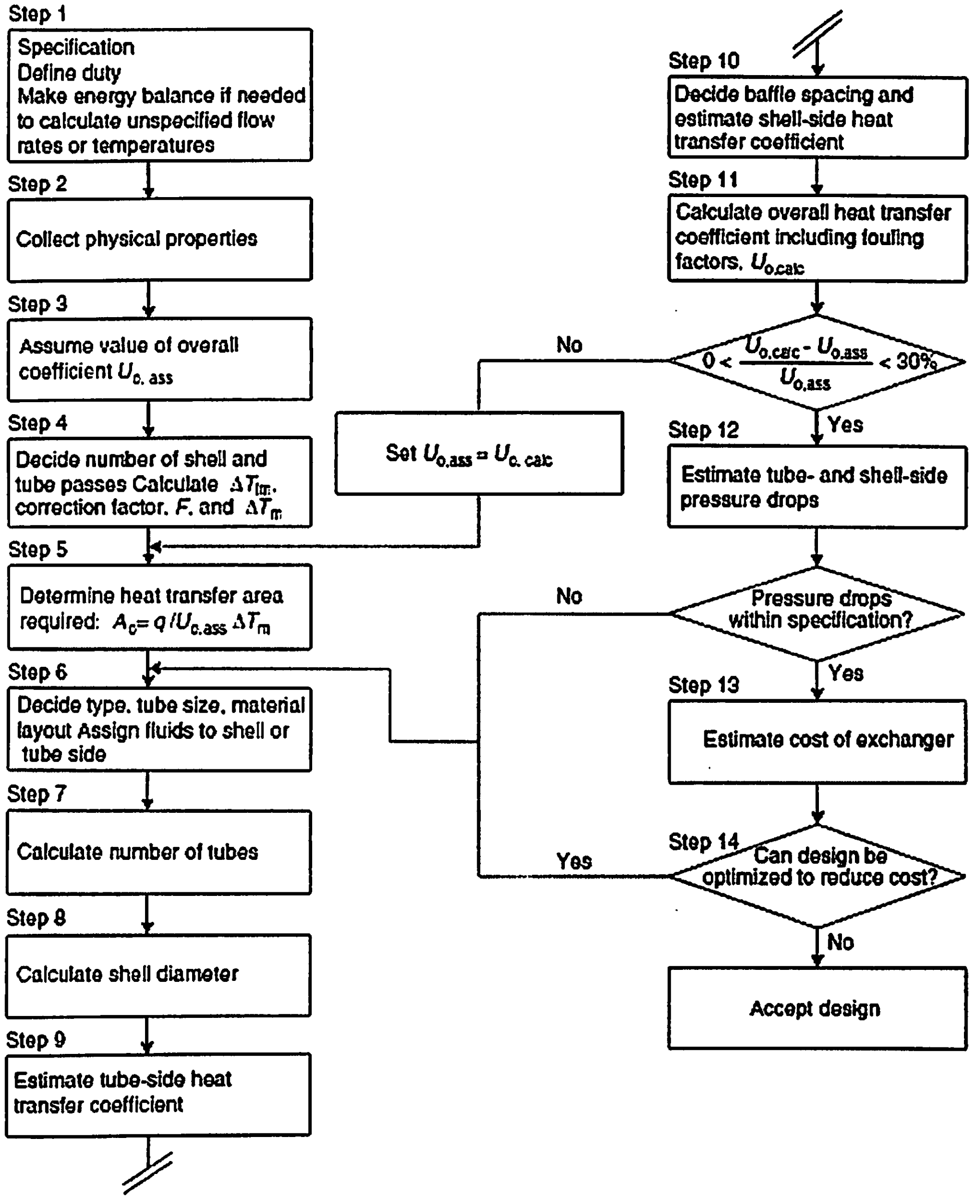


Fig: 2.1 Algorithm for designing

2.2 Steps for designing

2.2.1 Calculation of Heat Duty

a) For cooling or heating for no phase change, heat duty is calculated by equation:

$$\Phi = \dot{m} C p \Delta t$$

$\dot{m}$ = Mass flow rate of fluid, kg/s

C_p= Specific heat of fluid, KJ/kg^{0}C

Δt= Temperature difference to be carried out, ^0C

Φ= Heat duty required, KW

b) For condensation with sub cooling

$\Phi = \dot{m}\lambda + \dot{m}C_L\Delta T$

$\dot{m}$= mass flow rate of vapours, kg/s

C_L= Specific heat of condensation, KJ/kg^{0}C

λ= Latent heat of vaporization, KJ/kg

2.2.2 Mean Temperature Difference

Mean temperature difference (MTD) is calculated from equation

$\Delta T_m = Ft\Delta T_{lm}$

ΔT_{lm}= Logarithmic mean temperature difference, ^0C

$$\Delta Tlm = \frac{\Delta T1 - \Delta T2}{ln\left(\frac{\Delta T1}{\Delta T2}\right)}$$

ΔT_2 and ΔT_1 are terminal temperature differences

$\Delta T_1 = T_1 - t_2$

$\Delta T_2 = T_2 - t_1$

T_1= hot fluid temperature, inlet,

T_2= hot fluid temperature, outlet,

t_1= cold fluid temperature, inlet,

t_2= cold fluid temperature, outlet.

For 1-1 heat exchangers (one shell side pass and tube side pass), F_t=1

When shell side passes and/or tube side passes are more than one, F_t must be determined:

F_t= f(R,S) where R and S are temperature ratios

$$R = \frac{T1 - T2}{t1 - t2}$$

$$S = \frac{t2 - t1}{T1 - t1}$$

2.2.3 Fluid allocation:

a) In case of phase change: Condensing or vaporizing fluid is normally taken on shell side as in the shell liquid or vapour phases are easily separated. In case of a condenser, if condensation is carried out in tube side, then the entire liquid condensate forms a film on heat transfer surface which provides additional to heat transfer.

b) In case of no phase change: If there is no phase change in shell side and tube side fluid, then fluid allocation depends on following factors:

- Corrosion: Corrosive fluid is allocated in tube side as the repairing of only tubes is much economic than repairing of tube bundle and inner surface of shell.
- Fouling: In fixed tube sheet tube heat exchanger inside surface of tubes can be easily cleaned and hence the fluid which has the greatest tendency to foul on heat transfer surface should be placed in the tubes.
- Fluid Temperature: At very high temperature as well as at very low temperature use of especial alloy is required. Hence very hot or very cold fluid is placed in tube side to avoid the use of costly material for the shell. At moderate temperature, hotter fluid is better passed through tubes. If it is placed on shell side more insulation is required not only to reduce the heat loss but also for safety purpose.
- Operating Pressure: Higher pressure stream should be placed on tube side as high pressure tubes are cheaper than high pressure shell. Diameter of shell is very much greater than diameter of tubes hence its thickness is more sensitively changed which change in pressure as compared to that of the tubes.
- Fluid flow rates and Viscosity: Very low value of flow rate of fluid and high value of viscosity of fluid give low value of Reynolds number. Fluid which provides very low value of Reynolds number should be placed on shell side as the dependency of shell side heat transfer coefficient on Reynolds number is less as compared to the same of tube side heat transfer coefficient. Hence for the lower value of Reynolds number. shell side heat transfer coefficient is higher than tube side heat transfer coefficient.
- Low Heat transfer coefficient: If one stream has an inherently low heat transfer coefficient such as low pressure gases or viscous liquids, this stream is

preferentially put on the shell side so that extended surface may be used to reduce the total cost of heat exchanger.

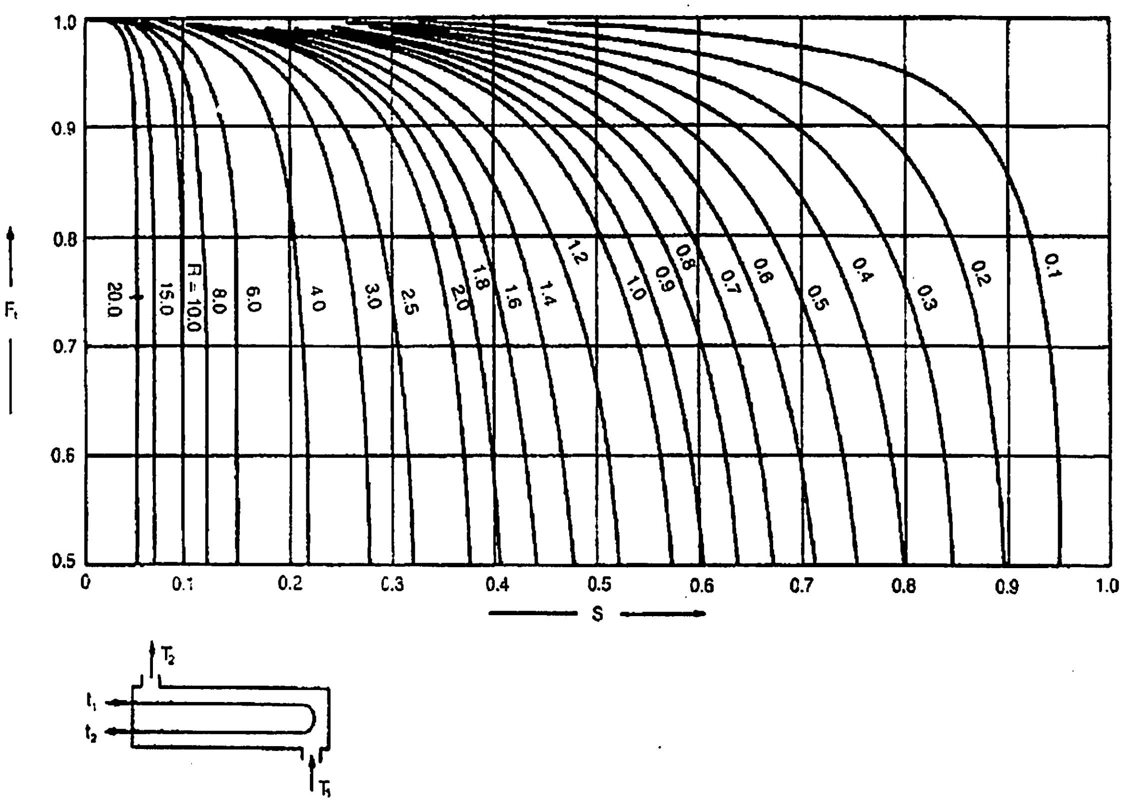


Fig 2.2 F_t Vs S graph

Graphs of F_t vs S for the different values of R, for the different cases given in figure gives values of F_t for even number of tube passes. If $F_t<0.75$, economic design of shell and tube heat exchanger cannot be achieved.

2.2.4 Overall coefficients for shell and tube Heat exchangers

Table 2.1 Overall heat transfer coefficient for different fluids

Shell Side	Tube side	U, $W/m^{2\,0}C$
	[A] Heat Exchangers	
Water	Water	800-1400
Organic Solvent	Water	300-850
Organic Solvent	Brine	200-500
Light Oil	Water	350-900
Heavy Oil	Water	60-300
Water	Brine	600-1200
Gas	Brine	15-250

Gas	Gas	10-50
Gas	Water	20-300
Naptha	Water	300-400
Lube oil	Water	150-500
Gasoline	Water	350-550
Kerosene	Water	150-300
Water	10 to 30%	500-1250
[B] Condensers		
Organic Vapours	Water	700-1000
Organic vapour + Gas	Water or brine	100-300
Low boiling hydrocarbons vapour mixtures	Water	450-1100
Naptha vapour	Water	300-425
Steam	Water	2000-5000
Alcohol vapour	Water	500-1000
Saturated organic vapour	Water or brine	280-680
[C] Vaporizers/Reboilers		
Light Organics	Steam	800-1200
Heavy Organics	Steam	600-900
Aqueous solution	Steam	1000-1500
Chlorine	Steam	700-1500
Ammonia	Steam	700-1500
Water	Steam	1250-2000
Refrigerants	Water	425-850

2.2.5 Heat transfer area

Heat Transfer Area based on this selected or assumed value of U.

$$A = \frac{\Phi}{U \times \Delta\mathrm{Tlm}}$$

A is the heat transfer area provided or actual heat transfer area of heat exchanger.

$$A = A_{pro} = N_t \prod d_0 L$$

d_0=Outside diameter of tube, m

L= Tube length, m

N_t= Total number of tubes

Values of d_0 and L are decided by designer

2.2.6 Bundle Diameter

Based on all these information inside diameter of shell is calculated.

$$D_b=d_0(N_t/k_1)^{(1/n1)}$$

D_b= Tube bundle diameter, mm

d_0= Tube outside diameter, mm

N_t= Total number of tubes

K_l and n_l are constants, values of which depend on ratio P_t/d_0 [=tube pitch/tube OD]

A. For P_t/d_0=1.25, triangular pitch

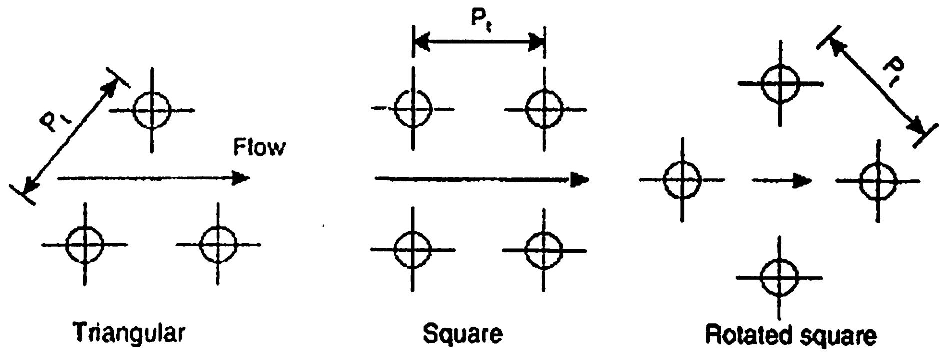
Table 2.2 traingular pitch

No. of tube side passes	1	2	4	6	8
k_l	0.319	0.249	0.175	0.0743	0.0365
n_l	2.142	2.207	2.285	2.499	2.675

B. For P_t/d_0=1.25, square pitch

Table 2.3 square pitch

No. of tube side passes	1	2	4	6	8
k_l	0.215	0.156	0.158	0.0402	0.0331
n_l	2.207	2.291	2.263	2.617	2.643



Tube patterns

Fig: 2.3 Tube patterns

2.2.7 Tube side heat transfer coefficient

For heating and cooling on tube side(no phase change), tube side heat transfer coefficient is determined by Siede-Tate equation. If Re ≤ 2000.

$$Nu = \frac{\text{hidi}}{\text{ki}} = 1.86\left(Re.Pr.\frac{d}{L}\right)^{0.33}\left(\frac{\mu}{\mu \text{w}}\right)^{0.14}$$

If Nu ≤ 3.5 then Nu is taken 3.5

If Re ≥ 4000, tube side heat transfer coefficient is determined by Dittus-Boelter equation:

$$Nu = \frac{hidi}{ki} = C.Re^{0.8}.Pr^{0.33}\left(\frac{\mu}{\mu w}\right)^{0.14}$$

Where, Nu = Nusselt number = $\frac{hi.di}{ki}$

Re = Reynolds number = $\frac{di\mu i\rho}{\mu} = \frac{diGt}{\mu}$

Pr = Prandtl number = $\frac{Cp\mu}{k}$

hi = Tube side heat transfer coefficient, W/(m^2 ^{0}C)

di = Tube ID, m

L = length of tube, m

μ = Viscosity of fluid at the bulk fluid temperature, (N.s/m^2)

μ_w= Viscosity of fluid at tube wall temperature, (N.s/m^2)

C= constants

C= 0.21 for gases

= 0.023 for non-viscous liquid = 0.027 for viscous liquid

G_t= Tube side mass velocity, Kg/(m^2.s)

$$G_t = \frac{\dot{m}}{at}$$

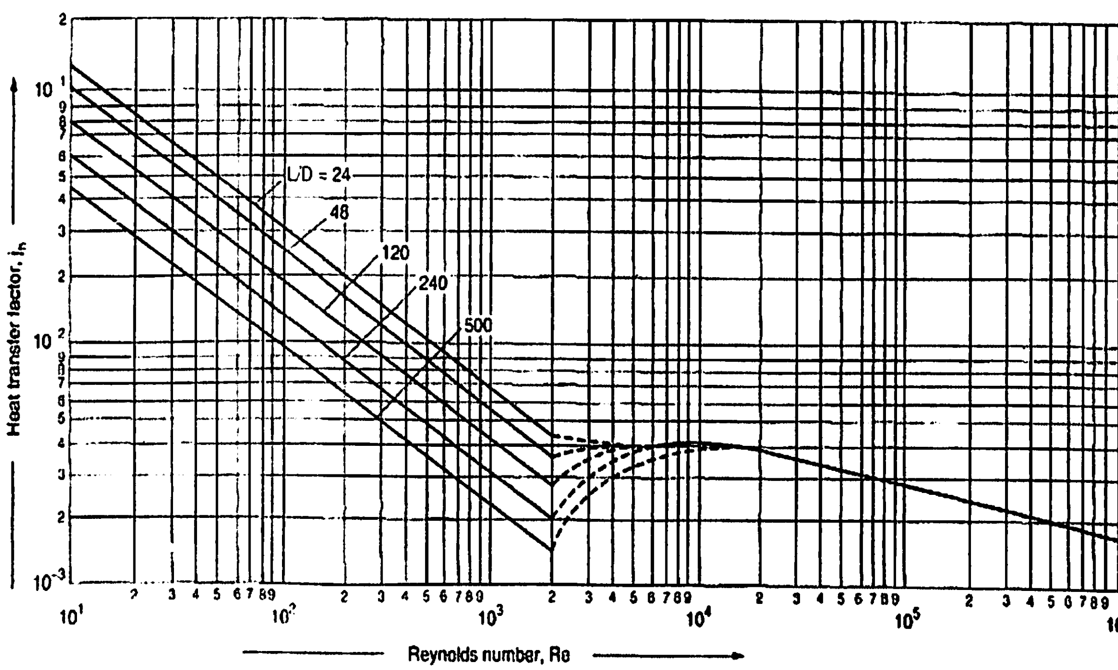
Where, a_t = Tube side flow area $= \frac{Nt}{Np} \times \frac{\Pi}{4} di^2$

U_t = Tube side velocity $= \frac{Gt}{\rho}$ m/s

ρ = Density of fluid, kg/m^3

Tube side heat transfer coefficient h_i can be calculated from the value of "heat transfer factor, J_h", for the entire range of Reynolds number (from Re=10 to 10^6), equation relating h_i and J_h is

$$N_u = h_i \frac{di}{k} = \text{Jh.Re.} Pr^{0.33} \left(\frac{\mu}{\mu w}\right)^{0.14}$$



Tube-side heat-transfer factor

Fig: 2.4 J_h Vs Re (L/D)

2.2.8 Tube side pressure drop

For no phase change in tube side fluid, pressure drop is calculated:

$$\Delta Pt = Np\left(8Jf\left(\frac{L}{di}\right)\left(\frac{\mu}{\mu w}\right)^{-m} + 2.5\right)\frac{\rho ut^2}{2}$$

M= 0.25 for Re≤ 2100 and m= 0.14 for Re>2100

Where, ΔP_t = Tube side friction pressure drop, N/m^2 or Pa

N_p = number of tube side passes

J_f = tube side friction factor

ρ = Density of tube side fluid, kg/m^3

U_t = Tube side fluid velocity, m/s

ΔP_t is actually permanent pressure loss. Calculated pressure drop (loss) should be less than maximum allowable pressure drop. In some applications, maximum allowable pressure drop is decided by process conditions. While in other applications, maximum allowable pressure drop is actually optimum pressure drop (loss). Heat exchanger design means the balance between two opposite factors, heat transfer coefficients related to fixed cost and pressure drop related to operating cost. Increase in heat transfer coefficient by modifying the heat exchanger design also increases the pressure drop. Hence, ideally actual pressure drop should be equal to optimum pressure drop which gives the total cost of heat exchanger (fixed cost + operating cost) minimum.

2.2.9 Shell Side Heat Transfer Coefficient

For heating or cooling on shell side or for no phase change on shell side

a) Shell side flow area

$$As = \frac{(Pt-d0)DsBs}{Pt}$$

As = Shell side flow area, m^2

P_t = Tube pitch, m

d_o= Outside diameter of tube, m

B_s = Baffle spacing, m

D_s = Shell inside diameter, m

b) Shell side mass velocity G_s and linear velocity (u_s)

$$Gs = \frac{\dot{m}s}{As} \quad and \quad us = \frac{Gs}{\rho s}$$

G_s = Shell side mass velocity, kg/(m^2.s)

$\dot{m}s$ = Shell side mass flow rate of fluid, kg/s

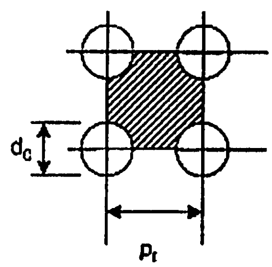
ρs = Density of shell side fluid, kg/m^3

c) Shell side equivalent diameter(d_e) for square pitch arrangement

$$de = \frac{4 \times cross\ sectional\ area}{wetted\ perimeter}$$

$$= \frac{4 \times \left(Pt^2 - \frac{\Pi}{4} d0^2\right)}{\prod do}$$

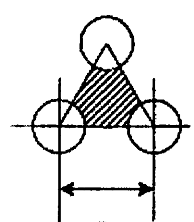
$$= \frac{127}{do}(Pt^2 - 0.0785do^2)$$



For triangular pitch arrangement

$$de = \frac{4 \times \left(0.5Pt \times Pt sin\ 60 - \frac{\Pi}{8} do^2\right)}{(\prod do/2)}$$

$$de = \frac{1.1}{do}(Pt^2 - 0.907do^2)$$



d) Shell side Reynolds number

$$Re = \frac{deGs}{\mu}$$

μ = Viscosity of shell side fluid at average temperature, kg/(m.s)

Prandtl number

$$Pr = \frac{Cp\mu}{k}$$

C_p, μ, k are the properties of shell side fluid at average temperature.

e) Shell side heat transfer coefficient

$$Nu = \frac{hodo}{k} = 0.036\, Re^{0.55} Pr^{0.33} (\frac{\mu}{\mu w})^{0.14}$$

This correlation is valid for the range of Reynolds number from 2000 to 1000000. Shell side heat transfer coefficient can be found from "shell side heat transfer factor. J_h".

$$Nu = \frac{hode}{k} = Jh.Re.Pr^{0.33} \left(\frac{\mu}{\mu w}\right)^{0.14}$$

Graph for J_h vs Re

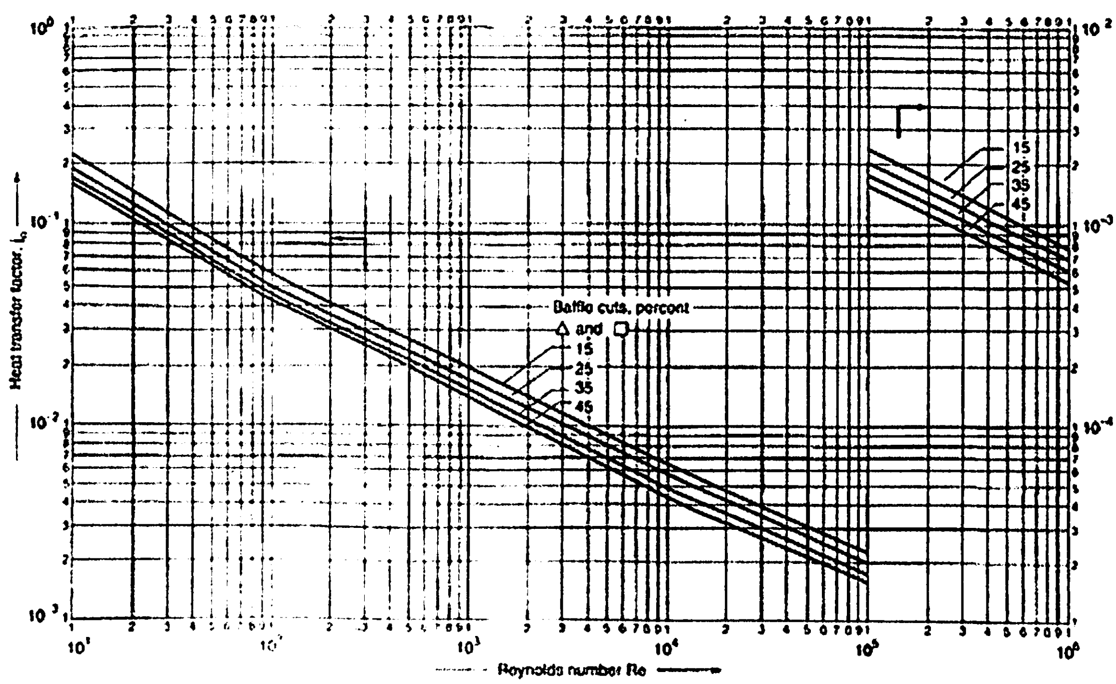


Fig: 2.5 J_h Vs Re (baffle cut)

2.2.10 Shell Side pressure drop

For no phase change on shell side or for heating or cooling on shell side, shell side pressure drop ΔP_s

$$\Delta Ps = 8Jf\left(\frac{Ds}{de}\right)\left(\frac{L}{Bs}\right)\frac{\rho sus^2}{2}\left(\frac{\mu}{\mu w}\right)^{-0.14}$$

Table 2.4 Optimum pressure drop based on economic consideration

Fluid	Optimum, ΔP_t or ΔP_s, Kpa
Liquids of $\mu_L < 1$ cP	33
Liquids of $\mu_L = 1$ to 10 cP	50 to 70
Gas or Vapour at 1 to 2 atm	13.8
Gas or vapour at high vacuum (up to 60 torr absolute pressure)	0.4 to 0.8 (3 to 6 torr)
Gas or vapour at high pressure > 10 atm	0.1 times operating pressure

2.2.11 Overall Heat transfer coefficient

$$Uo = \frac{1}{\frac{1}{ho} + \frac{1}{hod} + \frac{doln(\frac{do}{di})}{2kw} + \frac{do}{di} \times \frac{1}{hid} + \frac{do}{di} \times \frac{1}{hid}}$$

U_0= Overall heat transfer coefficient based on outside area of tubes. W/(m^2 ^{0}C)

2.3 PROBLEMS IN SHELL AND TUBE HEAT EXCHANGER:-

2.3.1 Thermal stress:-

Since the shell of the heat exchanger will be at a significantly different temperature than tubes, the shell will expand and contract relatively to the tubes, resulting in a stress existing in both components and being transmitted through the tube sheets. This results in buckling of the shell and damage of the equipment. The fixed tube sheet exchangers are especially vulnerable to this kind of damage because there is no provision made for accommodating differential expansion.

Solution for thermal stress:-

- Expansion joint on the shell: - The most obvious solution to the thermal expansion problem is to put expansion roll or joint in the shell. This becomes less attractive for large diameter shells or increasing shell-side pressure. However, very large diameter or high pressure shells have been designed with a partial ball-joint in the shell designed to allow the shell to partially rotate to accommodate stresses.
- Internal Bellows: - Internal Bellows are generally used in vertical thermosyphon reboilers, where only one pass is permitted on the tube side. These bellows have designed to operate successfully with high pressure boiling water on the tube side and high temperature reactor effluent gas on the shell.

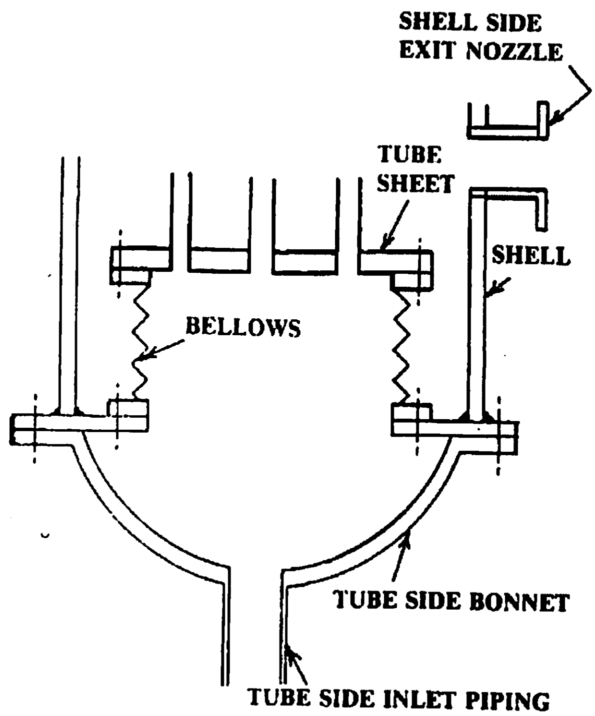


Fig: 2.6 Bellows

- The U-Tube Exchanger: - It is a design variation that allows independent expansion of tubes and shell is the U-tube configuration. Though this design can solve the problem of thermal expansion but it is very difficult to replace individual tubes except in the outer row in this configuration. The other major problem in this design is that it is very difficult to clean the tubes around the bends.
- Floating head design: - This configuration reduces thermal stresses and provide a means to remove the tube bundle for cleaning, several floating rear head designs have been established. The simplest floating head design is the "pull-through bundle" type. One to the tube sheet is made small enough that it and its gasketed bonnet may be pulled completely through the shell for shell-side inspection and cleaning. The tube side may be cleaned and individual tubes may be replaced without removing the bundle from the shell. The problem in this design is that many tubes must be omitted from the edge of the full bundle to allow for the bonnet flange and bolt circle. This problem can be solved in the "split-ring floating head" type by bolting the floating head bonnet to a split backing ring rather than to the tube sheet. At some cost in mechanical complexity, most of the tubes lost from the bundle in the pull-thorough design have been restored, and other features retained. The other types, the "outside-packed lantern ring" and "outside-packed stuffing box", but these are more prone to leakage to the atmosphere than the foregoing types and give up the advantage of positive sealing which is so important in high pressure or hazardous fluid service. They have the advantage of allowing single tube side pass design.

2.3.2 Mechanical stresses: -

Sources of Mechanical stresses: - Every exchanger is subject to mechanical stresses from a variety of sources in addition to temperature gradients. There are mechanical stresses which result from the construction techniques used on the exchanger, e.g., tube and tube sheet stresses resulting from rolling in the tubes. During the manufacture, shipping and installation of the exchanger there are many, frequently unforeseen stresses imposed. There are stresses caused by the support structure reacting to the weight of the exchanger, and stresses from the connecting piping; these stresses are generally very different during normal plant operation than during construction or shutdown. Finally, there are the stresses arising within the exchanger as a result of the process stream conditions – especially pressure – during operation.

Provision For Mechanical Stress: - To protect the exchanger from permanent deformation or weakening from these mechanical stresses, it is necessary to design the exchanger so that any stress that can be reasonably expected to occur will not strain or deform the metal beyond the point where it will spontaneously return to its original condition.

2.3.3 Vibrational problem:-

A very serious problem in the mechanical design of heat exchangers is flow – induced vibration of the tubes. There are several harmful consequences of tube vibration. The tube may vibrate against the baffles, which are eventually cut holes in the tubes. In extreme cases, the tubes can strike adjacent tubes, literally knocking holes in each other. The repeated stressing of the tube near a rigid support such as a tube sheet can result in fatigue cracking of tube, loosening of the tube joint, and accelerated corrosion. Vibration is caused by repeated unbalanced forces being applied to the tube. There are a number of such forces, but the most common one in heat exchanger is the eddying motion of the fluid in the wake of a tube as the fluid flows across the tube. The unbalanced forces are relatively small, but they occur tens, hundreds, or thousands of times a second, and their magnitudes increase rapidly with increased fluid velocity. Even so, these forces are ordinarily damped out with no damage to the tube. However, anybody can vibrate more easily at certain frequencies (called "natural frequencies") than at others. If the unbalanced forces are applied at "driving frequencies" that are at or near these natural frequencies, resonance occurs; and even small forces can result in very strong vibration of the tube.

The two best ways to avoid vibration problems are to support the tubes as rigidly as possible (e.g., close baffle spacing) and to keep the velocities low.

2.3.4 Fouling:-

One of the most common operational challenges encountered with heat exchangers is fouling. Fouling is the build up of sediments and debris on the surface area of a heat exchanger that inhibits heat transfer. Fouling will reduce heat transfer, impede fluid flow, and increase the pressure drop across the heat exchanger. As with many operational concerns, proper planning at the design stage can minimize the effects of fouling down the road.

Types of fouling

Crystallization: is one of the most common type of fouling. Certain salts commonly present in natural waters have a lower solubility in warm water than cold. Therefore, when cooling water is heated during the cooling process (particularly at the tube wall) these dissolved salts will crystallize on the surface in the form of scale.

Sedimentation: the depositing of dirt, sand, rust, and other small matter is also common when fresh water is used. This can be controlled to a degree by the heat exchanger design.

Biological Organic growth material: occurs from chemical reactions, and can cause considerable damage when built up.

Chemical Reaction Coking: appears where hydrocarbon deposits in a high temperature application.

Corrosion: can destroy surface areas of the heat exchangers, creating costly damage. Fouling will slow down heat transfer and damage equipment unless it is dealt with accordingly.

Freezing Fouling: results from overcooling at the heat transfer surface causing solidification of some of the fluid stream components.

Design consideration to decrease effects of fouling:-

- A high level of turbulence keeps sediments from settling on the surface of the heat exchanger, and also helps clean off any fouling so it is important to ensure that the design velocities are high enough to mitigate fouling but not too high to promote erosion.

- Try to keep a uniformly high velocity throughout the entire exchanger, so that sediments are not able to settle. Keep the amount of low velocity turns and 'dead' spots to a minimum, so that fouling will not accumulate.
- Consider how often the unit needs to be cleaned, and provide easy access to make this process easier.

Given data:

Kerosene

Table 3.1 kerosene properties

	parameter	Value	Unit
1	Mass flow rate	20000	Kg/hr
2	Inlet temperature(T_1)	200	°C
3	Outlet temperature(T_2)	90	°C
4	Specific heat	2.47	KJ/kg°C
5	Thermal conductivity	0.13	W/mt°C
6	Density	730	Kg/mt^3
7	Dynamic viscosity	0.43	Cp

Crude Oil

Table 3.2 Crude properties

	parameter	Value	Unit
1	mass flow rate	70000	Kg/hr
2	Inlet temperature(t_1)	40	°C
3	Outlet temperature(t_2)	78	°C
4	Specific heat	2.05	KJ/kg°C
5	Thermal conductivity	0.134	W/mt°C
6	Density	820	Kg/mt^3
7	viscosity	3.2	Cp

Allowable pressure drop on both side streams is 0.8 bar.

Appendix1 Solution as per done in excel:

	Parameter	value	unit
1	Heat Duty(øt)	5434000	KJ/hr
		1509.444444	KW
		1509444.444	W
	Terminal temperatures		
2	$\Delta T1$	122	^0C
3	$\Delta T2$	50	^0C
	Let the no. of passes(Np)	2	
	Allocating crude oil on tube side and kerosene on shell side		
4	R	2.894736842	
5	S	0.2375	
6	Ft(from fig.)	0.88	
7	ΔTln	80.71766621	^0C
8	ΔTm	71.03154627	
	Assuming Overall Heat Transfer coefficient(Uo)	300	$W/mt^{2o}C$
9	Heat transfer area(A)	70.83446364	
	Assuming length of each tube(L)	12	ft
		3.6576	mt
	Assuming diameter of each tube(d_0)=3/4 inches	0.01905	mt
10	No. of tubes(Nt)	323.5963008	
		324	
	Assuming tube pitch Pt=1.25d_0 (triangular pitch)		
	from table, for Np=2		
	k1	0.249	
	n1	2.207	
11	Bundle Diameter(Db)	490.9267674	mm
12	Shell ID	580.9267674	mm
	For 18 BWG tube Tube ID(di)	0.01574	mt
13	Tube side flow area(at)	0.031522043	mt^2
14	Tube side mass velocity(Gt)	616.8522984	kg/mt^2sec
15	Tube side velocity(ut)	0.752258901	mt/sec
	ut<1		

	Increasing no. of passes (Np)	4	
	From table, for Np=4		
	k1	0.175	
	n1	2.285	
16	Db	512.4354675	mm
17	Shell ID	602.4354675	mm
18	At	0.015741383	mt^2
19	Gt	1235.243692	kg/mt^2sec
20	Ut	1.692114647	mt/sec
	tube side velocity (ut>1)		
21	Reynolds no.(Re)	6075.854911	
22	Prandtl No.	48.95522388	
		49	
	By Dittus Boedter Eq(Re>2000)		
23	Nusselt No.(Nu)	88.35831702	
24	Oil side heat transfer coefficient(hi)	752.224554	
	To improve Oil side heat transfer coeff. increase length(L)	5	mt
25	Number Of tubes(Nt)	236.717166	
		236	
26	At	0.01148025	mt^2
27	Gt	1693.73004	kg/mt^2sec
28	Ut	2.065524439	mt/sec
29	Reynolds no.(Re)	8331.766808	
30	Prandtl No.(Pr)	48.95522388	
31	Nusselt No.(Nu)	113.7499474	
32	Oil side heat transfer coefficient(hi)	939.4023937	$W/mt^2{}^\circ C$
	Kerosene Side Heat transfer Coeff.(Shell Side)		
	Tube pitch(Pt)	$1.25d_o$	
	Shell inside diameter(Ds)	602.4355	mm
		0.602	mt
	For the first trial, Baffle spacing(Bs)	120.4871	mm
		0.12	mt

33	Shell Side flow area(As)	0.014448	mt^2
34	Shell Side Mass velocity, (Gs)	384.5207334	kg/mt^2sec
35	Shell Side velocity(us)	0.526740731	mt/sec
36	Shell side equivalent diameter for trainagular pitch(de)	0.013736003	mt
37	Reynolds no.(Re)	12283.20408	
38	Prandtl No.(Pr)	8.17	
	For Baffle cut=25% and Re=12232		
	Jh(from graph)	0.0052	
39	Nusselt No.(Nu)	128.6438503	
40	Kerosene Side Heat transfer Coeff.(hs)	1217.508554	$W/mt^{2}°C$
	From table fouling coeff. For tube side fluid(hid)	3000	$W/mt^{2}°C$
	From table fouling coeff. For shell side fluid(hod)	4000	$W/mt^{2}°C$
	Thermal conductivity of tube material	55	w/mt°C
41	$(1/U_0)$	0.002796198	
	Overall Heat ransfer coeff.(U_0)	357.6284599	$W/mt^{2}°C$
42	Heat Transfer area required(Ar)	59.4201566	mt^2
43	% Excess heat transfer area	19.20948663	%
	<u>Shell side pressure drop</u>		
	From fig. Jf	0.049	
44	ΔPs	72493.25886	Pa
	To decrease shell side pressure drop (Bs) should be increased	0.15	mt
45	Shell Side flow area(As)	0.01806	mt^2
46	Shell Side Mass velocity, (Gs)	307.6165867	kg/mt^2sec
47	Shell Side velocity(us)	0.421392585	mt/sec
48	Reynolds no.(Re)	9826.563265	
49	Prandtl No.(Pr)	8.17	
	For Re=9826, 25% baffle cut, Jh	0.006	
50	Nusselt No.(Nu)	117.9196506	
51	Kerosene Side Heat transfer Coeff.(hs)	1116.012797	$W/mt^{2}°C$
52	1/Uo	0.002870896	
	Overall Heat ransfer coeff.(U_0)	348.3233533	$W/mt^{2}°C$
53	Heat Transfer area required(Ar)	61.00750608	mt^2

44	% Excess heat transfer area	16.10778442	%
	From fig. Jf	0.07	
45	ΔPs	53023.64076	Pa
		53.02364076	Kpa
	Tube side pressure drop		
	From graph		Pa
46	Tube side pressure drop(ΔPt)	4966.074374	

RESULTS AND DISCUSSION:

Specification of design:

Heat duty required (øt) = 1509.44 KW

Mean temperature difference, ΔTm = 71.031

Heat transfer area(A) = 70.83446364 ft^2

Number of tubes(Nt) = 323.6

Number of passes= 4

Tube outer diameter, do = .011905 mt

Tube pitch, Pt = 1.25 do (triangular)

Tube side heat transfer coefficient, hi = 939.4023937 $W/mt^2 °C$

Shell Inner diameter = 602.4354675 mm

Shell side heat transfer coefficient = 1217.508554 $W/mt^2 °C$

Shell side pressure drop(ΔP_t) = 72493.26 Pa

Tube side pressure drop(ΔP_t) = 4966.07 Pa

The shell and tube heat exchanger is of class R as it has to serve severe duty of exchanging heat between crude and kerosene.

The tube layout pattern is triangular pitch, as the heat transfer and the no of tubes required here are very high and triangular pitch can accommodate more no of tubes in a given shell diameter than square pitch and can create more turbulence on the shell side fluid than square pitch.

CONCLUSION AND RECOMMENDATION:

Since the inlet temperature difference between the two fluids is very high (greater than 100° F) therefore floating head configuration should be used because it permits the differential thermal expansion or contraction between shell and tubes.

Floating head configuration should be used for the given operation as both the process fluid (crude and kerosene) will foul the heat exchanger and cleaning of floating head is much easier than other configurations.

The U-tube configuration cannot be used here because:

Unlike floating head in U-tube the tube material becomes weaker in bending portion with time and it is not recommended for severe conditions.

Since the fluid in the tube side is crude in this problem therefore U-tube configuration is not recommended because dirty fluid cannot be send to the tube side in U-tube heat exchanger.

In U-tube heat exchanger the numbers of tube side passes are fixed (two) which reduces the flexibility in design calculations.

Pull through floating head heat exchanger is preferred here over split-ring heat exchanger because maintenance of pull through is easier and economic than split-ring.

The maximum allowable pressure drop given here is very high so we can use triangular tube pitch layout as it is more economic than square pitch layout.

Nomenclature:

Φ=Heat Duty, W

K=Thermal conductivity of fluid, W/mt°C

A=Heat transfer surface area, mt^2

ΔT_m= Mean temperature difference, °C

U= Overall heat transfer coefficient, $W/mt^2\,°C$

h_o=Outside fluid film coefficient, $W/mt^2\,°C$

h_i=Inside fluid film coefficient, $W/mt^2\,°C$

h_{od} = Outside dirt coefficient(fouling factor), $W/mt^2\,°C$

h_{id}= Inside dirt coefficient, $W/mt^2\,°C$

k_w= Thermal conductivity of tube wall material, W/mt°C

d_i= Tube inside diameter, mt

d_o= Tube outside diameter, mt

ΔT_{lm}= Logarithmic mean temperature difference, °C

T_1= Inlet Shell side fluid temperature, °C

T_2= Outlet shell side fluid temperature, °C

t_1= Inlet tube-side temperature, °C

t_2= Outlet tube-side temperature, °C

F_t= Temperarure correction factor

C_p= Specific heat of fluid, KJ/kg°C

m=Mass flow rate, Kg/sec

ΔT=temperature difference carried out, °C

R,S= Temperature ratios

L= Tube length, mt

N_t= Total number of tubes

D_b= Tube bundle diameter, mm

λ= Latent heat of vaporization, KJ/kg

Nu= Nusselt number

Re= Reynolds number

Pr= Prandtl number

μ= Viscosity of fluid at the bulk fluid temperature, $N.s/mt^2$

μ_w= Viscosity of fluid at tube wall temperature, $N.s/mt^2$

G_t= Tube side mass velocity, Kg/mt^2sec

a_t= Tube side flow area, mt^2

u_t= Tube side velocity, mt/sec

ρ= Density of fluid, kg/mt^3

ΔP_t= Tube side friction factor, N/mt^2 or Pa

N_p= Tube side passes

J_f= Tube side friction factor

A_s= Shell side flow area, mt^2

P_t=Tube pitch, mt

B_s= Baffle spacing, mt

D_s= Shell inside diameter, mt

G_s= Shell side mass velocity, kg/mt^2 sec

ρ_s= Density of shell side fluid, kg/mt^3

d_e= Shell side equivalent diameter, mm

ΔP_s= Shell side pressure drop, N/mt^2

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