



**Sudan University of Science and Technology**



**College of Engineering**

**School of Mechanical Engineering**

**Power Department**

# **Design of Heat Exchanger used for Chemical Process**

**تصميم مبادل حراري يستخدم لإجراءات كيميائية**

**A research Submitted in Partial Fulfillment for the Requirements of  
the Degree of B.Sc (Honor) in Mechanical Engineering (Power  
Department)**

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## الايه

﴿اللَّهُ الَّذِي خَلَقَ السَّمَاوَاتِ وَالْأَرْضَ وَأَنْزَلَ مِنَ السَّمَاءِ مَاءً فَأَخْرَجَ بِهِ مِنَ الثَّمَرَاتِ

رِزْقًا لَكُمْ ۖ وَسَخَّرَ لَكُمُ الْفُلْكَ لِتَجْرِيَ فِي الْبَحْرِ بِأَمْرِهِ ۖ وَسَخَّرَ لَكُمُ الْأَنْهَارَ ۖ \* وَسَخَّرَ لَكُمُ

الشَّمْسَ وَالْقَمَرَ دَائِبَيْنِ ۖ وَسَخَّرَ لَكُمُ اللَّيْلَ وَالنَّهَارَ ۖ \* وَأَتَاكُمْ مِنْ كُلِّ مَا سَأَلْتُمُوهُ ۚ وَإِنْ تَعُدُّوا

نِعْمَتَ اللَّهِ لَا تَحْصُوهَا ۚ إِنَّ الْإِنْسَانَ لَظَلُومٌ كَفَّارٌ ۖ﴾

سورة ابراهيم الآيات ( 32-33-34 )

# *Dedication*

*We dedicate this humble research to our mothers, fathers, and brothers who have been very patient and determined to make us who we are now. To our lecturers in the school of mechanical engineering. And to whomsoever helped through our journey of education.*

# *Appreciation*

*Our gratitude to eng. Abdalla Mukhtar Mohammed Abdalla our supervisor on this research who played a major role on directing us to the right track , and he has been a great advisor through all period of preparation.*

*Special thanks to the administration of Khartoum Refinery Company, they were cooperative with us and they were very generous with their time and effort to complete this research*

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## **Abstract**

Heat exchangers are easily one of the most important and widely used pieces of process equipment found in industrial sites. Regardless of the particular industry in question, it will likely require some type of temperature regulation, and for that exchanger and likely come into play. Heat exchangers may be used for either heating or cooling, however, in the industrial sector, particularly with in plant refineries, they are overwhelmingly used for cooling.

There are many types of heat exchangers, each with their own advantages and drawbacks, yet tailored to best suit different purposes and industries. Heat exchangers have a very broad range of industrial applications. They are used as components in air conditioning and cooling systems or of heating systems. Many industrial purposes call for a certain degree of heat function; however, typically great care must be taken to keep these processes from getting too hot. Within industrial plants and factories heat exchangers are required to keep machinery, chemicals, water, and other substances within a safe operating temperature.

This research includes all the steps that have been implemented for the design of the heat exchanger (shell and tube) and program (MATLAB) to make the necessary calculations for heat exchanger.

## تجريد

معدات تستخدم في المنشآت الصناعية ومحطات توليد الطاقة الكهربائية وفي العديد من المنظومات الأخرى بهدف تبادل الحرارة بين مائع ساخن ومائع بارد مقابل فقد في الطاقة.

أن الغرض الأساسي من استعمال المبادلات الحرارية هو الأقتصاد في النفقات ، حيث أن تكاليف تسخين النفط الخام على سبيل المثال يحتاج الى الكثير من الوقود والطاقة ، في حين تجد في نفس الوحدة منتجات نفطية بحاجة الى تبريد قبل إرساله الى الخزانات لذا يمكن أداء الوظيفتين في مبادل حراري واحد أو وفي مجموعة من المبادلات الحرارية تستخدم مبدلات الحرارة بشكل واسع في التدفئة والتبريد ، وتكييف الهواء محطات الطاقة والمصانع الكيميائية والبتروكيميائية بالإضافة إلى مصافي النفط ومعامل معالجة الغاز.

هذا البحث يحتوي على المبادلات الحرارية وأنواعها وطرق تصميمها، وكذلك تصميم مبادل حراري لتبريد النفط وجميع المعادلات والحسابات اللازمة للتصميم ، واخير برنامج لاجراء الحسابات اللازمة لتصميم المبادل الحراري.

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4-	A picture of floating head and bundle tube.
5-	program to design of heat Exchanger
6-	Run of MATLAB program

# *Chapter One*

# Introduction

## 1.1 A heat exchanger:

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 particulates 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 fractionate distill 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 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 recuperate contrast exchangers in which there is intermittent heat exchange between the hot 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 exchanger automobile radiators condensers evaporator's air pre-heaters and cooling towers. If no phase change occurs in any 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. Combustion and chemical reaction 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 a recuperate 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.

## 1.2 The objectives:

- 1- To decide the size of pipes and material for heat exchanger.
- 2- To decrease the losses in the exchanger.
- 3- To develop computer program for design calculation.

## 1.3The steps of the research:

Activities	November	December	January	February	March	April	May	June	July	August	September
Introduction	■	■									
Previous researches		■	■	■							
Theory			■	■	■	■	■				
Field visits						■	■	■	■	■	
Calculations							■	■	■	■	■
Computer program										■	■
Conclusion											■

Tables (1.1) The steps of the research



#### **1.4 Problem statement:**

1. Improvement performance of heat exchanger
2. Increase the efficiency of heat exchanger.

#### **1.5 Research assumptions:**

The design of the heat exchanger (Shell and Tube) to cool the (NAFTA)  
.And to decrease the losses in the exchanger and chose material of the exchanger  
and program calculation.

## *Chapter Two*

# Theory and literature review

## 2.1 Classification of heat exchangers:

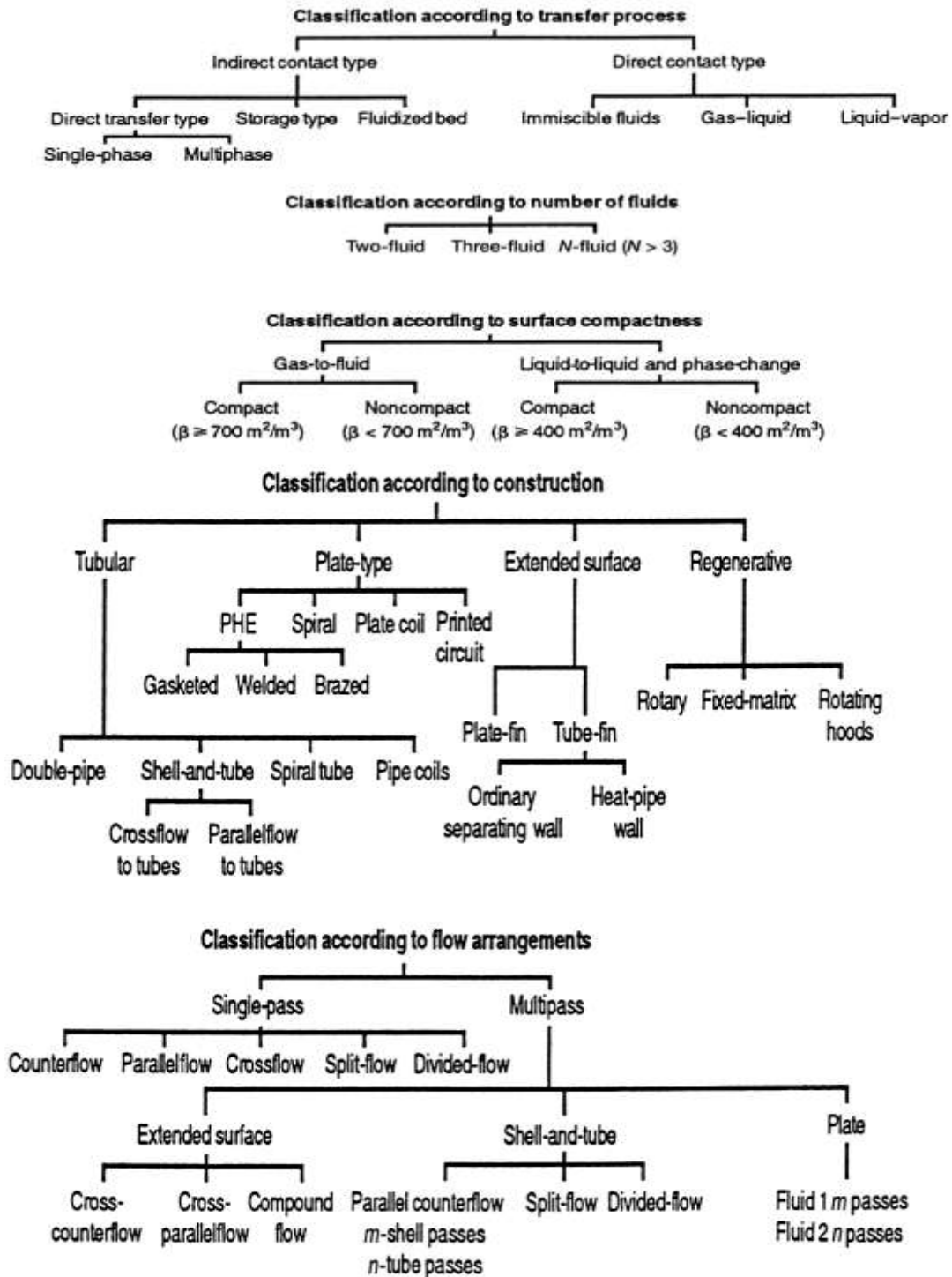


Figure (2.1): Classification of heat exchanger

### **2.1.1 Classification according to transfer processes:**

#### **(A) 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

#### **(B) Direct-Transfer Type Exchangers:**

In this type heat transfer continuously from the hot fluid to the cold fluid through a dividing wall. Although a simultaneous flow of two (or more) fluids is required in the exchanger, there is no direct mixing of the two (or more) fluids because each fluid flows in separate fluid passages. In general there are no moving parts in most such heat exchangers. This type of exchanger is designated as a recuperative heat exchanger or simply as a recuperator. Some examples of direct transfer type heat exchangers are tubular plate-type and extended surface exchangers. Note that the term recuperator is not commonly used in the process industry for shell-tube and plate heat exchangers although they are also considered as re-cuperators. Recuperators are further sub-classified as prime surface exchangers and extended-surface Exchangers. Prime surface exchangers do not employ fins or extended surfaces on any fluid side. Plain tubular exchangers, shell-and-tube exchangers with plain tubes and plate exchangers are good examples of prime surface exchangers. recuperators constitute a vast majority of all heat exchangers.

### **2.1.2 Classification 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 multi-fluid heat exchangers is algebraically very complex and is not covered in this book. Exclusively, only the design theory for two-fluid exchangers and some associated problems are presented in this book.

### **2.1.3 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.

#### **(A) Gas-to-Fluid Exchangers:**

The heat transfer coefficient ( $h$ ) for gases is generally one or two orders of magnitude lower than that for water, oil, and other liquids. Now, to minimize the size and weight of a gas to-liquid heat exchanger the thermal conductance, ( $\{hA\}$  products) on both sides of the exchanger should be approximately the same. Hence, the heat transfer surface on the gas side needs to have a much larger area and be more compact than can be realized practically with the circular tubes commonly used in shell-and-tube exchanger.

## (B) Liquid-to-Liquid and Phase-Change Exchangers:

Liquid-to-liquid and phase-change exchangers are gasketed plate and frame and welded plate, spiral plate, and printed-circuit exchangers.

### **2.1.4 Classification according to construction features:**

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 (Walker 1990). 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.

## (A) Tubular Heat Exchangers:

These exchangers are generally built of circular tubes, although elliptical, rectangular, or round/flat twisted tubes have also been used in some applications. There is considerable flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length, and arrangement. Tubular exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids.

Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications. They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and or pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchangers would work. These exchangers may be classified

## 1-shell and tube:

Shell and tube heat exchanger is built of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flow inside the tube. The other flow across and along the tubes. The major components of the shell and tube heat exchanger are tube bundle, shell, front end head, rear end head, baffles and tube sheets (Fig.1.2).

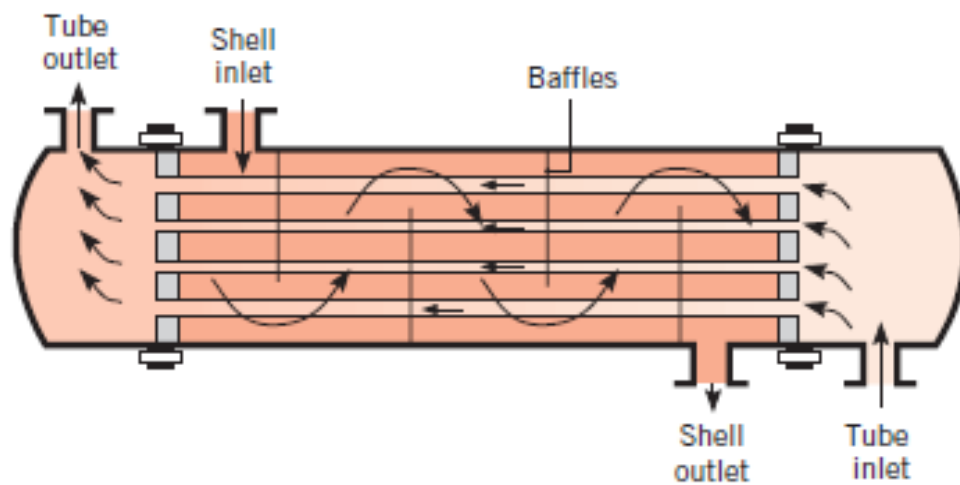


Figure (2.2): Shell and tube heat exchanger

The shell and tube heat exchanger is further divided into three categories as:

### a- Fixed-tube sheet:

A fixed-tube sheet heat exchanger (Figure 1.3) has straight tubes that are secured at both ends to tube sheets welded to the shell. The construction may have removable bonnet-type channel covers or integral tube sheets. The principal advantage of the fixed tube sheet construction is its low cost because of its simple construction. In fact the fixed tube sheet is the least expensive construction type, as long as no expansion joint is required.

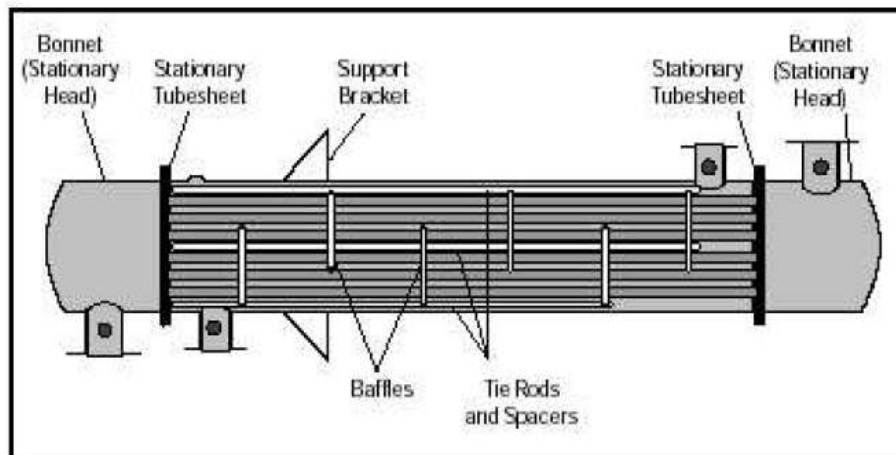


Figure (2.3): Fixed-tube sheet heat exchanger.

#### b- U tube:

As the name implies the tubes of a U-tube heat exchanger (Figure 1.4) are bent in the shape of a (U). There is only one tube sheet 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 fixed tube sheet exchanger.

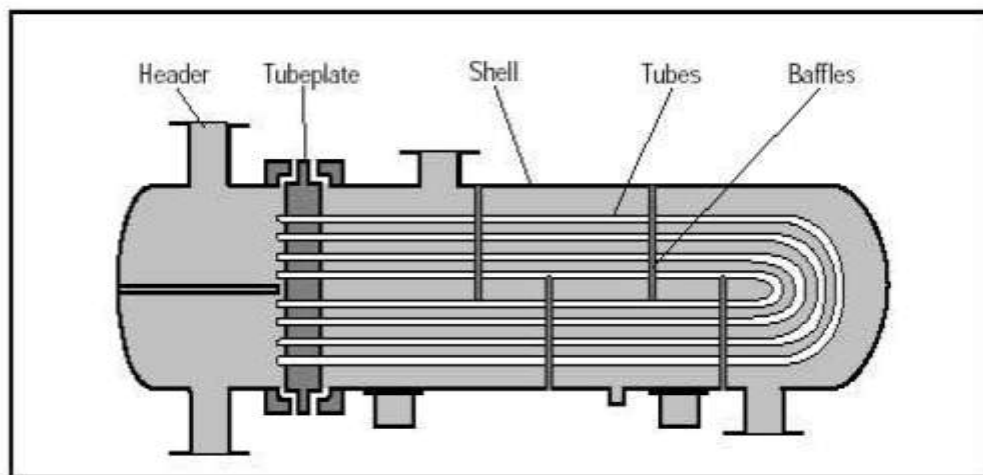


Figure (2.4): U-tube heat exchanger.



### c- Floating head:

The floating-head heat exchanger is the most versatile type of STHE 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 (SHTEs) 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.

### 2-double-pipe:

This is usually consists of concentric pipes One fluid flow in the inner pipe and the other fluid flow in the annulus between pipes. The two fluid may flow concurrent (parallel) or in counter current flow configuration.

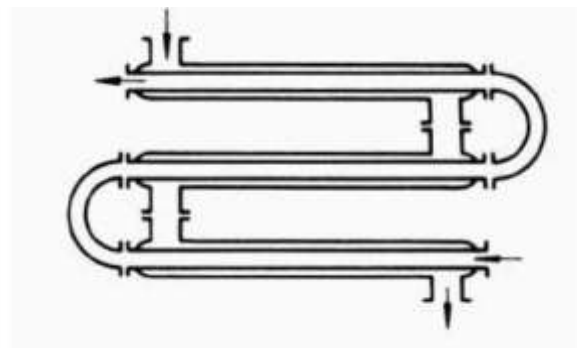


Figure (2.5): Double pipe heat exchanger.

### 3- A spiral tube exchangers:

Spiral tube heat exchanger consists of one or more spirally wound coils fitted in a shell(Fig. 1.6). Heat transfer associated with spiral tube is higher than that for a straight tube. In addition considerable amount of surface area can be accommodated in a given space by spiraling. Thermal expansion is no problem but cleaning is almost impossible.

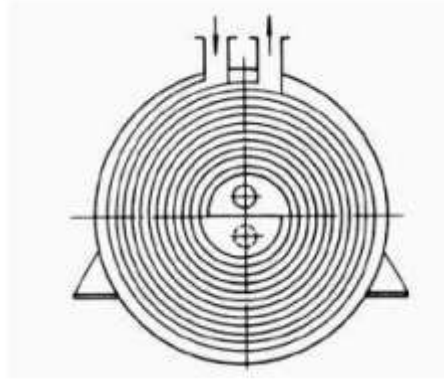


Figure (2.6): Spiral tube heat exchanger.

## (B) Plate-Type Heat Exchangers:

Plate-type heat exchangers are usually built of thin plates (all prime surface). The plates are either smooth or have some form of corrugation and they are either flat or wound in an exchangers. Generally these exchangers cannot accommodate very high pressures temperatures, or pressure and temperature differences. Plate heat exchangers (PHEs) can be classified as gasketed, welded (one or both fluid passages), or brazed, depending on the leak tightness required. Other plate-type exchangers are spiral plate, lamella, and plate coil exchangers.

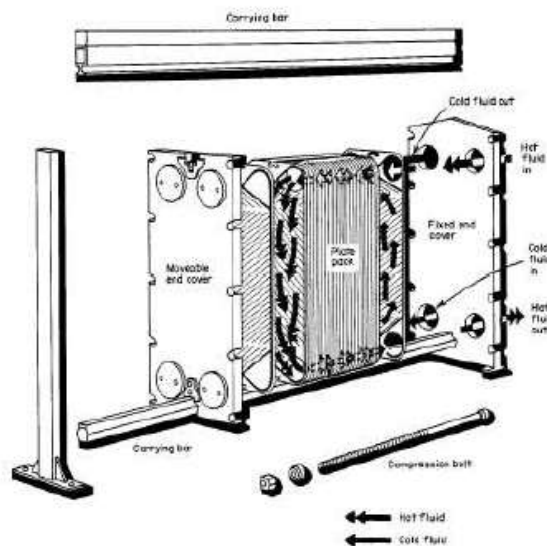


Figure (2.7): Plate-Type Heat Exchangers.

### (C) Extended Surface Heat Exchangers:

The tubular and plate-type exchangers described previously are all prime surface heat exchangers, except for a shell-and-tube exchanger with low finned tubing. Their heat exchanger effectiveness is usually 60% or below and the heat transfer surface area density is usually less than ( $700\text{m}^2/\text{m}^3$ ) in some applications, much higher (up to about 98%) exchanger effectiveness is essential. In some applications, much higher (up to about 98%) exchanger effectiveness is essential and the box volume and mass are limited so that a much more compact surface is mandated. Also in a heat exchanger with gases or some liquids the heat transfer coefficient is quite low on one or both fluid sides. This results in a large heat transfer surface area requirement. One of the most common methods to increase the surface area and exchanger

### (D) Regenerators:

The regenerator is a storage-type heat exchanger as described earlier. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. To have continuous operation.

## **2.1.5 Classification according to flow arrangements:**

### (1) Single-Pass Exchangers:

- Counter flow Exchanger.
- Parallel flow Exchanger.
- Cross flow Exchanger.

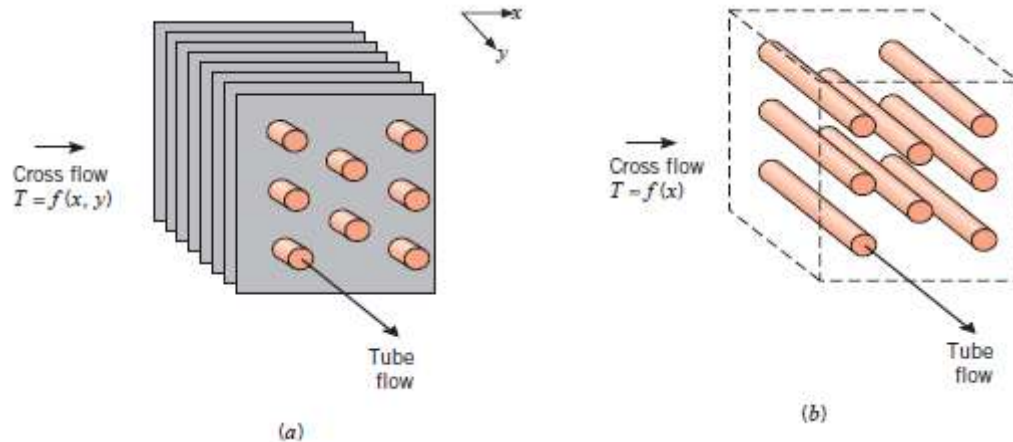


Figure (2.8): Cross-flow heat exchangers. (a) Finned with both fluids unmixed. (b) Un-finned

with one fluid mixed and the other unmixed:

- Split-Flow Exchanger.
- Divided-Flow Exchanger.

## (2) Multi-pass Exchangers:

When the design of a heat exchanger results in either an extreme length, significantly low fluid velocities, or a low effectiveness a multi pass heat exchanger or several single-pass exchangers in series, or a combination of both, is employed:

- 1- Multi-pass cross flow exchangers.
- 2- Multi-pass Shell and Tube exchangers.
- 3 - Multi-pass Plate Exchanger.

### 2.1.6 Classification according to heat transfer mechanisms:

The basic heat transfer mechanisms employed for transfer of thermal energy from the fluid on one side of the exchanger to the wall are single-phase convection 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.

## **2.2 Fouling in Heat Exchangers:**

Fouling refers to undesired accumulation of solid material (by-products of the heat transfer processes) on heat exchanger surfaces which results in additional thermal resistance to heat transfer, thus reducing exchanger performance. The fouling layer also blocks the flow passage/area and increases surface roughness, thus either reducing the flow rate in the exchanger or increasing the pressure drop or both. The foulant deposits may be loose such as magnetite particles or hard and tenacious such as calcium carbonate scale; other deposits may be sediment, polymers, coking or corrosion products, inorganic salts, biological growth, etc. Depending upon the fluids, operating conditions, and heat exchanger construction, the maximum fouling layer thickness on the heat transfer surface may result in a few hours to a number of years. Fouling could be very costly depending upon the nature of fouling and the applications. It increases capital costs:

- (1) Oversizing heat exchanger.
- (2) Provisions for cleaning.
- (3) Use of special materials and constructions/surface features. It increases maintenance costs:

- Cleaning techniques.
- Chemical additives.
- Troubleshooting.

## **2.3 Applications of the shell and tube heat exchanger:**

Shell and tube heat exchangers represent the most widely used vehicle for heat transfer in process applications. They frequently are selected for duties such as:

- Process liquid or gas cooling.
- Process or refrigerant vapor or steam condensing.
- Process liquid, steam or refrigerant evaporation.
- Process heat removal and preheating of feed water.
- Thermal energy conservation efforts and heat recovery.

## **2.4 Factors affecting the performance of heat exchanger:**

Since the size and resulting cost of a heat exchanger depend greatly on the log-mean-temperature difference (LMTD) a process designer should consider the effect of the operating temperature levels in the early stages of process design.

A high (LMTD) generally results in a smaller heat exchanger. There fore when considering operating temperature levels a larger can be achieved by increasing the temperature level of the cooling medium. Close temperature approaches, where small differentials exist between the inlet of one fluid stream and the outlet of the other will result in very low (LMTD).

There are no specific rules for determining the optimum operating temperatures.

These should be selected based on the service and utility of the heat exchanger. Inefficient design and poor heat-exchanger performance can result when the (LMTD) is too high or too low. For a good design that is to cover many services the lesser temperature difference between the shell-side and the tube-side fluids

should be greater than (10 °F). The greater temperature difference should exceed (40°F).

Flow quantities fluid flow rates (lb/h) on both the shell and tube sides can affect the size and design of an exchangers. A designer may be forced to resort to multiple shells in series when the (LMTD) and flow quantities are low, and when a large temperature difference exists between the shell-side and the tube-side fluids under these conditions, a counter current flow pattern must be maintained.

Under low-flow conditions, the designer may resort to multiple shells in series, to achieve reasonable fluid velocities and heat-transfer rates- When flow rates are extremely high for the surface requirement, multiple shells in parallel may be needed to achieve reasonable velocities, pressure drops, and an efficient heat-exchanger design.

## **2.5 Fouling factors:**

Dirt scale or other deposits formed on the tube inside and/or outside-which results in resistance to the flow of heated is called (fouling). The size and cost of a heat exchanger are related to specified fouling resistance; haphazard guessing of fouling can lie costly. Inasmuch as fouling factors are difficult to determine, they should be based on experience. Fouling varies and depends on the material of construction of the tubes, the types of fluids involved, temperatures, velocities and oilier operating conditions. Thus, the selection of fouling factors is arbitrary. Many complaints in heat exchanger operation that cannot be traced to dims in thermal design are generally traced to fouling. If heavy fouling is anticipated for a particular service, the user should make provisions for periodic chemical or mechanical cleaning of the exchanger. If heavy tube-side fouling is foreseen, a straight-tube heat exchanger will) with larger diameter tubes should be specified. But when heavy shell-side fouling is anticipated, the purchaser should specify a

removable-bundle design, with tubes on a square pitch for mechanical cleaning of the bundle.

## **2.6 Allowable pressure drop:**

Selecting the optimum allow-able pressure drop involves consideration of the overall process. However, high pressure drops may result in a smaller size (less costly) heat exchanger for other than isothermal-service requirements. The savings in the initial cost of a heat exchanger must be evaluated against a possible increase in operating costs. For reasonable designs, the allowable pressure drops should be 5 psi or higher for operating pressures in excess of 10 psig. In some instances it is not practical to use all of the available pressure drop because the resulting high fluid velocities could cause erosion or vibration damage to heat exchanger components.

## **2.7 Design requirements:**

Selecting a heat-exchanger design must not be done casually; a considerable amount of time may easily be spent in heat-transfer and mechanical design calculations .A purchaser should establish his requirements for his heat exchanger at time he asks for bids. The practice of imposing on a vendor the design and pricing of alternative selections is wasteful and in many cases costly .To provide an optimum heat-exchanger (it-sign), the manufacturer or designer should be furnished by the user with the following minimum service information:

- Total heat load.
- Fluid quantities entering and leaving the exchanger .
- Specific heat thermal conductivity, viscosity, molecular weight or specific gravity of the fluids.
- Heat-exchanger ingoing and outgoing temperature.



- Operating pressures.
- Allowable pressure drops.
- Fouling factors.
- Design pressure and temperature.
- Heat-exchanger type.
- Materials of construction.
- Tube-wall thickness for corrosion considerations.
- corrosion allowance.
- Specifications, codes and standards.
- Size or space limitations.
- Horizontal or vertical installation.

## **2.8 Design of primary exchanger components:**

A shell-and-tube heat exchanger is essentially a bundle of tubes enclosed in a shell and so arranged that one fluid flows through the tubes and another fluid flows across the outside of the tubes, heat being transferred from one fluid to the other through the tube wall. A number of other mechanical components are required to guide the fluids into, through, and out of the exchanger, to prevent the fluids from mixing, and to ensure the mechanical integrity of the heat exchanger. In the design process, it is important to consider the mechanical integrity under varying operational conditions and the maintainability (especially cleaning) of the exchanger as equally important with the thermal-hydraulic design.

## **2.9 Tubes:**

Tubes used in exchangers range from 6.35 mm ( 1/4 in.) to 50.8 mm (2 in.) and above in outside diameter with the wall thickness usually being specified by the Birmingham wire gauge (BWG). Tubes are generally available in any desired length up to 30 m (100 ft) or more for plain tubes. While plain tubes are widely

used, a variety of internally and/or externally enhanced tubes is available to provide special heat transfer characteristics when economically justified (see subsection on enhancement in Section 4.15). Low fin tubes having circumferential fins typically 0.8 to 1.6 mm (0.032 to 0.062 in.) high, spaced 630 to 1260 fins/m (16 to 32 fins/in.) are often employed, especially when the shell-side heat transfer coefficient is substantially smaller than the tube-side coefficient. The outside heat transfer area of a low fin tube is three to six times the inside area, resulting in a smaller heat exchanger shell for the same service, which may offset the higher cost of the tube per unit length.

## **2.10 Tube sheets:**

The tubes are inserted into slightly oversized holes drilled (or, occasionally, punched) through the tube sheets. The tubes are secured by several means, depending upon the mechanical severity of the application and the need to avoid leakage between the streams. In some low-severity applications, the tubes are roller-expanded into smooth holes in the tube sheet. For a stronger joint, two shallow circumferential grooves are cut into the wall of the hole in the tube sheet and the tube roller-expanded into the grooves; to eliminate the possibility of leakage, a seal weld can be run between the outer end of the tube and the tube sheet. Alternatively, the tubes may be strength-welded into the tube sheet.

## **2.11 Tube Supports:**

It is essential to provide periodic support along the length of the tubes to prevent sagging and destructive vibration caused by the fluid flowing across the tube bank. A secondary role played by the tube supports is to guide the flow back and forth across the tube bank, increasing the velocity and improving the heat transfer on the shell side (but also increasing the pressure drop). The tube support

is usually in the form of single segmental baffles circular plates with holes drilled to accommodate the tubes and with a segment sheared off to form a (window) or (turnaround) to allow the shell-side fluid to pass from one cross-flow section to the next. The baffles must overlap at least one full row of tubes to give the bundle the necessary rigidity against vibration. When minimizing shell-side pressure drop is not a priority, a baffle cut of 15 to 25% of the shell inside diameter is customary. Baffle spacing is determined first by the necessity to avoid vibration and secondarily to approximately match the free cross-flow area between adjacent baffles to the flow area in the window; i.e., small baffle cuts correspond to closer baffle spacing. In situations such as low-pressure gas flows on the shell side where pressure drop is severely limited, double segmental and strip baffle arrays can be used. More recently, a helical baffle arrangement has been introduced which causes the shell-side fluid to spiral through the exchanger giving improved heat transfer vs. pressure drop characteristics. Where vibration prevention and/or minimum pressure drop are the main concerns, grids of rods or strips can be used.

## **2.12 Code and standards:**

The objective of codes and standards are best described by (ASME). The objectives of code rules and standards (apart from fixing dimensional values) is to achieve minimum requirements for safe construction in other words, to provide public protection by defining those materials, design, fabrication and inspection requirements. whose omission may radically increase operating hazards. Experience with code rules has demonstrated that the probability of disastrous failure can be reduced to the extremely low level necessary to protect life and property by suitable minimum requirements and safety factors. Obviously, it is impossible for general rules to anticipate other than conventional service. Suitable precautions are therefore entirely the responsibility of the design engineer guided

by the needs and specifications of the user. Over years a number of standardization bodies have been developed by individual country, manufacturers and designers to lay down nomenclatures for the size and type of shell and tube heat exchangers. These include among other:

- ❖ (TEMA) standards (Tubular Exchanger Manufacturer Association 1998).
- ❖ (HEI) standards (Heat Exchanger Institute 1980).
- ❖ (API) (American Petroleum Institute).

### **2.12.1 TEMA Designations:**

In order to understand the design and operation of the shell and tube heat exchanger, it is important to know the nomenclature and terminology used to describe them and the various parts that go to their construction. Only then we can understand the design and reports given by the researchers, designers, manufacturer and users. It is essential for the designer to have a good working knowledge of the mechanical features of STHEs and how they influence thermal design. The principal components of an STHE are:

- Shell.
- Shell cover.
- Tubes.
- Channel.
- Channel cover.
- Tube sheet.
- Baffles.
- Nozzles.

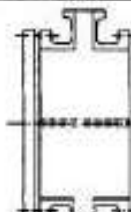
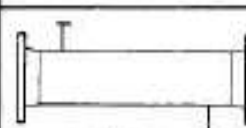
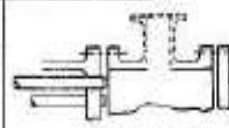
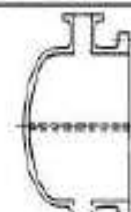
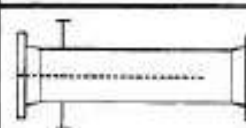
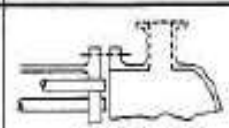
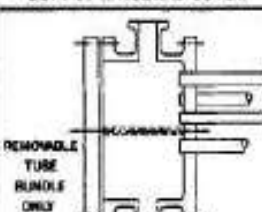
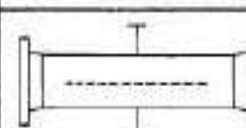
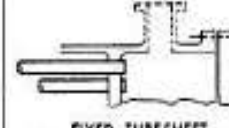
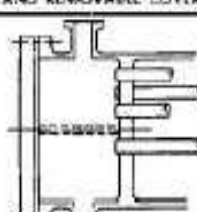
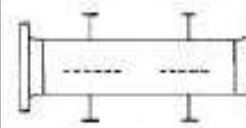
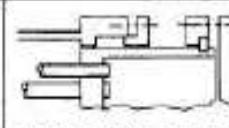
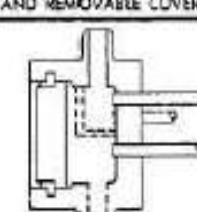
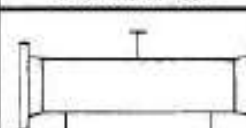
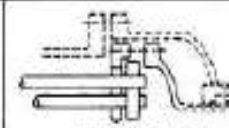
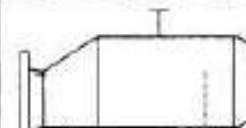
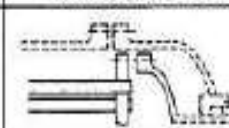
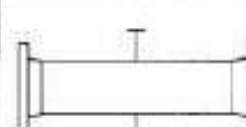
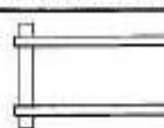
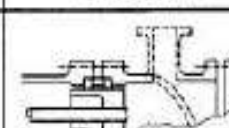
FRONT END STATIONARY HEAD TYPES		SHELL TYPES		REAR END HEAD TYPES	
A		E		L	
	CHANNEL AND REMOVABLE COVER				
B		F		M	
	BONNET (INTEGRAL COVER)				
C		G		N	
	REMOVABLE TUBE BUNDLE ONLY CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER				
N		H		P	
	CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER				
D		J		S	
	SPECIAL HIGH PRESSURE CLOSURE				
		K		T	
		X		U	
				W	

Figure (2.9): TEMA-type designations for shell and tube heat exchangers.

## **2.13 Study of a district heating system with the ring network technology and plate heat exchangers in a consumer substation:**

Plate heat exchangers (PHE) have consolidated their position key components of modern heating processes. They are widely accepted as the most suitable design for heat transfer application various processes, including the field of energy-efficient district heating. Their frequent occurrence originates from their superb capability to produce remarkably high heat transfer coefficients with minimal fouling factors and physical size.

This study consists of a new district heating network coupling and mass flow control combined with a consumer substation. The concept includes such features as a new low temperature program for the supply and return temperature curves which yields higher temperature cooling and diminishes the DH water flows and pressure losses. Here the mass flow control is real is end on both the primary and secondary sides as a function of time and the outdoor temperature .Time-dependent numeric models were constructed for (DH) network building structures, and heating heat exchangers at a sub-station. The simulation was executed with a constant (DH) water supply temperature for the day being studied. These (DH) supply temperatures were selected to be close to the values meant for the corresponding outdoor mean temperatures. The calculation procedure of the whole system was sketched. It was found out that a steady state in the structures of the buildings or in the (DH) network piping may not be reached during calculation cycles and the results are transient solutions by nature .The plate heat exchanger (PHE) and its operation in a substation were studied more closely. For that purpose a time-dependent model was constructed in the MAT-LAB and SIMULINK environments. The input values for the simulation were obtained from the (DH) network and the building models. A corrugated plate heat exchanger model for

heating was constructed. Heat exchangers were dimensioned to have the flow rates the (DH) water almost equal on the primary and secondary sides. The heat exchanger was divided into five vertical parts and 10 elements for which thermal balances were written between the hot and cold sides. The variations in the mass flow rates, pressure losses of the primary and secondary sides, and local overall heat transfer coefficient in the vertical parts of the exchanger were presented for the selected autumn and/or winter days. It was detected that the pressure losses with mass flow control are only a fraction of their traditional values. Selecting approximately equal heat capacity flow rates between the hot and cold side fluids of the (PHE) gives a small temperature difference between the fluids and maximum temperature cooling at the substation and a lower return temperature towards the heating plant. The key performance factors of the plate heat exchangers (NTU) and effectiveness ( $\epsilon$ ) were monitored and the mean values that were obtained for the winter day that was studied were 9.2 and 0.9, respectively.

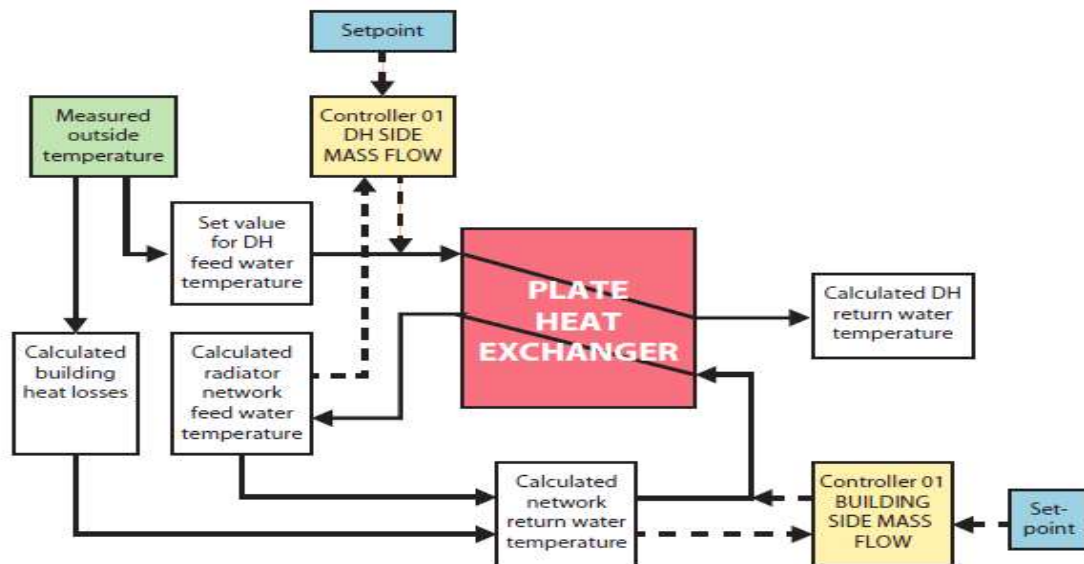


Figure (2.10): Structure of operating principle of the heat exchanger model

## **2.14 Heat exchanger design in combined cycle engines:**

Detailed cycle analysis at Reaction Engines Ltd has shown the need for three categories of heat exchangers two metallic and one ceramic. Their characteristics are determined by the pressures and temperatures at which the fluids operate across the heat transfer surface. The pre-cooler and the regenerator designs are dominated by the need to minimize entropy rise in the cycle and differ in the pressure difference between the two fluids resulting in tubular and micro-channel heat exchangers accordingly. The design of (HX3) is limited only by materials technology in that it operates at very high temperatures in order to minimize the surface area and mass. Reaction Engines (LTD) have the technology to build the pre-cooler, achieve the required wall thickness draw reliability, high braze efficiency, and can reclaim joints that crack. Preliminary tests with low temperature materials. Aluminum for the heat rejection heat exchangers have been very successful. We are looking at tooling manufacturing to produce channels in harder materials.

Overall the engine relies on high but achievable component efficiencies. Heat exchanger technology must be pushed to the limits of manufacturing capability. This design of engine appears to be the only one, provided the engine is optimized in conjunction with the vehicle, which could lead to a credible (SSTO) capability including polar orbits.

## **2.15 Design and Development of Shell and Tube Heat exchanger for Beverage:**

The design of (STHE) involves a large number of geometric and operating variables as a part of the search for heat exchanger geometry that meets the heat duty requirement and a given set of design constraints. A (STHE) is the most common type of heat exchanger 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 vessel) with a bundle of tubes inside it. One fluid runs through the tubes and the second runs over the tubes (through the shell) to transfer heat between the two fluids. This section explains the details of existing industrial scenario of design of (STHE).

## 1-Thermal Design:

The thermal design of (STHE) includes:

- A- Consideration of process fluids in both shell and tube side.
- B- Selection of required temperature specifications.
- C- Limiting the shell and tube side pressure drop.
- D- Setting shell and tube side velocity limits.
- E- Finding heat transfer area including fouling factor.

## 2- Mechanical Design:

- A- Selection of (TEMA) layout-based on thermal design.
- B- Selection of tube parameters such as size, thickness layout pitch material.
- C- Limiting the upper and lower design on tube length.
- D- Selection of shell side parameters such as material baffles spacing, and clearances.
- E- Thermal conductivity of tube material.
- F- Setting upper and lower design limits on shell diameter and baffle Spacing.

The problem presented in this paper is to design & develop a (STHE) conforming to the (TEMA/ASME) standards, based on following Input Data:

- 1) Inlet & Outlet Temperatures of fluids on Shell & Tube Side.
- 2) Tube length = 10,000 mm.
- 3) Tube OD = 38.1 mm.
- 4) Shell OD = 1350 mm.

As per the requirement the objective of the preset work is to perform thermal and mechanical design of (STHE) using (TEMA/ASME) standards to reduce time.

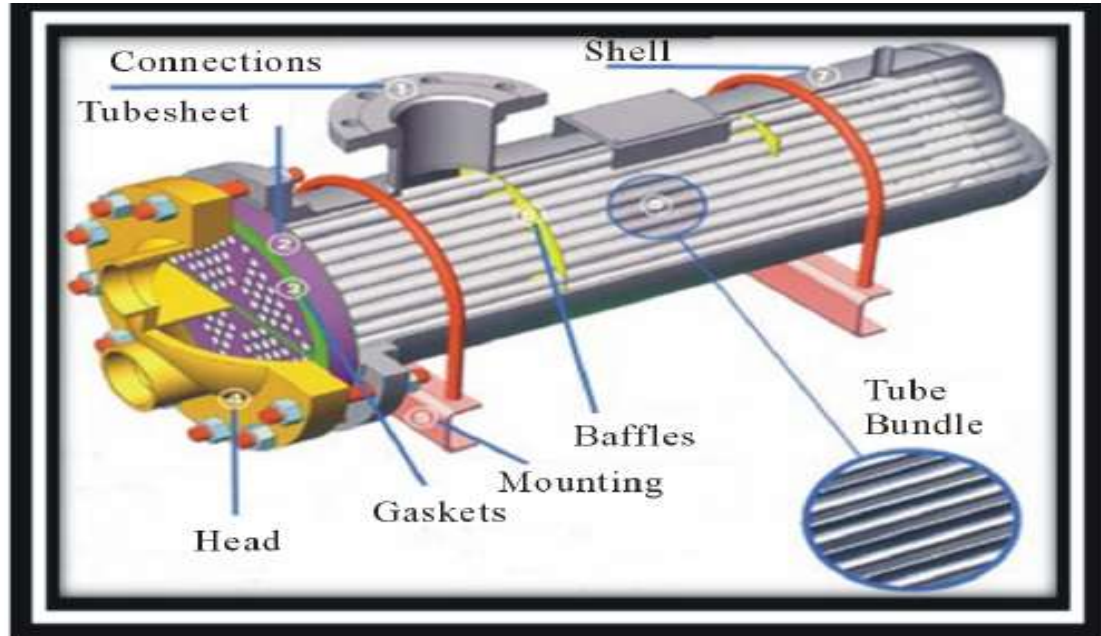


Figure (2.11): Major components of a typical STHE

### 3-Conclusion:

The design of (STHE) thermal and mechanical design was carried out using (TEMA/ASME) standards both manually and using software.

# *Chapter Three*

# **Methodology of research**

## **3.1 Heat exchanger design methodology:**

Design is an activity aimed at providing complete descriptions of an engineering system, part of a system, or just of a single system component. These descriptions represent an unambiguous specification of the system/component structure, size, and performance, as well as other characteristics important for subsequent manufacturing and utilization. This can be accomplished using a well defined design methodology.

From the formulation of the scope of this activity, it must be clear that the design methodology has a very complex structure. Moreover, a design methodology for a heat exchanger as a component must be consistent with the life-cycle design of a system. Lifecycle design assumes considerations organized in the following stages.

## **3.2 Problem formulation (including interaction with a consumer):**

Concept development (selection of workable designs preliminary design).  
Detailed exchanger design (design calculations and other pertinent considerations).  
Manufacturing. utilization considerations (operation, phase out disposal).

## **3.3 Major design considerations include:**

- (1) Process and design specifications.
- (2) Thermal and hydraulic design.
- (3) Mechanical design.
- (4) Manufacturing considerations and cost.
- (5) Trade-off factors and system-based optimization.

### **3.3.1 Process and Design Specifications:**

Process or design specifications include all necessary information to design and optimize exchangers for a specific application. It includes problem specifications for operating conditions, exchanger type, flow arrangement, materials and design (manufacturing) operation consideration. In addition, the heat exchanger design engineer provides necessary and missing information on the minimum input specifications required.

#### **3. 3.1.1 Problem Specifications:**

The first and most important consideration is to select the design basis. The design basis would require the specification of operating conditions and the environment in which the heat exchanger is going to be operated. These include fluid mass flow rates (including fluid types and their thermo physical properties) inlet temperatures and pressures of both fluid streams, required heat duty and maximum allowed pressure drops on both fluid sides, fluctuations in inlet temperatures and pressures due to variations in the process or environment parameters, corrosiveness and fouling characteristics of the fluids, and the operating environment (from safety, corrosion/erosion, temperature level, and environmental impact points of view). In addition, information may be provided on overall size, weight, and other design constraints, including cost, materials to be used, and alternative heat exchanger types and flow arrangements.

#### **3.3.1.2 Exchanger Specifications:**

Based on the problem specifications and the design engineer's experience, the exchanger construction type and flow arrangement are first selected. Selection of the construction type depends on the fluids (gas, liquid, or condensing/evaporating) used on each side of a two-fluid exchanger, operating pressures, temperatures, fouling and clean ability fluids and material compatibility,

corrosiveness of the fluids, how much leakage is permissible from one fluid to the other fluid, available heat exchanger manufacturing technology, and cost.

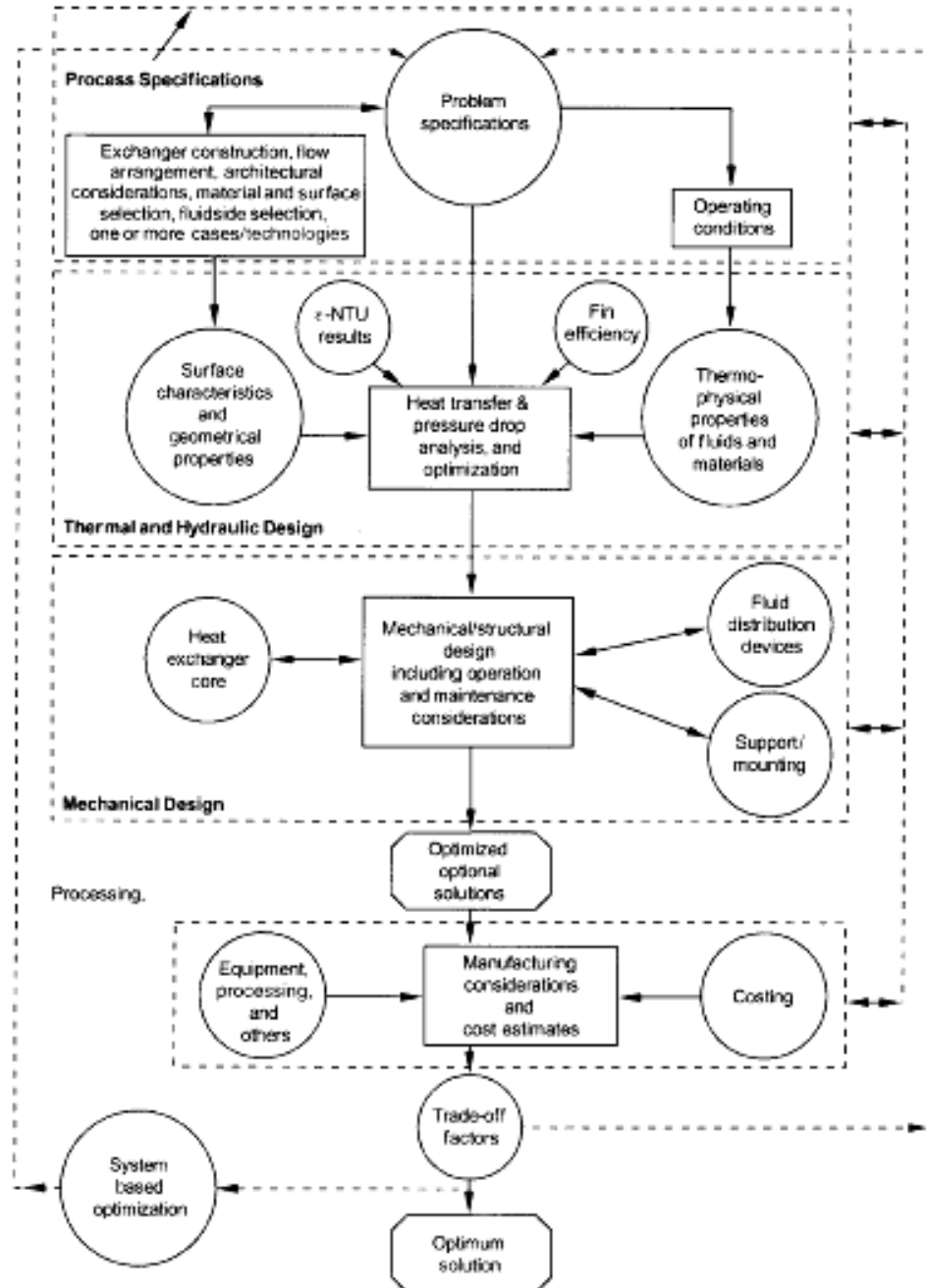


Figure (3.1): Heat exchanger design methodology.

### 3.3.2 Thermal and hydraulic design:

Only two important relationships constitute the entire thermal design procedure. These are:

1- Enthalpy rate equation

$$q = q_j = m_j \Delta h_j \quad (3.1)$$

$q \equiv$  Heat transfer rate

$m_j \equiv$  Mass flow rate

One for each of the two fluids ( $j = 1, 2$ )

2- Heat transfer rate equation or simply the rate equation

$$q = UA\Delta T_m \quad (3-2)$$

$U \equiv$  The overall heat transfer coefficient

$\Delta T_m \equiv$  The true mean temperature difference

$A \equiv$  The heat transfer area

#### 3.3.2.1 Heat Exchanger Thermal Design Problems:

a- Rating Problem:

Determination of heat transfer and pressure drop performance of either an existing exchanger or an already sized exchanger (to check vendor's design) is referred to as a rating problem. Inputs to the rating problem are the heat exchanger construction, flow arrangement and overall dimensions, complete details on the materials and surface geometries on both sides, including their non dimensional heat transfer and pressure drop characteristics ( $j$  or  $Nu$  and  $f$  vs.  $Re$ ), { fluid flow rates, inlet temperatures, and fouling factors. The fluid outlet temperatures, total

heat transfer rate, and pressure drops on each side of the exchanger are then determined in the rating problem. The rating problem is also sometimes referred to as the performance or simulation problem.

#### **b- Sizing Problem:**

In a broad sense, the design of a new heat exchanger means the determination/selection of an exchanger construction type, flow arrangement, tube/plate and fin material, and the physical size of an exchanger to meet the specified heat transfer and pressure drops within all specified constraints.

#### **3.3.2.2 Basic Thermal and Hydraulic Design Method:**

Based on the number of variables associated with the analysis of a heat exchanger, dependent and independent dimensionless groups are formulated. The relationships between dimensionless groups are subsequently determined for different flow arrangements. Depending on the choice of dimensionless groups, several design methods are being used by industry. These methods include  $\epsilon$ -NTU, P-NTU, MTD correction factor, and other methods.

#### **3.3.2.3 Surface Basic Characteristics:**

Surface basic characteristics on each fluid side are presented as  $j$  or  $Nu$  and  $f$  vs.  $Re$  curves in dimensionless form and as the heat transfer coefficient  $h$  and pressure drop  $\Delta p$  vs. the fluid mass flow rate  $\dot{m}$  or fluid mass velocity  $G$  in dimensional form. Accurate and reliable surface basic characteristics are a key input for exchanger thermal and hydraulic design.

#### **3.3.2.4 Surface Geometrical Properties:**

For heat transfer and pressure drop analyses, at least the following heat transfer surface geometrical properties are needed on each side of a two-fluid exchanger: minimum free-flow area  $A_o$ ; core frontal area  $A_{fr}$ ; heat transfer surface area  $A$  which includes both primary and fin area, if any; hydraulic diameter  $D_h$ ;



and flow length  $L$ . These quantities are computed from the basic dimensions of the core and heat transfer surface. On the shell side of a shell-and-tube heat exchanger, various leakage and bypass flow areas are also needed.

### **3.3.2.5 Thermo physical Properties:**

For thermal and hydraulic design, the following thermophysical properties are needed for the fluids: dynamic viscosity  $\mu$ , density  $\rho$ , specific heat  $c_p$ , and thermal conductivity  $k$ . For the wall, material thermal conductivity and specific heat may be needed.

### **3.3.2.6 Thermal and Hydraulic Design Problem Solution:**

Solution procedures for rating and sizing problems are of an analytical or numerical nature, with empirical data for heat transfer and flow friction characteristics and other pertinent characteristics. Due to the complexity of the calculations, these procedures are often executed using commercial or proprietary computer codes. Since there are many geometrical and operating condition-related variables and parameters associated with the sizing problem, the question is how to formulate the best possible design solution (selection of the values of these variables and parameters) among all feasible solutions that meet the performance and design criteria. This is achieved by employing mathematical optimization techniques after initial sizing to optimize the heat exchanger design objective function within the framework of imposed implicit and explicit constraints.

### **3.3.3 Mechanical Design:**

Mechanical design is essential to ensure the mechanical integrity of the exchanger under steady-state, transient, startup, shutdown, upset, and part-load operating conditions during its design life. The heat exchanger core is designed for the desired structural strength based on the operating pressures, temperatures, and corrosiveness or chemical reaction of fluids with materials. Pressure/thermal stress

calculations are performed to determine the thicknesses of critical parts in the exchangers, such as the fin, plate, tube, shell, and tubesheet. A proper selection of the material and the method of bonding (such as brazing, soldering, welding, or tension winding) fins to plates or tubes is made depending on the operating temperatures, pressures, types of fluids used, fouling and corrosion potential, design life, and so on.

### **3.3.3.1 Manufacturing Considerations and Cost Estimates:**

#### **a- Manufacturing Considerations:**

Manufacturing considerations may be subdivided into manufacturing equipment considerations, processing considerations, and other qualitative criteria. The equipment considerations that may influence which design should be selected include existing tooling versus new tooling; availability and limitations of dies, tools, machines, furnaces, and manufacturing space; production versus offline setup; and funding for capital investment. Processing considerations are related to how individual parts and components of a heat exchanger are manufactured and eventually assembled. This includes manufacturing of individual parts within specified tolerances; flow of parts; stacking of a heat exchanger core and eventual brazing, soldering, welding, or mechanical expansion of tubes or heat transfer surfaces; leak-free mounting (joining) of headers, tanks, manifolds, or return hairpins on the heat exchanger core; mounting of pipes; washing/cleaning of the exchanger; leak testing, mounting of the exchanger in the system; and structural supports

#### **b - Costing:**

The overall total cost, also called lifetime costs, associated with a heat Exchanger may be categorized as the capital, installation, operating, and sometimes also disposal costs. The capital (total installed) cost includes the costs associated with design, materials, manufacturing (machinery, labor, and overhead),

testing, shipping, installation, and depreciation. Installation of the exchanger on the site can be as high as the capital cost for some shell-and-tube and plate heat exchangers. The operating cost Consists of the costs associated with fluid pumping power, warranty, insurance, main- tenancies, repair, cleaning, lost production/downtime due to failure, energy cost associated with the utility (steam, fuel, water) in conjunction with the exchanger in the network, and decommissioning costs. Some of the cost estimates are difficult to obtain and best estimates are made at the design stage.

# *Chapter Four*

# Calculation

## 4.1 Formal analogy between thermal and electrical entities:

Heat Transfer Rate)

Electric Current flow

$$q = \frac{\Delta T}{(UA)^{-1}} = \frac{\Delta T}{R} \quad \Delta T = \frac{\Delta E}{R} \quad \Delta E = Ri \quad (4.1)$$

$\Delta T \equiv$  Temperature difference

$\Delta E \equiv$  Electromotive force (potential) difference

Parameter		Electrical		Thermal
Analogies				
Current	$i$	ampere, A	$q$	W, Btu/hr
Potential	$E$	volts, V	$\Delta T$	$^{\circ}\text{C}(\text{K}), ^{\circ}\text{F}(^{\circ}\text{R})$
Resistance	$R$	ohms, $\Omega$ , V/A	$R = 1/UA$	$^{\circ}\text{C}/\text{W}, ^{\circ}\text{F}\cdot\text{hr}/\text{Btu}$
Conductance	$G$	siemens, S, A/V	$UA$	$\text{W}/^{\circ}\text{C}, \text{Btu}/\text{hr}\cdot^{\circ}\text{F}$
Capacitance	$C$	farads, F, A s/V	$\bar{C}$	$\text{W}\cdot\text{s}/^{\circ}\text{C}, \text{Btu}/^{\circ}\text{F}$
Time constant	$RC$	s	$R\bar{C}$	s, hr
Nonanalogies				
Power	$iE$	W	$q$	W, Btu/hr
Energy	$\int_0^{\tau} iE d\tau$	J, W · s	$\int_0^{\tau} q d\tau$	J, Btu

Table (4.1) Analogies and nonalogies between Thermal and Electrical Parameters

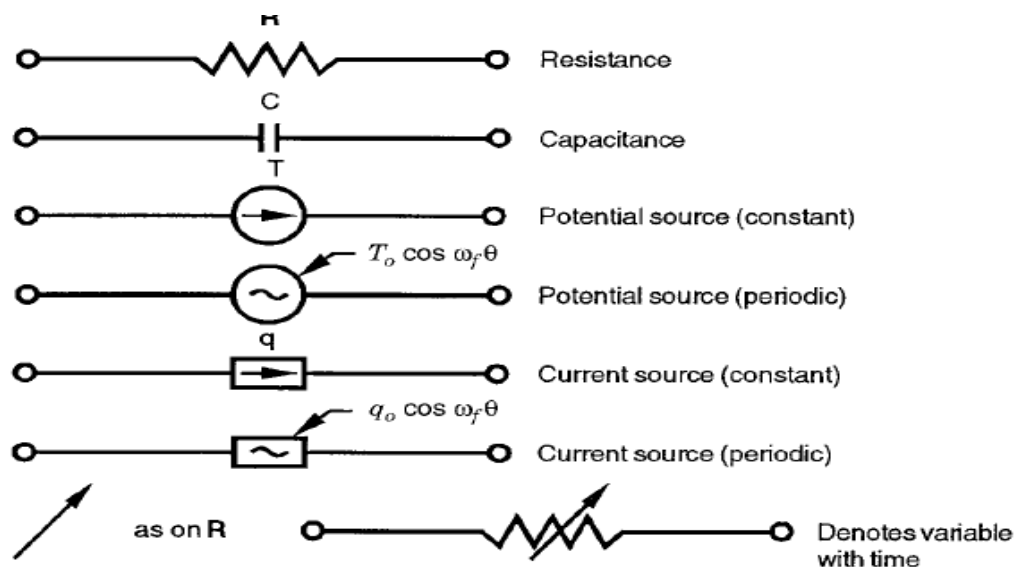


Figure (4.1) Thermal circuit symbolism

## 4.2 Heat exchanger variables and thermal circuit:

### 4.2.1 Assumptions for Heat Transfer Analysis:

1- The heat exchanger operates under steady-state conditions [i.e., constant flow rates and fluid temperatures (at the inlet and within the exchanger) independent of time.

2- Heat losses to or from the surroundings are negligible (i.e. the heat exchanger outside walls are adiabatic).

3- There are no thermal energy sources or sinks in the exchanger walls or fluids, such as electric heating, chemical reaction, or nuclear processes.

4- The temperature of each fluid is uniform over every cross section in counter flow and parallel flow exchangers (i.e., perfect transverse mixing and no temperature gradient normal to the flow direction). Each fluid is considered mixed or unmixed from the temperature distribution viewpoint at every cross section in single-pass cross flow exchangers, depending on the specifications. For a multi pass exchanger, the foregoing statements apply to each pass depending on the basic flow arrangement of the passes; the fluid is considered mixed or unmixed between passes as specified.

5-Wall thermal resistance is distributed uniformly in the entire exchanger.

6-Either there are no phase changes (condensation or vaporization) in the fluid streams flowing through the exchanger or the phase change occurs under the following condition. The phase change occurs at a constant temperature as for a single-component fluid at constant pressure, the effective specific heat ( $C_{p,eff}$ ) for the phase-changing fluid is infinity in this case, and hence ( $c_{max} = m \cdot c_{p,eff} \rightarrow \infty$ ) where  $m$  is the fluid mass flow rate.

- 7-Longitudinal heat conduction in the fluids and in the wall is negligible.
- 8-The individual and overall heat transfer coefficients are constant (independent of temperature, time, and position) throughout the exchanger, including the case of phase-changing fluids in assumption 6.
- 9-The specific heat of each fluid is constant throughout the exchanger, so that the heat capacity rate on each side is treated as constant. Note that the other fluid properties are not involved directly in the energy balance and rate equations, but are involved implicitly in (NTU) and are treated as constant.
- 10-For an extended surface exchanger, the overall extended surface efficiency is considered uniform and constant.
- 11-The heat transfer surface area  $A$  is distributed uniformly on each fluid side in a single-pass or multi pass exchanger. In a multi pass unit, the heat transfer surface area is distributed uniformly in each pass, although different passes can have different surface areas.
- 12-For a plate-baffled (1– $n$ ) shell-and-tube exchanger, the temperature rise (or drop) per baffle pass (or compartment) is small compared to the total temperature rise (or drop) of the shell fluid in the exchanger, so that the shell fluid can be treated as mixed at any cross section. This implies that the number of baffles is large in the exchanger.
- 13-The velocity and temperature at the entrance of the heat exchanger on each fluid side are uniform over the flow cross section. There is no gross flow maldistribution at the inlet.
- 14-The fluid flow rate is uniformly distributed through the exchanger on each fluid side in each pass i.e., no passage-to-passage or viscosity-induced

misdistribution occurs in the exchanger core. Also, no flow stratification, flow bypassing, or flow leakages occur in any stream. The flow condition is characterized by the bulk (or mean) velocity at any cross section.

### 4.3 Basic Definitions:

#### 4.3.1 The overall heat-transfer coefficient:

$$q = \frac{T_A - T_B}{1/h_1 A + \Delta x / kA + 1/h_2 A} \quad (4.2)$$

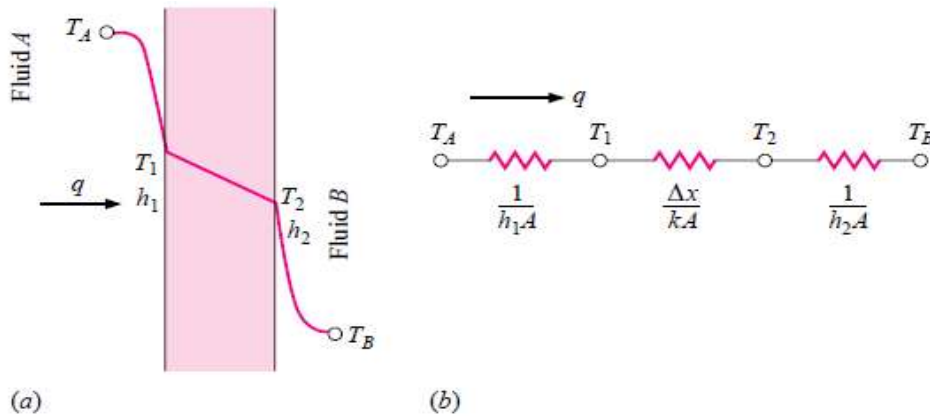


Figure (4.2): Over all heat transfer through a plane wall

The definitions of the mean overall heat transfer coefficient and mean temperature difference are introduced first.

$$\int_q \frac{dq}{\Delta T} = \int_A U dA \quad (4.3)$$

Now we define the mean temperature difference and mean overall heat transfer coefficient as follows using the terms in Eq (3-5).

$$\frac{1}{\Delta T_m} = \frac{1}{q} \int_q \frac{dq}{\Delta T} \quad (4.4)$$

$$U_m = \frac{1}{A} \int_A U dA \quad (4.5)$$



Substitution of Eq. (3-6) and (3.7) into Eq. (3.5) results into the following equation after rearrangement:

$$q = U_m A \Delta T_m \quad (4.6)$$

### 4.3.2 Thermal Circuit and UA:

In the steady state, heat is transferred from the hot fluid to the cold fluid by the following processes: convection to the hot fluid wall, conduction through the wall, and subsequent convection from the wall to the cold fluid. In many heat exchangers, a fouling film is formed as a result of accumulation of scale or rust formation, deposits from the fluid, chemical reaction products between the fluid and the wall material, and/or biological growth. This undesired fouling film usually has a low thermal conductivity and can increase the thermal resistance to heat flow from the hot fluid to the cold fluid. Thus, the heat transfer rate per unit area at any section  $dx$  (having surface areas  $dA_h$ ,  $dA_c$ , etc.) can be presented by the appropriate convection and conduction rate equations as follows:

$$dq = \frac{T_h - T_{h,f}}{dR_h} = \frac{T_{h,f} - T_{w,h}}{dR_{h,f}} = \frac{T_{w,h} - T_{w,c}}{dR_w} = \frac{T_{w,c} - T_{c,f}}{dR_{c,f}} = \frac{T_{c,f} - T_c}{dR_c} \quad (4.7)$$

Alternatively

$$dq = \frac{T_h - T_c}{dR_o} = U dA (T_h - T_c) \quad (4.8)$$

Where the overall differential thermal resistance ( $dR_o$ ) consists of component resistances in series.

$$\frac{1}{U dA} = dR_o = dR_h + dR_{h,f} + dR_w + dR_{c,f} + dR_c \quad (4.9)$$

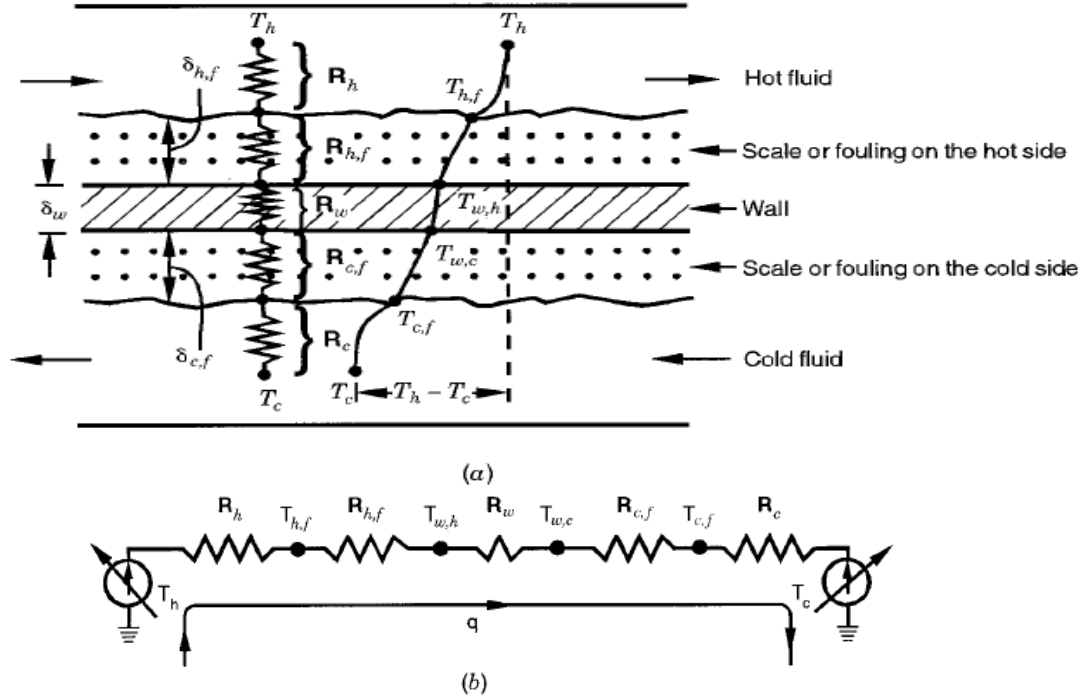


Figure (4.3): Thermal resistances: (b) thermal circuit for a heat exchanger(a)

Various symbols in this equation are defined after Eq. (4.9). If we idealize that the heat transfer surface area is distributed uniformly on each fluid side, the ratio of differential area on each fluid side to the total area on the respective side remains the same; that is

$$\frac{dA}{A} = \frac{dA_h}{A_h} = \frac{dA_c}{A_c} = \frac{dA_w}{A_w} \quad (4.10)$$

Replacing differential areas of Eq. (4.4) by using corresponding terms of Eq. (4.6), we get:

$$\frac{1}{UA} = \frac{1}{(\eta_o h A)_h} + \frac{1}{(\eta_o h_f A)_h} + R_w + \frac{1}{(\eta_o h_f A)_c} + \frac{1}{(\eta_o h A)_c} \quad (4.11)$$

It should be emphasized that (U) and all his in this equation are assumed to be local. Using the overall rate equation [Eq. (4.11)], the total heat transfer rate will be

$$q = U_m A \Delta T_m = U_m A (T_{h,e} - T_{c,e}) = \frac{1}{R_o} (T_{h,e} - T_{c,e}) \quad (4.12)$$

and a counterpart of Eq. (3-5) for the entire exchanger is:

$$q = \frac{T_{h,e} - T_{h,f}}{R_h} = \frac{T_{h,f} - T_{w,h}}{R_{h,f}} = \frac{T_{w,h} - T_{w,c}}{R_w} = \frac{T_{w,c} - T_{c,f}}{R_{c,f}} = \frac{T_{c,f} - T_{c,e}}{R_c} \quad (4.13)$$

Where the subscript e denotes the effective value for the exchanger.  $((T_{h,e} - T_{h,f}) = \Delta T_m)$  To be more precise, all individual temperatures in Eq. (4.9) should also be mean or effective values for respective fluid sides. However, this additional subscripts not included for simplicity.

$$\frac{1}{U_m A} = R_o = R_h + R_{h,f} + R_w + R_{c,f} + R_c \quad (4.14)$$

$$\frac{1}{U_m A} = \frac{1}{(\eta_o h_m A)_h} + \frac{1}{(\eta_o h_{m,f} A)_h} + R_w + \frac{1}{(\eta_o h_{m,f} A)_c} + \frac{1}{(\eta_o h_m A)_c} \quad (4.15)$$

For constant and uniform (U) and h is throughout the exchanger, Eq. (4.15) and (4.14) are identical. In that case,  $(U_m = U)$  and we will use (U) throughout the book except for Section 4.2. In Eq. (4.14) and (4.15), depending on the local or mean value, we define:

$$R_h = \text{hot fluid side convection resistance} = \frac{1}{(\eta_o h A)_h} \text{ or } \frac{1}{(\eta_o h_m A)_h}$$

$$R_{h,f} = \text{hot fluid side fouling resistance} = \frac{1}{(\eta_o h_f A)_h} \text{ or } \frac{1}{(\eta_o h_{m,f} A)_h}$$

$$R_{c,f} = \text{cold fluid side fouling resistance} = \frac{1}{(\eta_o h_f A)_c} \text{ or } \frac{1}{(\eta_o h_{m,f} A)_c}$$

$$R_c = \text{cold fluid side convection resistance} = \frac{1}{(\eta_o h A)_c} \text{ or } \frac{1}{(\eta_o h_m A)_c}$$

$$R_w = \text{wall thermal resistance.}$$

### 4.3.3 Effectiveness (NTU) method:

The (LMTD) approach to heat-exchanger analysis is useful when the inlet and outlet temperatures are known or are easily determined. The (LMTD) is then easily calculated and the heat flow surface area or overall heat-transfer coefficient may be determined. When the inlet or exit temperatures are to be evaluated for a given heat exchanger, the analysis frequently involves an iterative procedure because of the logarithmic function in the (LMTD). In these cases the analysis is performed more easily by utilizing a method based on the effectiveness of the heat exchanger in transferring a given amount of heat. The effectiveness method also offers many advantages for analysis of problems in which a comparison between various types of heat exchangers must be made for purposes of selecting the type best suited to accomplish a particular heat-transfer objective.

In the (NTU) method, the heat transfer rate from the hot fluid to the cold fluid in the exchanger is expressed as:

$$q = C_{min}(T_{h,i} - T_{c,i}) = C_{min}\Delta T_{max} \quad (4.16)$$

#### 1-Heat Exchanger Effectiveness E:

Effectiveness is a measure of thermal performance of a heat exchanger. It is defined for a given heat exchanger of any flow arrangement as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate.

( $q_{max}$ ) Thermodynamically permitted:

$$\epsilon = \frac{q}{q_{max}} \quad (4.17)$$

## 2-Heat Capacity Rate Ratio $C^*$ :

( $C^*$ ) is simply a ratio of the smaller to larger heat capacity rate for the two fluid streams so that ( $C^* \leq 1$ ) A heat exchanger is considered balanced when( $C^* = 1$ ).

$$C^* = \frac{C_{\min} (m \cdot c_p)_{\min}}{C_{\max} (m \cdot c_p)_{\max}} = \begin{cases} (T_{c,o} - T_{c,i}) / (T_{h,i} - T_{h,o}) & \text{for } C_h = C_{\min} \\ (T_{h,i} - T_{h,o}) / (T_{c,o} - T_{c,i}) & \text{for } C_c = C_{\min} \end{cases} \quad (4.18)$$

## 3-Number of Transfer Units (NTU):

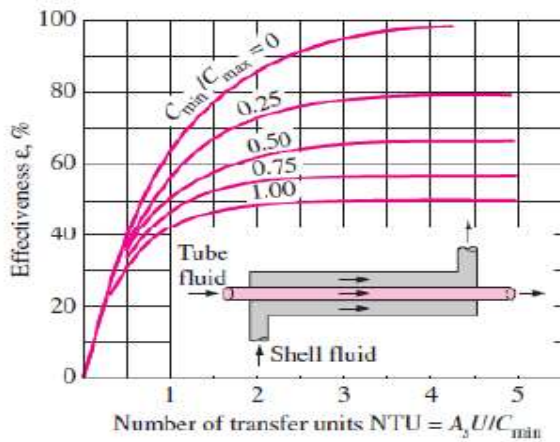
The number of transfer units (NTU) is defined as a ratio of the overall thermal conductance to the smaller heat capacity rate:

$$NTU = \frac{UA}{C_{\min}} = \frac{1}{C_{\min}} \int_A U dA \quad (4.19)$$

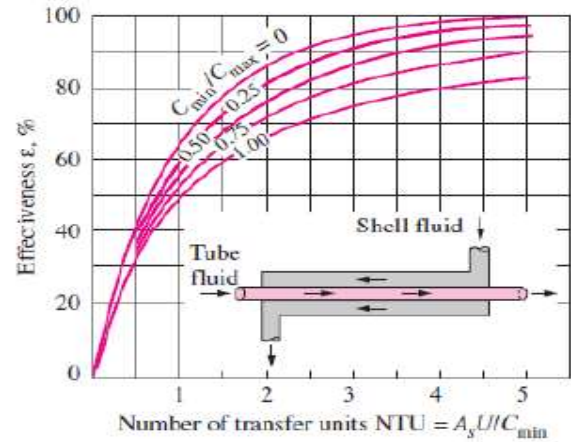
## 4.4 Effectiveness–number of transfer unit relationships:

Heat exchanger type	Effectiveness relation
1 <i>Double pipe:</i> Parallel flow	$\epsilon = \frac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counter-flow	$\epsilon = \frac{1 - \exp[-NTU(1 - c)]}{1 - c \exp[-NTU(1 - c)]}$
2 <i>Shell and tube:</i> One-shell pass 2, 4, ... tube passes	$\epsilon = 2 \left\{ 1 + c + \sqrt{1 + c^2} \frac{1 + \exp[-NTU\sqrt{1 + c^2}]}{1 - \exp[-NTU\sqrt{1 + c^2}]} \right\}^{-1}$
3 <i>Cross-flow</i> (single-pass) Both fluids unmixed	$\epsilon = 1 - \exp \left\{ \frac{NTU^{0.22}}{c} [\exp(-c NTU^{0.78}) - 1] \right\}$
$C_{\max}$ mixed, $C_{\min}$ unmixed	$\epsilon = \frac{1}{c} (1 - \exp[1 - c(1 - \exp(-NTU))])$
$C_{\min}$ mixed, $C_{\max}$ unmixed	$\epsilon = 1 - \exp \left\{ -\frac{1}{c} [1 - \exp(-c NTU)] \right\}$
4 <i>All heat exchangers with <math>c = 0</math></i>	$\epsilon = 1 - \exp(-NTU)$

Table (4.2) Effectiveness relations for heat exchangers

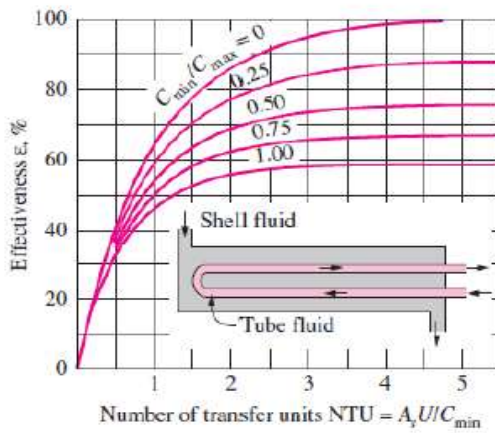


(a) Parallel-flow

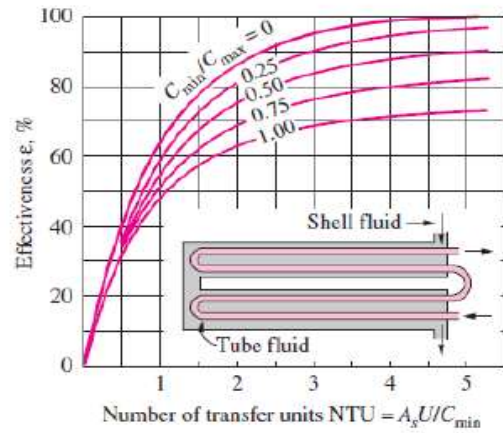


(b) Counter-flow

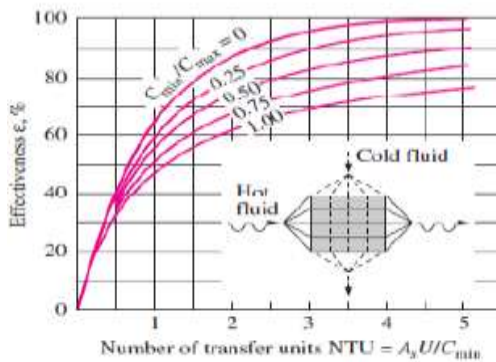
Figure (4.4): Number of transfer units (a) parallel flow (b) counter flow



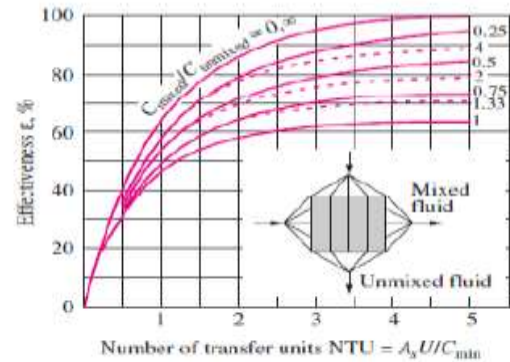
(c) One-shell pass and 2, 4, 6, ... tube passes



(d) Two-shell passes and 4, 8, 12, ... tube passes



(e) Cross-flow with both fluids unmixed



(f) Cross-flow with one fluid mixed and the other unmixed

Figure (4.5) Effectiveness–number of transfer unit relationships

## 4.5 The mean temperature difference method:

### 4.5.1 Log Mean Temperature Difference, LMTD:

(LMTD) method is very suitable for determining the size of a heat exchanger to realize prescribed outlet temperatures when the mass flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified. The log-mean temperature difference (LMTD or  $\Delta T_{lm}$ ) is defined as

$$LMTD = \Delta T_{lm} = \frac{\Delta T_I - \Delta T_{II}}{\ln(\Delta T_I / \Delta T_{II})}$$

### 3.13.2 Log Mean Temperature Difference Correction Factor (F):

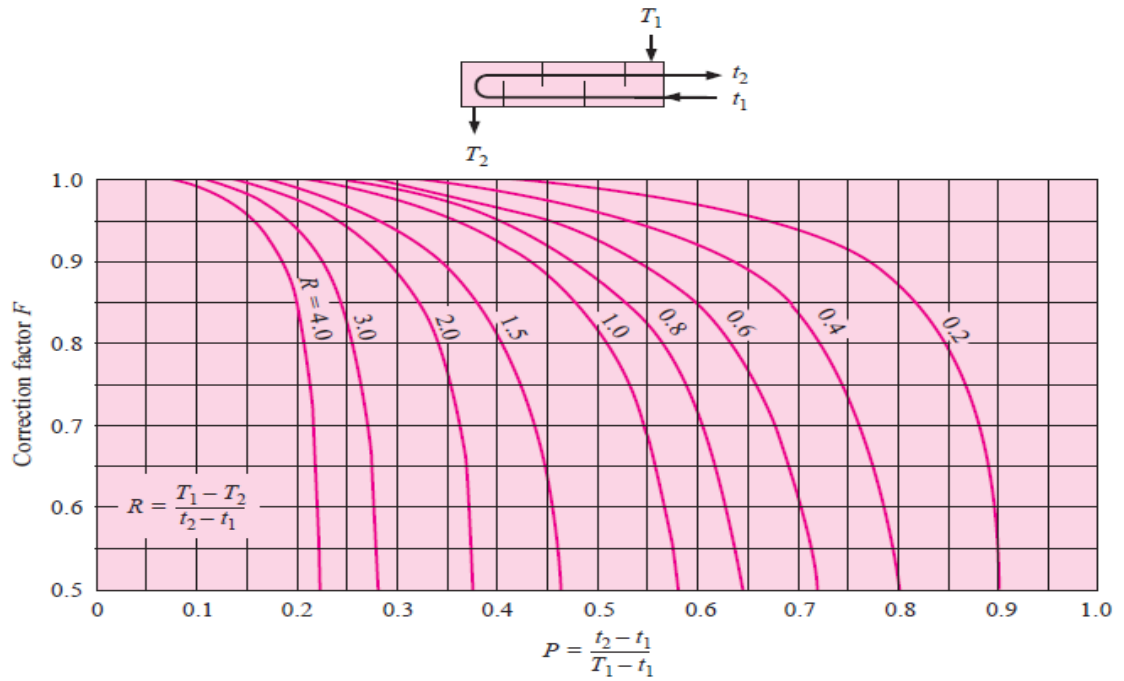
$$F = \frac{\Delta T_m}{\Delta T_{lm}} = \frac{q}{UA \Delta T_{lm}} \quad (4.20)$$

F as an Explicit Function of (P1) and (R1) only for the Specific Heat Exchanger Flow Arrangements Listed Here

Flow Arrangement	Formula
Counterflow	$F = 1$
Parallelflow	$F = 1$
Crossflow (single-pass) fluid 1 unmixed, fluid 2 mixed, stream asymmetric	$F = \frac{\ln[(1 - R_1 P_1)/(1 - P_1)]}{(R_1 - 1) \ln[1 + (1/R_1) \ln(1 - R_1 P_1)]}$ $= \frac{\ln[(1 - R_2 P_2)/(1 - P_2)]}{(1 - 1/R_2) \ln[1 + R_2 \ln(1 - P_2)]}$
fluid 1 mixed, fluid 2 unmixed, stream asymmetric	$F = \frac{\ln[(1 - R_1 P_1)/(1 - P_1)]}{(1 - 1/R_1) \ln[1 + R_1 \ln(1 - P_1)]}$ $= \frac{\ln[(1 - R_2 P_2)/(1 - P_2)]}{(R_2 - 1) \ln[1 + 1/R_2 \ln(1 - R_2 P_2)]}$
1-2 TEMA E, shell fluid mixed, stream symmetric	$F = \frac{D_1 \ln[(1 - R_1 P_1)/(1 - P_1)]}{(1 - R_1) \ln \frac{2 - P_1(1 + R_1 - D_1)}{2 - P_1(1 + R_1 + D_1)}}$ $= \frac{D_2 \ln[(1 - R_2 P_2)/(1 - P_2)]}{(1 - R_2) \ln \frac{2 - P_2(1 + R_2 - D_2)}{2 - P_2(1 + R_2 + D_2)}}$ <p>where <math>D_1 = (1 + R_1^2)^{1/2}</math> and <math>D_2 = (1 + R_2^2)^{1/2}</math></p>
All exchangers with $R_1 = 0$ or $\infty$	$F = 1$

Table(4.3):Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of tube passes.





Figure(4.6): Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of tube passes.

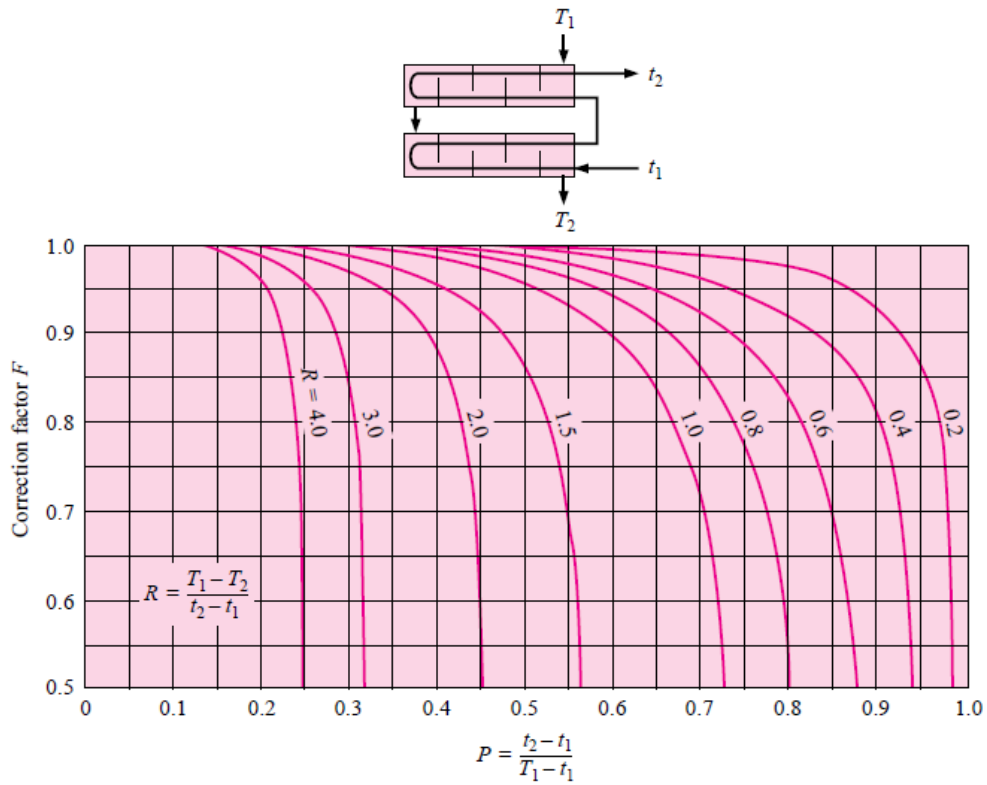


Figure (4.7): Correction-factor plot for exchanger with two shell passes and four, eight, or any multiple of tube passes.

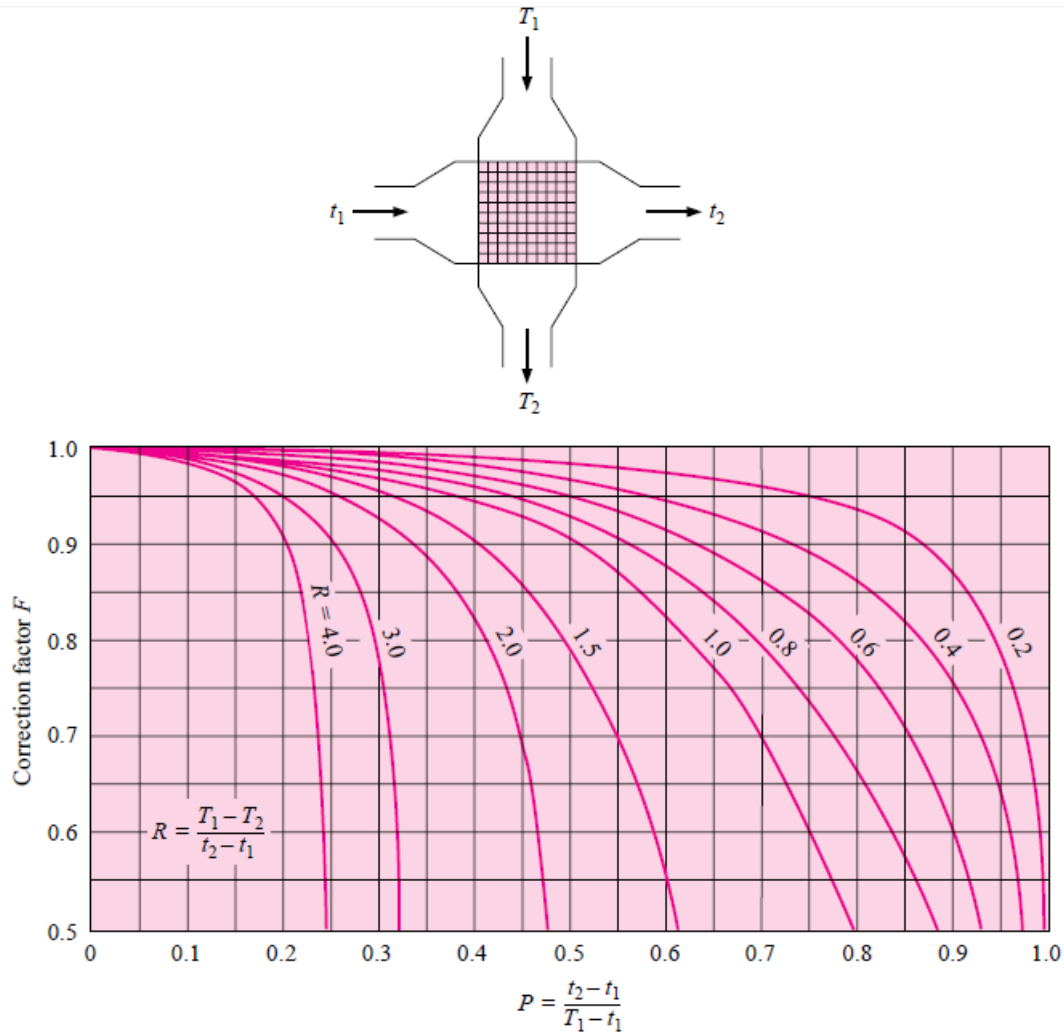


Figure (4.8): Correction-factor plot for single-pass cross-flow exchanger, both fluids unmixed.

## 4.6 Steps for design of Heat Exchanger:

Shell and tube heat exchanger have counter flow. In tube side cold naphtha is flow and hot naphtha is flow in shell side .assume (c) for cold naphtha and (h) for hot naphtha.

- Cold naphtha inlet  $T_{ci} = 48^{\circ}\text{C}$  .
- Cold naphtha outlet  $T_{co} = 130^{\circ}\text{C}$  .
- Cold naphtha flow rate  $\dot{m}_c = 9.16 \text{ Kg/s}$  .
- Hot naphtha inlet  $T_{hi} = 187^{\circ}\text{C}$  .
- Hot naphtha outlet  $T_{ho} = 106^{\circ}\text{C}$  .

- Hot naphtha flow rate  $m_{\dot{h}} = 8.3 \text{ Kg/s}$  .

Assume tube diameter  $d_o = 0.0508 \text{ m}$  and tube length  $L = 6\text{m}$ .

Assume fouling factor based on inside and outside tubes:

$$\frac{1}{h_{d_i}} = \frac{1}{h_{d_o}} = 0.0009$$

Assume material of construction for the tubes is carbon steel and the thermal conductivity founded  $50 \text{ W/m}^2 \text{ }^\circ\text{C}$ .

#### 4.6.1 Log Mean Temperature Difference, (LMTD). For counter current:

$$\begin{aligned} LMTD &= \frac{(Thi - Tco) - (Tho - Tci)}{\ln \frac{(Thi - Tco)}{(Tho - Tci)}} \\ &= \frac{(187 - 130)(106 - 48)}{\ln \frac{(187-130)}{(106-48)}} = 57.49 \end{aligned}$$

#### 4.6.2 Heat calculation:

$$\begin{aligned} q &= m_c \times c_p (T_{c,o} - T_{c,i}) \\ &= 9.16 \times 1.72 \times (130 - 48) \\ &= 1292.8 \text{ Kw} \end{aligned}$$

#### 4.6.3 Based of the exchanger configuration obtains the Temperature correction factor:

For one shell and one tube pass exchanger ( $F_t = 1$ ).

1. Calculate the mean temperature difference using

$$DT_m = Ft \times LMTD$$

$$= 1 \times 57.49 = 57.49 \text{ } ^\circ\text{C}$$

2. Assume overall heat transfer coefficient as initial guess from the table bellow:

$$U=300 \text{ W/m}^2 \text{ } ^\circ\text{C}$$

Shell and tube exchangers		
Hot fluid	Cold fluid	$U \text{ (W/m}^2\text{ } ^\circ\text{C)}$
<i>Heat exchangers</i>		
Water	Water	800–1500
Organic solvents	Organic solvents	100–300
Light oils	Light oils	100–400
Heavy oils	Heavy oils	50–300
Gases	Gases	10–50
<i>Coolers</i>		
Organic solvents	Water	250–750
Light oils	Water	350–900
Heavy oils	Water	60–300
Gases	Water	20–300
Organic solvents	Brine	150–500
Water	Brine	600–1200
Gases	Brine	15–250
<i>Heaters</i>		
Steam	Water	1500–4000
Steam	Organic solvents	500–1000
Steam	Light oils	300–900
Steam	Heavy oils	60–450
Steam	Gases	30–300
Dowtherm	Heavy oils	50–300
Dowtherm	Gases	20–200
Flue gases	Steam	30–100
Flue	Hydrocarbon vapours	30–100
<i>Condensers</i>		
Aqueous vapours	Water	1000–1500
Organic vapours	Water	700–1000
Organics (some non-condensables)	Water	500–700
Vacuum condensers	Water	200–500
<i>Vaporisers</i>		
Steam	Aqueous solutions	1000–1500
Steam	Light organics	900–1200
Steam	Heavy organics	600–900

Table (4.4): overall heat transfer coefficient

#### 4.6.4 Calculate the provisional area:

$$A = \frac{q}{U \cdot DT_m}$$
$$= \frac{1292.86}{300 \times 57.49} = 74.964 \text{ m}^2$$

#### 4.6.5 Number of tubes:

$$N_t = \frac{A}{\pi \cdot d_o \cdot L}$$
$$= \frac{74.946}{\pi \times 0.0508 \times 6} = 78.5 \cong 80 \text{ Tubs}$$

#### 4.6.6 Tube pitch and the bundle diameter:

$$P_t = 1.25d_o$$
$$= 1.25 \times 0.0508 = 0.0635 \text{ m}$$

$$D_b = d_o \left( \frac{N_t}{K_1} \right)^{\frac{1}{n_1}}$$

$N_t \equiv$  Number of tube

$D_b \equiv$  Bundle diameter (m)

$d_o \equiv$  Tube outside diameter (m)

$$= 0.0508 \times \left( \frac{80}{0.319} \right)^{\frac{1}{2.142}}$$
$$= 0.6698 \text{ m}$$

Where  $K_1$  and  $n_1$  are obtained from the table below based on the type of tube arrangement (Triangular pitch):

Triangular pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
$K_1$	0.319	0.249	0.175	0.0743	0.0365
$n_1$	2.142	2.207	2.285	2.499	2.675

Table (4.5) based on the type of tube arrangement (Triangular pitch)

#### 4.6.7 Calculate the shell diameter:

$$Ds = Db + BDC$$

$$= 0.6698 + 0.093 = 0.7628 \text{ m}$$

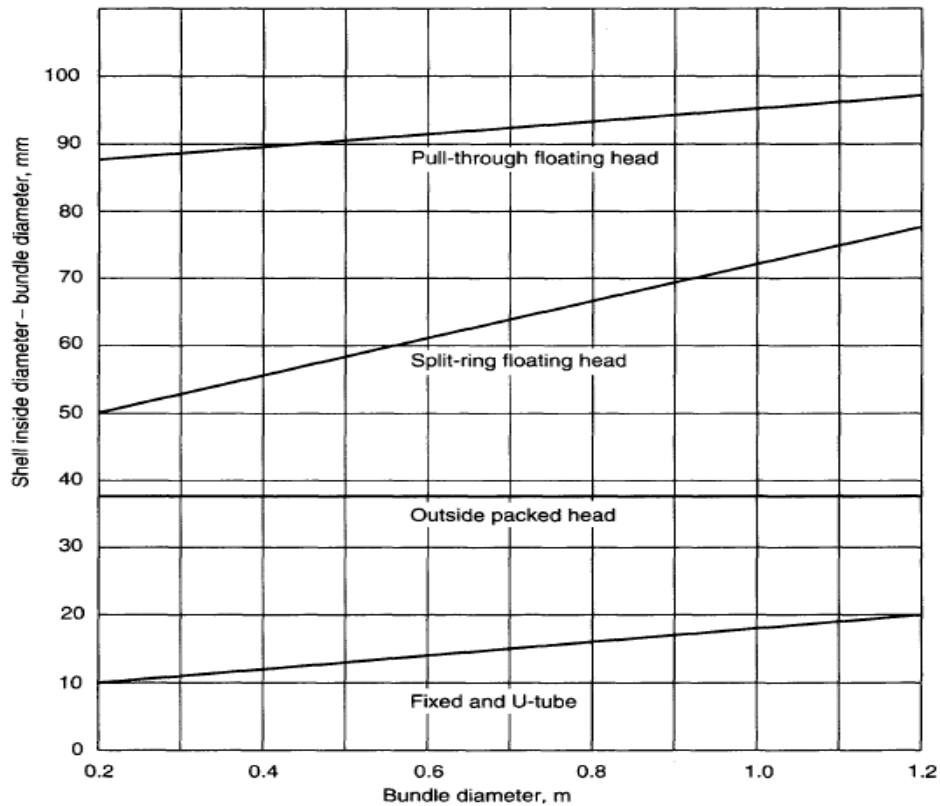


Table (4.6) : Type of floating head of the exchanger and the bundle diameter clearance.

#### 4.6.8 The baffle spacing:

$$Bs = 0.4Ds$$

$$= 0.04 \times 0.7628 = 0.30512 \text{ m}$$

#### 4.6.9 The area for cross-flow:

$$A_s = \frac{(p_t - d_o) D_s B_s}{p_t}$$
$$= \frac{(0.0635 - 0.0508) \times 0.07628 \times 0.030512}{0.0635} = 0.0477 \text{ m}^2$$

#### 4.6.10 The shell-side mass velocity:

$$G_s = \frac{\text{shell - side flow rate [kg/s]}}{A_s}$$
$$= \frac{8.3}{0.0477} = 174.14 \text{ Kg/m}^2 \cdot \text{s}$$

#### 4.6.11 The shell equivalent diameter:

For an equilateral triangular pitch arrangement:

$$d_e = \frac{1.10}{d_o} (p_t^2 - 0.917 d_o^2)$$

where  $d_e \equiv \text{equivalent diameter (m)}$

$$= \frac{1.10}{0.0508} (0.0635^2 - 0.917 \cdot 0.0508^2)$$
$$= 0.036 \text{ m}$$

#### 4.6.12 The shell-side Reynolds number:

$$R_e = \frac{G_s \times d_e}{\mu}$$
$$= \frac{173.75 \times 0.036}{0.00135} = 4633$$

#### 4.6.13 The Prandtl number:

$$\begin{aligned} \text{Pr} &= \frac{\mu.Cp}{k} \\ &= \frac{0.00135 \times 1.72}{0.15} = 15.48 \end{aligned}$$

#### 4.6.14 The shell-side heat transfer coefficient

$$Nu = \frac{h_s d_e}{k_f} = j_h Re Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$

Assume  $\left( \frac{\mu}{\mu_w} \right) = 1$

Where  $j_h$  is obtained from the chart bellow

$$j_h = 0.037$$

$$Nu = 0.037 \times 4633 \times 15.48^{1/3} \times (1)^{0.14} = 427$$

$$h_s = \frac{427 \times 0.15}{0.037} = 1780.22 \text{ W/m}^2 \text{ } ^\circ\text{C}$$



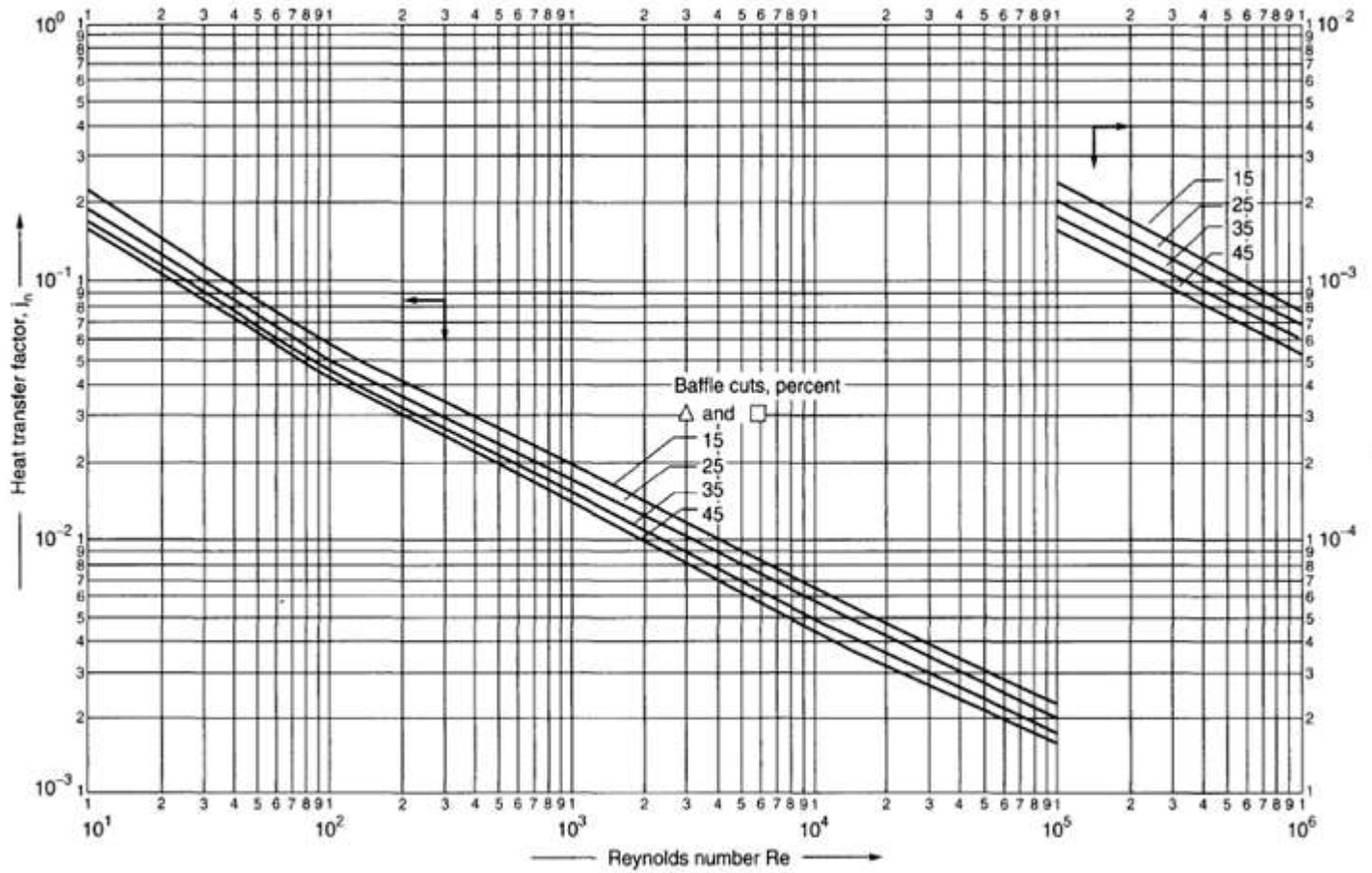


Table (4.7): shell-side heat transfer factors, segmental baffles

#### 4.6.15 The pressure drop in the shell:

$$\Delta P_s = 8 j_f \left( \frac{D_s}{d_e} \right) \left( \frac{L}{l_B} \right) \frac{\rho u_s^2}{2} \left( \frac{\mu}{\mu_w} \right)^{-0.14}$$

where  $L$  = tube length,  
 $l_B$  = baffle spacing.

$$\Delta P_s = 8 \times 0.15 \times \left( \frac{0.7628}{0.036} \right) \times \left( \frac{6}{0.030512} \right) \times \frac{665.11172^2}{2} \times (1)^{-0.14} = 20.71 \text{ KN/m}^2$$

Where  $j_f$  may be obtained from the chart below:

$$j_f = 0.15$$

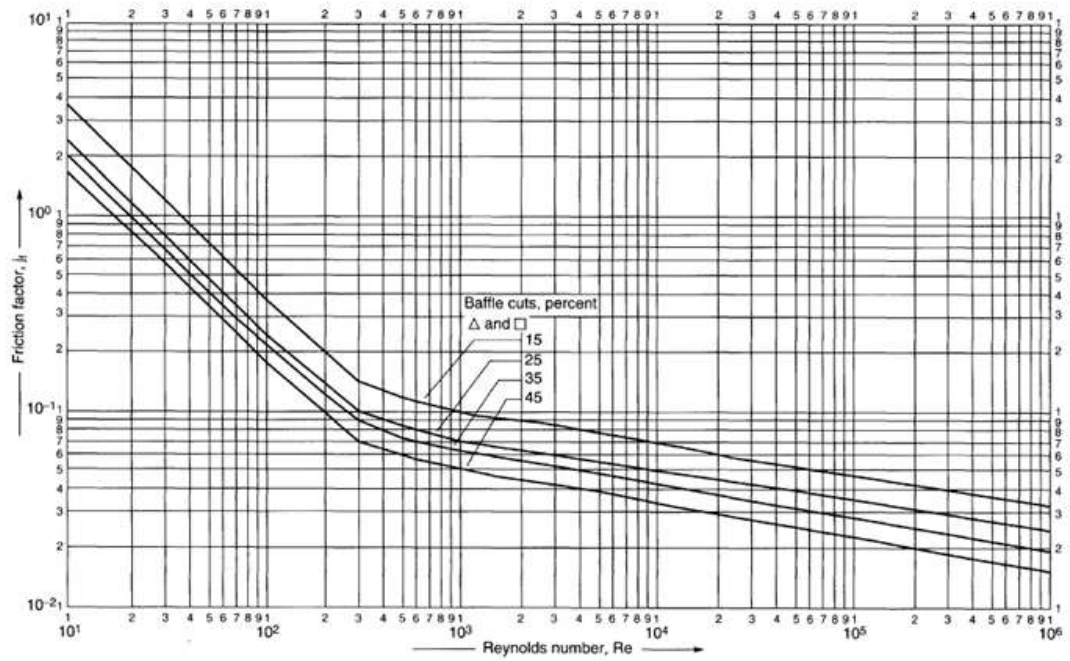


Table (4.8): between Reynolds number and Friction factor

#### 4.6.16 The number of tubes per pass:

$$N_{tpp} = N_t / \text{number of passes}$$

$$N_{tpp} = \frac{80}{1} = 80$$

#### 4.6.17 Tube-side mass velocity:

$$G_m = \frac{\text{tube - side flowrate[kg/s]}}{N_{tpp} \times \pi d_i^2 / 4}$$

$$= \frac{9.16}{80 \times \frac{\pi \times (0.0448)^2}{4}} = 72.16 \text{ Kg/m.s}$$

#### 4.6.18 Tube-side velocity:

$$v = \frac{G_m}{\rho_i}$$

$$= \frac{72.16}{665} = 0.1085 \text{ m/s}$$

Where  $\rho_i$  is the density of fluid inside tubes.

$$\rho_i = 665 \text{ Kg/m}^3$$

#### 4.6.19 Prandtl and Reynolds numbers for fluids inside tubes:

$$\text{Pr} = \frac{\mu \cdot C_p}{k} = 15.48$$

$$\text{Re} = \frac{\rho_i d_i v}{\mu_i}$$

$$= \frac{665 \times 0.0448 \times 0.1085}{0.00135} = 2394$$

where subscript  $i$  refers to fluid inside tubes

#### 4.6.20 Heat transfer coefficient ( $h_i$ ) by using either the following relations:

If  $\text{Re} > 2100$  (Transition and Turbulent)

$$h_i = 0.023 \frac{k_f}{d_i} \text{Re}^{0.8} \text{Pr}^{0.33} \left(1 + \frac{d_i}{L}\right)^{0.7}$$

$$= 0.023 \times \frac{0.15}{0.0448} \times 2394^{0.8} \times 15.48^{0.33} \left(1 + \frac{0.0448}{6}\right)^{0.7}$$

$$= 96.544 \text{ W/m}^2 \text{ } ^\circ\text{C}$$

#### 4.6.21 The overall heat transfer factor Based on (inside tubes flow):

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{1}{h_{di}} + \frac{d_i \ln(d_o / d_i)}{2k_w} + \frac{d_i}{d_o h_{do}} + \frac{d_i}{d_o h_o}}$$

$$= \frac{1}{\frac{1}{96.54} + .0009 + \frac{.0448 \ln\left(\frac{.0448}{.0508}\right)}{100} + \frac{.0448}{.0508 \times 1780} + \frac{.0448}{0.0508} \times .0009}$$

$$= 79.343 \text{ W/m}^2 \text{ } ^\circ\text{C}$$

Or based on (outside tubes flow)

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{1}{h_{do}} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{d_o}{d_i h_o} + \frac{d_o}{d_i h_{di}}}$$

Where ( $h_{di}$ ) and ( $h_{do}$ ) are the heat transfer coefficients for the scales (dirt) inside and outside tubes, respectively.

$$\begin{aligned} & \frac{1}{\frac{1}{1780} + .0009 + \frac{.0508 \ln(\frac{.0508}{.0448})}{100} + \frac{.0508}{.0448 \times 1780} + \frac{.0508}{0.0448} \times .0009} \\ & = 314.16 \text{ w/m}^2 \text{ } ^\circ\text{C} \end{aligned}$$

$$\begin{aligned} U_{\text{error}} &= \frac{U_{\text{assume}} - U_{\text{calculate}}}{U_{\text{assume}}} \\ &= \frac{300 - 314.16}{300} = 4.45\% \end{aligned}$$

**4.6.22 The tube-side pressure drop may be calculated using the relation:**

$$\begin{aligned} \Delta P &= \left( 1.5 + N_t \left[ 2.5 + \frac{8j_f L}{d_i} + \left( \frac{\mu}{\mu_w} \right)^{-m} \right] \right) \frac{\rho_i v^2}{2} \\ &= (1.5 + 80 \left[ 2.5 + \frac{8 \times 0.15 \times 6}{0.0448} + 1 \right]) \frac{665 \times 0.1085^2}{2} = 51.42 \text{ KN/m}^2 \end{aligned}$$

# **Chapter Five**

# Conclusion and Recommendations

## 5.1 Conclusion

It has been decided the size of pipes and material for heat exchanger to cool the (NAFTA). And decrease the losses in the exchanger, and develop computer program (MATLAB) for design calculation.

## 5.2 Recommendations:

The importance of heat exchanger in understanding the knowledge of heat transfer for it includes various types of heat exchangers, there for it should be an obligation for the students of bachelor degree and master degree to make calculations of design.

The importance of visiting thermal power plants and factories to understand the heat exchangers (types, how it is work, their used and how to design it's).

The need to engage engineers chemists to understand the properties of fluid and heat exchange intended them to choose the appropriate article for the design of the heat exchanger.

The good heat exchangers Characterized by which are used to achieve the purpose for which must provide the following requirements:

- 1- High thermal effect shall give a higher coefficient of thermal transfer
- 2- The pressure loss as little as possible.
- 3- Be resistant to the effects of corrosion and erosion resulting from the ongoing fluids to conduct heat exchange and its impact on the materials used in manufactured.
- 4- To bear pressure and high degrees of operational temperature.

- 5- Be less investment and operational cost, and with a long operational life.
- 6- It is easy to maintain, with easy switching parts.

### 5.3 References:

1-Fundamentals of heat and mass transfer seven edition.

Author: Theodor albergman.

Company amacmillian company-2011

2-Holman Heat Transfer 10<sup>th</sup> txt-book.

Author: Jack P. Holman.

Company: Copyright © 2010 by the McGraw-Hill Companies.

3-Fundamentals of Momentum Heat and Mass Transfer 5<sup>th</sup> Edition

Author : James R. Welty.

Company : James R. Welty 2008.

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Company: Copyright by John Wiley & Sons, Inc. All rights reserved.

5- Fundamentals of heat and mass transfer third edition.

Author: C.P. kothandarman.

Company : new age international (p) limited-2006.

6-heat and mass transfer mechanical engineering handbook

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Author: warren M. rohsenow-james p. hartnett-young l. cho-1998.

8- Heat transfer second edition.

Author: B.F. Carlicar.

Company: West publishing co-1982.



# Appendix

## 5.4 Appendix:



Appendix No (1): tubes and bundle.



Appendix No (2): bundle.



Appendix No (3): U tube.



Appendix No (4): floating head and bundle tube.

## 5.4 program to design of heat Exchanger:

```
Tci=input('cold naphtha inlet') % cold naphtha inlet
Tco=input('cold naphtha outlet'); % cold naphtha outlet
Mc=input('cold naphtha flow rate') % cold naphtha flow
rate
Thi=input('hot naphtha inlet') % hot naphtha inlet
Tho=input('hot naphtha outlet') % hot naphtha outlet
Mh=input('hot naphtha flow rate') % hot naphtha flow rate
do=input('tube diameter outside ') % tube diameter
L=input('tube length')%tube length
hd=input('fouling factor based on inside and outside
tubes') %fouling factor based on inside and outside
tubes
Km=input('thermal conductivity of material')%thermal
conductivity of material
Ft=input('correction factor')% correction factor
U=input('overall heat transfer coefficient ')%overall
heat transfer coefficient
Ki=input('const for Triangular or square pitch')%const
for Triangular or square pitch
ni=input('const for Triangular or square pitch')
%const for Triangular or square pitch
BDC=input('Bundle diameter clearance') %Bundle diameter
clearance
W=input('viscosity')%viscosity
Z=input('denstiy')%denstiy
Kf=input('thermal conductivity of liquid')%thermal
conductivity of liquid
cp=input('specific heat capcity')% specific heat
capacity
Jh=input('heat transfer factor')%heat transfer factor
Jf=input('friction factor')%friction factor
Nb=input('number of passes')%number of passes
di=input('tube diameter inside')
format short g
x=log((Thi-Tco)/(Tho-Tci));
y=(Thi-Tco)-(Tho-Tci);
LMTD=y./x
DTm=Ft.*LMTD
q=Mc.*cp.*(Tco-Tci).*1000
```

```

A=q./ (DTm.*U)
Nt=A./ (pi.*do.*L)
Pt=1.25.*do
pp=1./ni
Db=do.* (250.7.^pp)
Ds=Db+BDC
Bs=0.4.*Ds
As=(Pt-do) .*Ds.*Bs./Pt
Gs=Mh./As
de=(1.10./do) .* ((Pt.^2) - ((0.917*do^2)))
Re=Gs.*de./W
Pr=(W.*cp./Kf) .*1000
Nu=Jh.*Re.*(Pr.^(1/3))
ho=Nu.*Kf./de
Ntpp=Nt./Nb
Gm=(Mc.*4) ./ (Ntpp.*pi.*di.^2)
V=Gm./Z
R2=(Z.*di.*V) ./ (W)
pressuredrop=(8.*Jf) .* (Ds./de) .* (L./Bs) .* (Z.*(V.^2) ./ (2
))
hi=(.023.*Kf.*R2.^(.8) ./di) .* (Pr.^(.33)) .* (1+di./L) .^(.
7)
bb=di.*log(di./do) ./ (2.*Km)
bbb=(1./hi) +(1./hd) +bb+ (di./ (do.*hd) + (di./ (do.*ho)))
Ui=1./bbb
cc=do.*log(do./di) ./ (2.*Km)
ccc=(1./ho) +(1./hd) +cc+ (do./ (di.*ho) + (do./ (di.*hd)))
Uo=(1./ccc)
Uer=(U-Uo) ./U
tubesidepressuredrop=1.5+Nt.*(2.5+(8.*Jf.*L./di)+1) .* ((
Z.*V.^2) ./2)

```

## 5.6 Run of MATLAB program:

```
cold naphtha inlet[48]
```

```
Tci =
```

```
48
```

```
cold naphtha outlet[130]
```

```
cold naphtha flow rate[9.16]
```

```
Mc =
```

```
9.1600
```

```
hot naphtha inlet[187]
```

```
Thi =
```

```
187
```

```
hot naphtha outlet[106]
```

```
Tho =
```

```
106
```

```
hot naphtha flow rate[8.3]
```

```
Mh =
```

```
8.3000
```

```
tube diameter outside [.0508]
```

```
do =
```

```
0.0508
```

```
tube length[6]
```

```
L =
```

```
6
```

```
fouling factor based on inside and outside tubes[1111.11]
```

```
hd =
```

```
1.1111e+003
```

```
thermal conductivity of material[50]
```

```
Km =
```

```
50
correction factor[1]

Ft =

1
overall heat transfer coefficient [300]

U =

300
const for Triangular or square pitch[.319]

Ki =

0.3190
const for Triangular or square pitch[2.142]

ni =

2.1420
Bundle diameter clearance[.093]

BDC =

0.0930
viscosity[.00135]

W =

0.0014
denstiy[665]

Z =

665
thermal conductivity of liquid[.15]

Kf =

0.1500
specific heat capcity[1.72]

cp =
```



1.7200

heat transfer factor[.037]

$J_h =$

0.0370

friction factor[.15]

$J_f =$

0.1500

number of passes[1]

$N_b =$

1

tube diameter inside[.0448]

$d_i =$

0.0448

LMTD =

57.499

$DT_m =$

57.499

$q =$

1.2919e+006

$A =$

74.896

$N_t =$

78.216

$P_t =$

0.0635

pp =

0.46685

Db =

0.66976

Ds =

0.76276

Bs =

0.3051

As =

0.046544

Gs =

178.33

de =

0.036071

Re =

4764.7

Pr =

15.48

Nu =

439.37

ho =

1827.1

Ntpp =

78.216

Gm =

74.294

V =

0.11172

R2 =

2465.5

pressuredrop =

2071

hi =

98.843

bb =

-5.6308e-005

bbb =

0.012237

Ui =

81.719

cc =

6.385e-005

ccc =

0.0031523

$U_o =$

317.23

$U_{er} =$

0.057425

tubewise pressuredrop =

53306