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HEAT TRANSFER ANALYSIS ON CRUDE OIL AND PETROLEUM PRODUCTS USING THE HEAT EXCHANGERS USED IN PETROLEUM INDUSTRIES

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A thesis submitted by Mahmood BANDER KANAAN in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE** is approved by the committee on **26-4-2017** in Department of Mechanical Engineering , Program of Heat Processing .

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February, 2017

Mahmood BANDER KANAAN

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LIST OF SYMBOLS

Q heat duty of heat exchanger, W
Ch specific heat of the hot fluid, J/kgK
Cc specific heat of the cold fluid, J/kgK
Thi temperature of the hot fluid inside, K
Tho temperature of the hot fluid outside, K
Tci temperature of the cold fluid inside, K
Tco temperature of the cold fluid outside, K
m· Mass flow rate, kg/sec
C heat capacity
K thermal conductivity, W/mK
Mw dynamic viscosity of water fluid, Ns/m2
Mb shell fluid dynamic viscosity at average temperature, Ns/m2
Gs mass velocity of shell side, kg/m2. S
De equivalent diameter of shell side, m
ΔP pressure drop for shell side, Pa
Ǿs viscosity correction factor for shell side fluid
Ds shell diameter, m
C clearance, m
B baffle spacing, m
As area of the shell, m2
Pt tube pitch, m
ho heat transfer coefficient, W/m2.k
Nt number of tubes
Gt mass velocity of tube, kg/m2.s
Atp heat transfer area based on tube surface, m2
ρ density of fluid at average temperature, kg/m3
di inner diameter of tube, m

LIST OF ABBREVIATIONS

LMTD	log mean temperature difference
Nb	number of baffles
NTU	number of transfer units
Pr	Prandtl number
Re	Reynolds number
ΔTLM	Logarithmic mean temperature difference (K)

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ABSTRACT

HEAT TRANSFER ANALYSIS ON CRUDE OIL AND PETROLEUM PRODUCTS USING THE HEAT EXCHANGERS USED IN PETROLEUM INDUSTRIES

Mahmood BANDER KANAAN

Department of Mechanical Engineering

MSc. Thesis

Advisor: Prof. Dr. Hasan Alpay HEPERKAN

Shell-and-tube heat exchangers are utilized for the most part in the mechanical and chemical processes, petroleum industries and power plants, particularly in refineries, since various advantages they offer over different models of heat exchangers. Numerous data are available about their development and plan. The present notes are proposed only to assist as a brief introduction. It is necessary to operate heat exchangers at optimum condition which serves high thermal efficiency in acceptable conditions and low running costs.

This research is planned to help anybody with general technical experience, but maybe constrained to the learning of heat transfer equipment. In this study temperatures of tube and shell side are taken as input parameters with a concurred bundle arrangement of square pitch. The thermal analysis has been done taking crude oil inside the tube and oil products on shell side. The design of shell and tube exchanger applying Kern method for combination has been proved by well-known Dittus-Boelter formula of turbulent flow inside tube. The analysis is extended by applying the Kern method above in heat exchangers with different fluid combinations and proof parameters such as heat transfer coefficient, friction coefficient, area, length and pressure drop are determined. In this study, the calculated numerical heat output from four heat exchangers was compared to those which were determined experimentally by its producer.

This study focuses on finding a proof of numerical model with the real data sheet from the refinery. “MATLAB” Program has been written to evaluate the parameters above. The tables have been written to describe the behavior of different fluid combinations in different heat exchangers. The aim of this study is to prove a numerical model for the tested heat exchangers by the refinery at real experimental operating conditions in the company. The results we got are tabulated.

Keywords: Heat Exchangers (shell and tube), crude oil, Kern Method, heat transfer.

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ÖZET

PETROL SANAYİSİNDE KULLANILAN ISI DEĞİŞTİRİCİLER YARDIMIYLA YAPILAN HAM PETROL VE PETROL ÜRÜNLERİNİN ISI AKTARIM ANALİZİ

Mahmood BANDER KANAAN

Makine Mühendisliği Bölümü
Yüksek Lisans Tezi

Tez Danışmanı: Prof. Dr. Hasan Alpay HEPERKAN

Gövde borulu ısı değiştiricileri diğer model ısı değiştiricilerinden bir çok yönden daha avantajlı olduğundan dolayı genel olarak kimyasal proseslerde, mekanikte, petrol endüstirisinde, enerji santrallerinde özellikle refinerilerde kullanılmaktadır. Tasarım ve yapımı konusunda bir çok bilgi bulunduğu için bu çalışmada sadece kısa bir giriş yapılmıştır. Kabul edilebilir koşul ve düşük işletme maliyetlerinde yüksek verim elde etmek için ısı değiştiricilerinin ideal koşularda çalıştıtırılması gerekmektedir.

Bu tezde boru ve gövde tarafındaki sıcaklıklar giriş parametersi olarak alınmış boru demeti dizimi kare seçilmiştir. Termal analizi boru içinde ham petrol, gövde kısmında yağ alınarak yapılmıştır. Ken metodu uygulanarak tasarlanan gövde borulu ısı değiştiricisi çok iyi bilinen Dittus-Boelter boru içindeki türbülanslı akış formülleriyle doğrulanmıştır. Analiz, yukarıda belirtilen Kern yöntemi farklı akışkan kombinasyonlarına sahip ısı değiştiricilerinde uygulanarak genişletilmiş, ısı transfer katsayısı, sürtünme katsayısı, alan, uzunluk ve basınç düşüşü parametreleri belirlenmiştir. Ayrıca, dört ısı değiştiricisi için matlab kodu ile sayısal olarak hesaplanmış ısı transferi, üretici tarafından deneysel olarak belirlenen ısı transferi ile karşılaştırılmıştır. Dört farklı ısı değiştircisi ve dört farklı akışkan için elde edilen sonuçlar tablolar halinde sunulmuş ve karşılaştırılmıştır.

Anahtar Kelimeler: Isı değiştiricleri (Gövde borulu), ham petrol, Kern Yöntemi, ısı transferi.

CHAPTER 1

INTRODUCTION

1.1 Literature Review

Heat exchangers are one of the most important tools of the mechanical and petroleum industries. Many mechanical procedures involve the transfer of heat and during that frequent event, it must be controlled by a heat transfer handle. Per, Oko [1], heat exchangers are a type of hardware with a specific limit in which heat is traded between two sides; with one side being cold and the opposite side being hot.

There are several kinds of heat exchangers, but the one most commonly utilized as a part of modern applications and refineries is the shell and tube. As its name suggests, this model of heat exchanger (shell and tube) consists of a shell with a bundle of tubes inside the heat exchangers.

One kind of fluid travels into a smooth motion through the tubes, and another fluid streams over the tubes to transfer heat between these liquids. The tube bundle may encompass various types of tubes (longitudinally finned, plain), and so forth. To ensure that the shell side fluid streams over the tubes and in this manner, leads to a higher heat transfer, baffles are set into the shell to force the shell-side fluid to travel crosswise over the tube to create the heat exchange and to keep an equal separation between the tubes.

Holman, [2], As the two fluids in the heat exchanger are at different temperatures, the heat exchanger plan and investigation in this manner includes both conduction and convection. Two critical issues in heat exchanger investigation is rating present heat exchangers and measuring heat exchangers for a focus. Rating includes resolving the issues of assessing a transfer, the adjustment in temperature of the two fluids and the pressure drop through the heat exchanger.

Measuring consists of collecting data from a particular heat exchanger from those directly accessible or characterizing the blueprint for the planned use of another heat exchanger, as well as determining the required rate of heat transfer and suitable pressure drops (Thirumarimurugan et al., [3]).

There are format charts, for example, Effectiveness, Numbers of Transfer Unit (∈-NTU), Logarithm Mean Temperatures Difference (LMTD), and correction factor curves for the analysis of simple types of heat exchangers. Comparable design graphs don't exist for the stream arrangements. In recent times, a few specialists' have done studies on the performance analysis, design, and recreation considerations of heat exchangers.

Modeling and Simulation of Heat Exchangers (Shell and Tube) under Milk Fouling were completed. A dynamic model for Heat Exchangers (Shell and Tube) was examined. Shell and Tube Heat exchangers are useful where high temperature and pressure necessities are noteworthy and can be utilized for a procedure requiring extensive amounts of liquid stream to be cooled or heated.

A trial to produce such diagrams and bends was created via many looks into the basics of thedesign of (shell and tube) heat exchangers, inventors carefully broke down this problem both experimentally and numerically. Kern [4] provided redress calculated charts to a various number of shells and a considerable number of tube passes. Kern basically alluded to the correction factor (F), as a component of two factors (S and R), which relies upon the inlet and outlet temperatures of the heat exchangers of both the liquids.

Nicole and Roetzel [5] have determined the latent effectiveness of obvious portrayals of LMTD, and correction factors creating modernized bundles for heat exchanger outlines. Tinker [6] has recommended a schematic stream design, which distributed the shell–side flow into a few streams. Tinker's demonstration has been the premise of the "Stream examination method," which uses a troublesome reiterative approach and it is especially appropriate for computer calculations, as opposed to hand calculations.

In Saunders' book [7], he proposed a practical method and straightforward outline. Elements are provided and the technique is utilized quickly for an unfaltering arrangement of geometrical parameters. In this work, the correction factors are utilized for heat transfer and pressure drop correlations. Wills and Johnston [8] built up a stream investigation technique that is clearly for hand computations. Wills and Johnston built up another, and a correct hand count method for shell and tube stream appropriations and pressure drop.

Reppich and Zagermann [9], had a paper that offers a PC based plan model to restrict the ideal measurements of fragment confederate baffled heat exchangers (shell and tube) by ascertaining the perfect shell and tube side pressure drops from the condition given in his work. The six-enhanced dimensional parameters are the number of tubes, length of tube, diameter of shell, number of baffles, baffle cut, and baffle spacing.

The proposed display additionally completes a similar cost analysis. Lam and Lo [10] presented the stream models and compared the investigation of four round barrels subjected to a cross stream. Examinations were completed at sub basic Re of 2100. Square course of action of the barrels with variable dividing proportions and limits of juncture were examined.

Mottos et al. [11], clarified a two-dimensional heat transfer analysis in circular and curved tube heat exchangers in his review. The technique for limited component is utilized for heat transfer conditions, liquid stream and four gestured, Two Dimensional Isoperimetric. Linear elements strategy was executed for the FEA advance. An and Choi [12], created a procedure for the point by point issues in shell and tube heat exchangers to anticipate the heat and mass transfer attributes of heat exchangers (shell and tube). Moghadassi and Hosseini [13], had emphasized the significance of heat exchangers in chemical and industries of petrochemical, analysis of heat exchangers and heat transfer calculations were given. The traditional and predominant techniques (like KERN strategy) are displayed. Liljana Markovska and Vera Mesko [14] gave streamlining of shell and tube heat exchangers achievement by utilization of the analyzer programming bundle. The objective capacity is characterized together with the understood limitations. The synchronous equation measuring technique is utilized to understand the conditions that show the procedure. Lebele_Alawa and Victor Egwanwo [15], displayed an answer of numerical technique, capable of considering temperature subordinate variety of the warmth exchange and liquid properties. Field information was collected for three diverse mechanical heat exchangers and fundamental overseeing conditions which were connected. The parameters broken down contain: outlet temperatures, the heat transfer.

coefficients and heat exchanger viability.

Naik and Mata Wala [16] contemplated the plan and evaluation of a counter stream shell and tube heat exchanger by entropy era minimization strategy. McAdams [17] was one of the most timely specialists to quantitatively explain. That McAdams' examination was a simple idea with respect to a type of tubular heat exchanger. By considering the cost and settled cost of powering the heat exchanger, transferred heat per unit, straightforward expressions for assessing the ideal mass speeds for both in and outside tubes liquids are produced.

Along these lines, they created the most detailed and reasonable work to date on the enhancement of single shell and tube heat exchangers. The issue with their strategy at the same time, is that it is limited to shell and tube heat exchangers fitted with a plain tube. Expansion to other exchanger types requires new conditions.

1.2 Objective of the Thesis

The present study focuses mainly on the design of heat exchangers known to be shell and tube heat exchangers.

The aim in this study is to prove a numerical model for the tested heat exchangers by their refinery in real life experimental operating conditions in their company.

1.3 Hypothesis

This study offers performance analyses of shell and tube heat exchangers. An analytical technique was utilized to create a relationship for the performance analysis. The program was composed in MATLAB to check for the thermal analyses and hydraulic suitability of the heat exchangers. The program was tried with information from four diverse industrial heat exchangers from Refinery. The results obtained showed reasonable agreement with the actual field data, subsequently showing that the program is dependable and can be applied in the performance analysis of heat exchangers (shell and tube).

CHAPTER 2

GENERAL INFORMATION

2.1 Introduction

The heat exchanger is one of the necessary tools in the process industries. It is used in most refineries and other large chemical and mechanical processes and is also suitable for higher temperature and pressure purposes. The heat exchangers are applied to a transfer heat between two process flows.We can understand their use in that any process which includes heating, cooling, boiling, condensation or evaporation will require a heat exchanger designed for these purposes.

Generally, process fluids must be cooled or heated by the procedure or undergo a phase change. Different heat exchangers are named according to their application. For example, it is known as a condensare heat exchanger when used to condensate, the same thing heat exchanger for boiling purposes are called boilers. Efficiency and performance of a heat exchanger is evaluated through the quantity of heat transfer using the minimum area of pressure drop and heat transfer.

The best way to manage its performance and efficiency is done by determining the over-all heat transfer coefficient. Pressure drop and area required for a specific amount of heat transfer, provides an insight into the capital cost and power requirements (running cost) of a heat exchanger. Generally, there are many theories and literature available to design a heat exchanger depending on the requirements. There are two types of heat exchangers:

- Direct contact, when both sides between which heat is exchanged are in direct contact with each other.

- Indirect contact, when they are separated using both sides of the wall through which heat is transferred to the path in any combination. A typical type of heat exchanger, usually used for higher temperature and pressure applications, is the shell and tube heat exchanger. Heat exchanger shell and tube of the indirect contact type is shown in Figure 2.1.

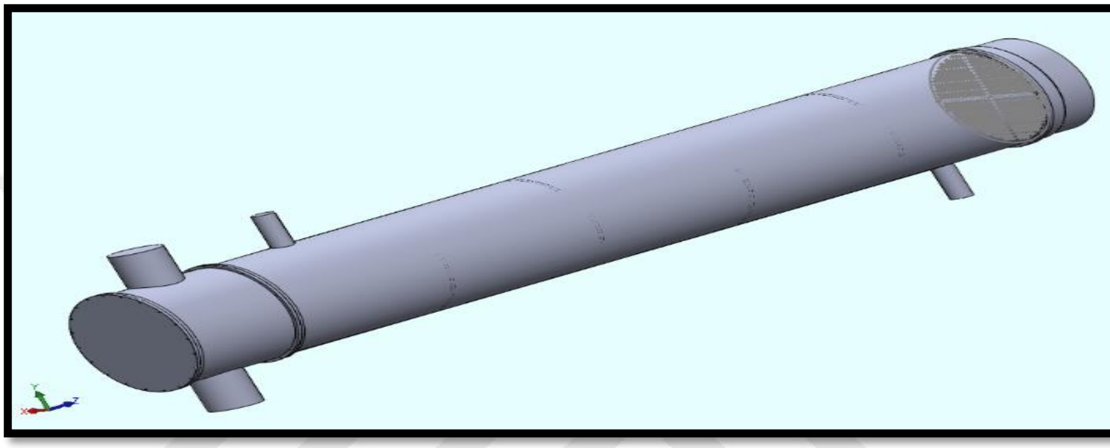

Figure 2. 1: Indirect heat exchanger (shell & tube)

It involves a series of tubes, through which one of the fluids runs. The main parts of a shell and tube heat exchangers comprise the shell, tubes, tube sheets, and shell-side nozzles, tube side channels and channel covers, baffles, nozzles, pass divider, etc. This type of heat exchanger comprises a shell shown in Figure 2.2 (a large pressure vessel) with a bundle of tubes inside it as shown in Figure 2.3, 2.4. One fluid flows through the shell (over the tubes), another fluid flows through the tubes to transfer heat between the two fluids [18].The shell is a compartment for the shell's fluids. Usually, the shell's shape is cylindrical in cross section a circular, although a different shape of shells is used in specific applications.

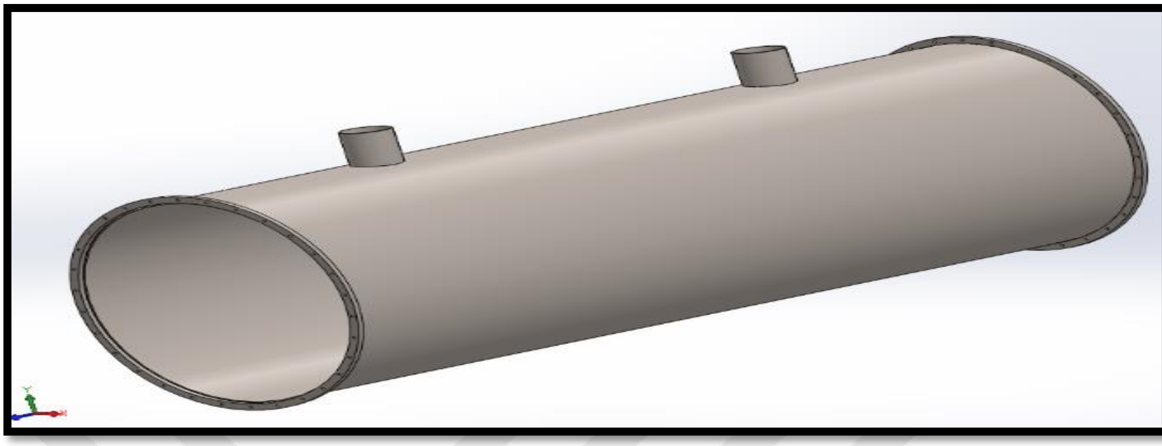

Figure 2.2: Shell

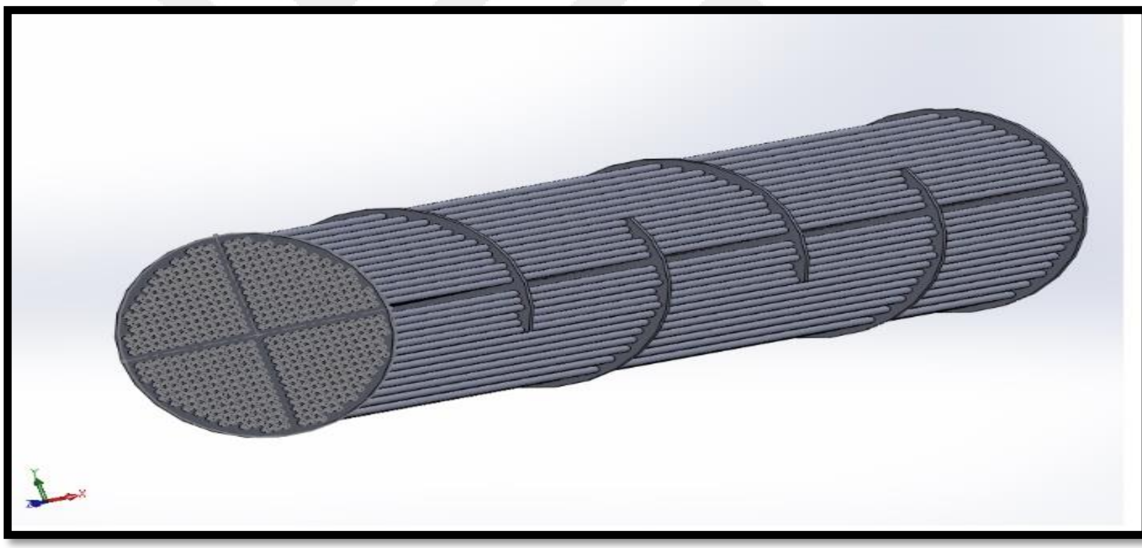

Figure2.3: Bundle

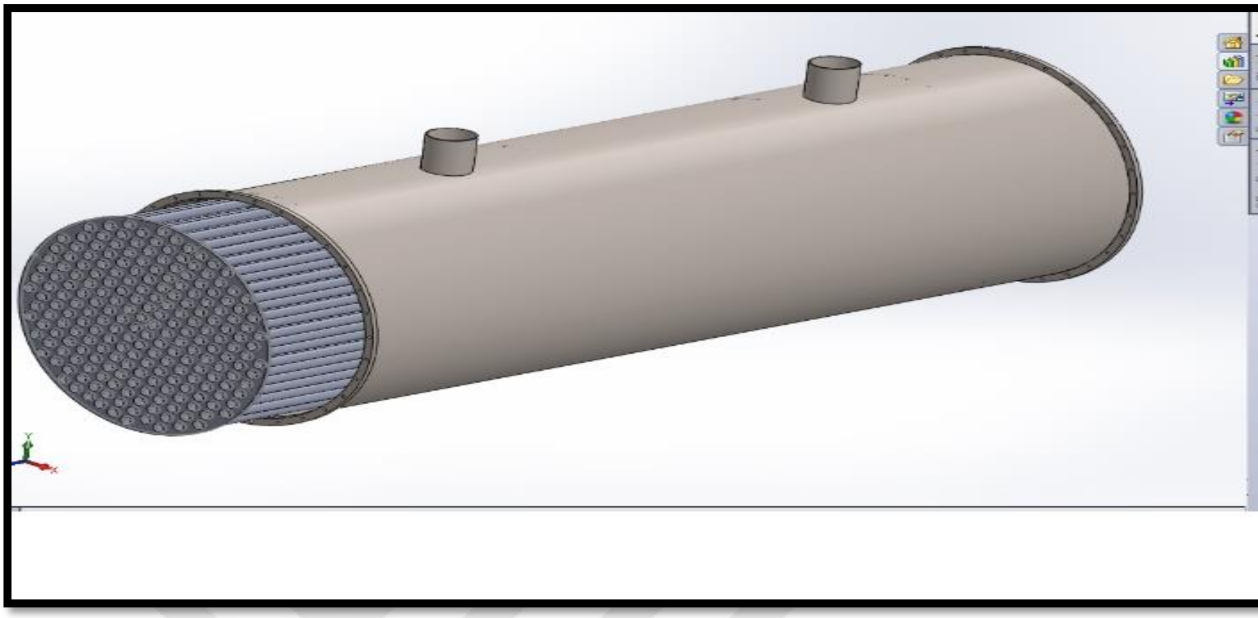

Figure 2.4: Bundle inside shell

The shell that is most commonly used depends on its simplicity and low cost, and has the highest log-mean temperature-difference (LMTD) correction factor. Also, the tubes have single or multiple passes, while the other fluid flows inside the shell over the tubes to be cooled or heated. The fluids of tube and shell sides are separated by the tube sheet shown in Figure 2.5.

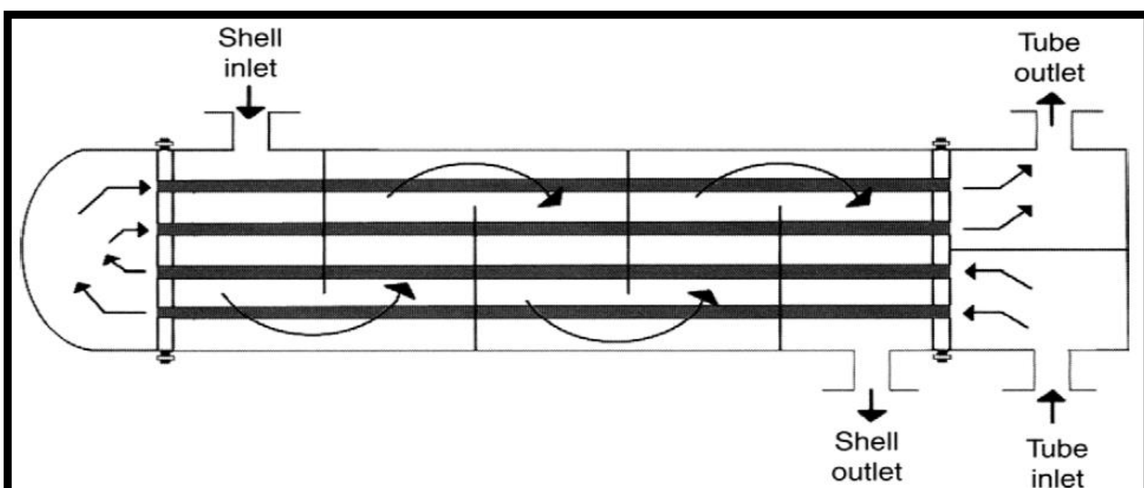

Figure 2.5: A typical Heat Exchanger Shell and Tube, two tube passes _ one shell pass [30]

The baffles are used to support the structural rigidity of the tubes, preventing vibration and bending of tubes and to turn the flow of fluid through the container to get the best of the heat transfer coefficient. Baffle spacing (B) is defined as the center line distance between two nearby baffles. Baffle is provided with a baffle cut (Bc) which is expressed as the percentage of the high sector to a shell's inside diameter. Baffle cut can change between 15% to 45% of the inside diameter of the shell. Generally, conventional shell and tube heat exchanger results in the in-height pressure drop on the shell side and formation of recirculation regions near the baffles.

Most of the reviews nowadays are carried out on helical baffles, which give better performance then single fragment baffles; however, they include a high assembling cost, maintenance and construction costs. The effectiveness and cost are two important parameters in heat exchanger design. So, to enhance the thermal performance at a sensible cost of a shell and tube heat exchanger, baffles in the present review are given some slant to keep up a sensible pressure drop across the heat exchanger [19].

The LMTD technique can be-effectively utilized when the outlet and inlet temperatures of both the cold and hot fluids are known. At the point when the outlet temperatures are not known, the LMTD must be utilized in an iterative plan. For this situation, the effectiveness of the NTU strategy can be utilized to rearrange the analysis. The decision on the type of heat exchanger specifically influences the procedure's performance and impacts plant size, plant design, length of tube runs and the quality and size of supporting structures.

The shell-and tube heat exchanger is the type most usually utilized, the optimal outline of which is the principle aim of this review. These bundles join different design alternatives for the heat exchangers incorporating the varieties in the tube measurement, tube pitch, shell type, number of tube passes, baffle cut percentage, baffles spacing, etc.

An essential aimed at in the Heat Exchanger Design (HED) is the estimation of the base heat exchange range required for a given heat obligation, as it controls the general cost of the HE. In any case, there is no solid target work that can be communicated unequivocally as a component of the outline factors, many quantities of discrete blends of the plan factors are conceivable as is explained underneath. The tube diameter, tube length, shell sorts and so forth are overall established and are accessible only in specific sizes and geometry. Thus, the outline of a shell-and-tube heat exchanger typically includes a trial and error method in which a specific blend of the plan factors into the heat exchange territory and after that, another mix is attempted to check if there is any probability of decreasing the heat exchange area.

The multifaceted nature with experimental methods includes quantitative adjective of stream phenomena utilizing estimations managing one amount at a time for a restricted scope of a problem and working conditions.

2.2 Heat Exchanger Classification

Nowadays heat exchangers can be found in many configurations. Using their applications, fluid flow, process fluids and mode of heat transfer, heat exchangers can be classified [20]. Heat exchangers can transfer heat from side to side with indirect contact by the fluid or via direct techniques. Also, they can classify a heat exchanger based on the passes of shell and tube, types of baffles, arrangement angle of tubes (triangular, square etc.) and smooth or baffled surfaces.

These are also classified via the way flow arrangements work, as fluids can flow in parallel (same direction), counter flow (opposite to each other) and cross flow (normal flow). The choice of a heat exchanger configuration depends on several factors. These factors may involve, the area requirements, maintenance, flow rates, and fluid phase (Figure 2.6 [21]).

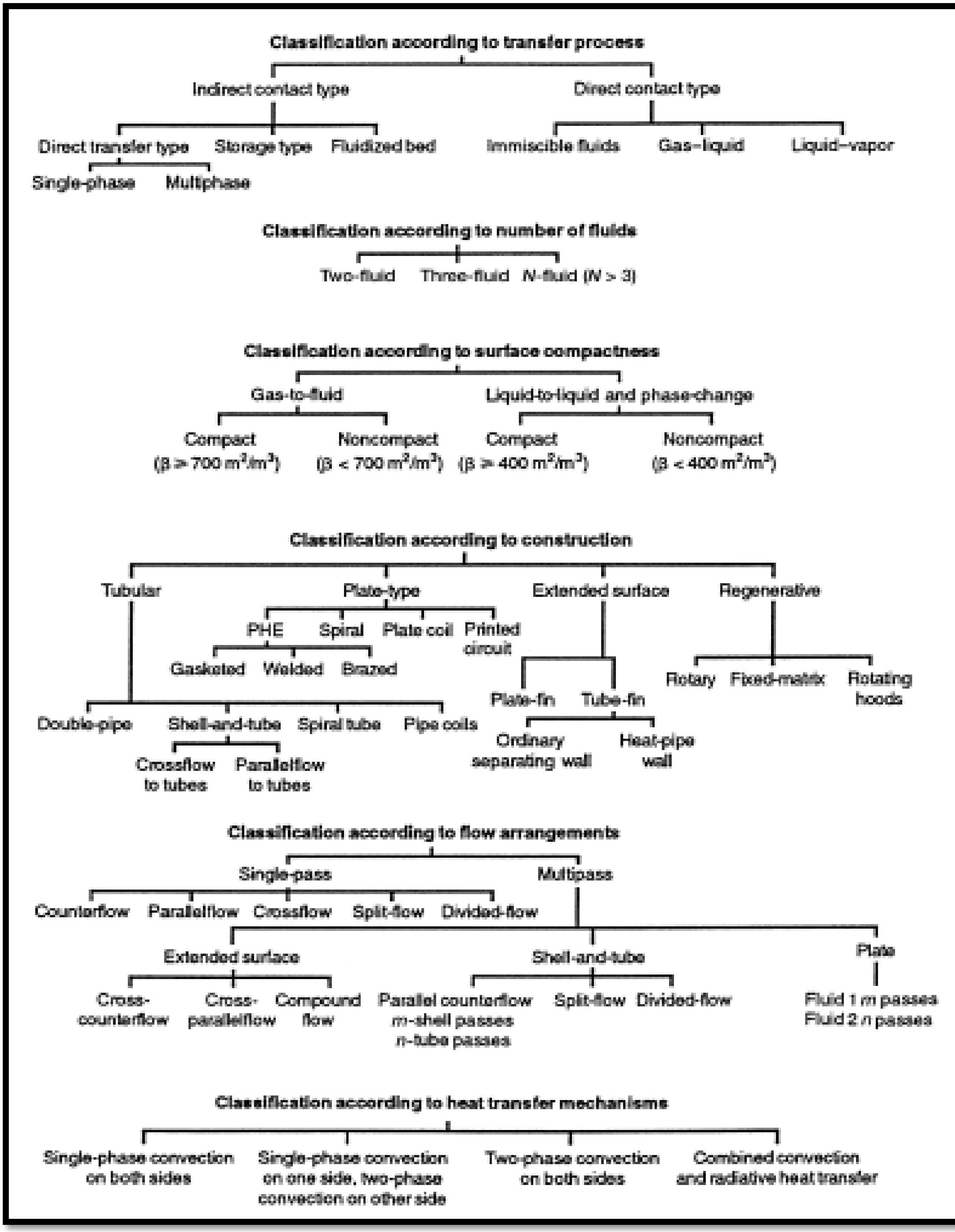

Figure 2.6 Classification of heat exchangers [21]

2.3 Applications of Heat exchangers

Applications of heat exchangers is a very vast topic and would require a separate thorough study to cover each side. Among all applications are their purpose in process industry, mechanical equipment's industry, petroleum industry and home appliances. Heat exchangers can be found working for heating area systems, largely being used nowadays. At the same time, refrigerators and air conditioners settle the heat exchangers to cause the evaporation or condensation of the fluid. In addition, these have also been utilized in milk processing units for pasteurization. In the Table 2.1 w.r.t, different industries explain additional details for the purposes of the heat exchangers [22].

Table 2.1: Heat Exchanger Applications in Different Industries

Industries	Applications
Food and Beverages	Ovens, cookers, pr-heating and Food processing, Milk pasteurization, juices, beer cooling and pasteurization for syrup, or chilling or cooling the final product to desirable temperatures.
Petroleum	Brine cooling, crude oil heating, crude oil heat treatments, Fluid interchange cooling, acid gas condenser.
Hydro carbon processing	Preheating of methanol, liquids hydrocarbon products cooling, feed pre-heaters, Recovery or removal of carbon dioxide, production of ammonia.
Polymer	Production of polymers, Reactor jacket cooling for the production of polyvinyl chlorides.
Pharmaceutical	purgation of water and steam, for point of use cooling on Water for Injection ring.
Automotive	Pickling, Rinsing, Priming, Painting.
Power	Radiators, Cooling circuit, Oil coolers, air conditioners and heaters, energy recovery.
Marine	Marine cooling systems, Fresh water distiller, Diesel fuel

2.4 Tubular Heat Exchanger

2.4.1 Heat Transfer

Heat transfer is the fundamental process of all process industries. During the process of heat transfer, one fluid at higher temperature transfers its energy in the form of heat to the other fluid at a lower temperature. Fluid can transfer its heat through various mechanisms.

These systems of heat transfer are radiation, convection and conduction. Radiation is not a basic method of heat transfer in process industries; however, in a few processes, it plays an imperative role in heat transfer, for example in burning heater. Two other methods of heat transfer, i.e. conduction and convection, are the most common methods of heat transfer in process industries [23] [24]. Overall energy balance of a heat transfer system can be generalized by Equations 2.1 and 2.2.

$$Q_h = m_h C_p \left(T_{h,i} - T_{h,o}\right) \quad (2.1)$$

$$Q_c = m_c C_p \left(T_{c,o} - T_{c,i}\right) \quad (2.2)$$

The actual temperature, even if the fluid heats the liquid at low temperature, is not exactly equal because of the losses and resistance in the form of fouling the wall. The proposition is made that the quantity of heat transferred from the hotter fluid is equal to the quantity of heat transferred to the colder fluid.

Generally, heat exchangers are insulated to decrease the environmental losses. Thus, we can write down equally:

$$Q_h = Q_c = Q$$

A graphical representation of these equations makes the process easier and simpler to understand.

Then we can write the equations in the standard form,

$$\mathrm{Q} = \mathrm{UA}\Delta\mathrm{T}_{\mathrm{LM}} \quad (2.3)$$

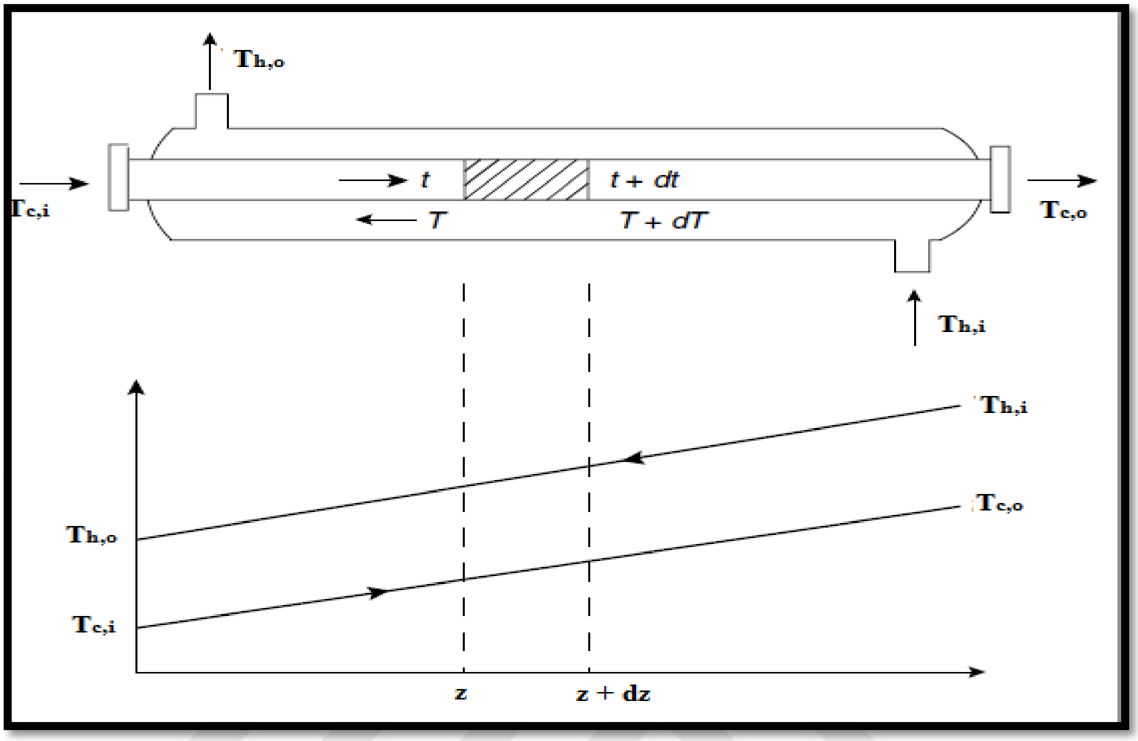

Figure 2.7: Heat Transfer in Heat Exchanger adopted from "Heat Transfer for Process Engineering (2009) "[25].

Where,

Q = Heat transfer rate (W)

A = Heat transfer area (m^2)

U = Overall heat transfer coefficient ($^{W}/_{m^2.K}$)

ΔTLM = Logarithmic mean temperature difference (K)

These three equations (from 2.1 to 2.3) are the fundamental equations for all heat transfer problems. These equations are derived with the help of different assumptions. Mostly, the specific heat capacity and overall heat transfer coefficient are taken into consideration as constant for the heat exchangers. In real training, these amounts can change depending upon the fluids' properties and temperatures. It is applied that specific heat capacity of several industrial fluids such as water, remains constant for an array of temperatures. For example, specific heat capacity of water at 374K and atmospheric pressure = 4226 J /kg. K; specific heat capacity of water at 273.5K and atmospheric pressure =4218 J /kg. Thus, we can say that this assumption works well for such temperature ranges. Specific heat of a fluid is the property of a fluid by which it transfers heat. In other words, it is the quantity of heat needed by one kilogram of fluid to increase its temperature by one degree Celsius. The log mean temperature difference (LMTD) is designed to estimate the average temperature difference during the heat exchange. It is principally the logarithmic average of temperature difference [25].

As for a heat transfer, the driving force is always the temperature difference, thus a higher log mean temperature difference will ensure better heat transfer. It is related to the area of heat exchanger in a way that a higher LMTD will cause less heat transfer area and lower LMTD will need a larger heat transfer area. Generally, LMTD is a process condition and one cannot do much about it as the inlet and outlet temperatures of fluids are usually pre-decided for a heat exchanger design. Area can certainly be reduced by making full use of available LMTD by an efficient heat transfer.

2.4.2 Overall Heat Transfer Coefficient

The overall heat transfer of heat exchangers is the capability of transferring heat across different resistances. It depends upon the properties of the temperatures, process fluids' flow rates and the geometrical arrangement of the heat exchanger. For example, number of baffles, the number of passes and baffle spacing, etc. It is defined in Equation 2.4. This equation basically contains all the resistances encountered during the heat transfer and taking the reciprocal gives us the overall heat transfer coefficient [26].

$$\frac{1}{U} = \frac{1}{h_h} + \frac{\Delta x}{k} + \frac{1}{h_c} + R_f \qquad (2.4)$$

Where:

h_h = Hot side heat transfer coefficient $(W/_{m^2.K})$

h_c = Cold side heat transfer coefficient $(W/_{m^2.K})$

Δx= Exchanger tube wall thickness (m)

k = Exchanger wall material thermal conductivity $(W/_{m.K})$

R_f = Fouling coefficient $(W/_{m^2.K})$

The equation for the overall heat transfer coefficient can be written as in Equation 2.5.

$$\frac{1}{U} = \frac{1}{h_h} + \frac{1}{h_c} + R_f \quad (2.5)$$

h_h and h_c are the individual film coefficients and are defined as the measure of heat transfer for unit area and unit temperature difference. These are calculated separately for both outside and inside fluids. The temperature difference of the average temperatures of bulk fluid (hot and cold) and wall temperature (inside and outside) is the driving force for the respective fluids. Δx=k is usually ignored as it doesn't have a significant effect on the overall heat transfer coefficient. [27].

2.5 Design Guidelines

Thermal design of shell-and-tube heat exchangers is achieved by advanced software programming. Nonetheless, knowledge of the fundamental standards of heat exchanger configuration is expected to utilize this programming adequately. It describes the basics of an exchanger thermal outline, covering such themes as: components of STHE; arrangement of STHEs as per development and administration; information needed for thermal plan; tube side plan; shell side plan, including tube design, baffling, and pressure drop for shell side; and log mean temperature distinction. The fundamental conditions for tube side and shell side thermal exchange and pressure drop are outstanding; here we concentrate on the application of these connections for the perfect design of shell and tube heat exchangers [28].

The propelled points in shell-and-tube heat exchanger plan, for example, assignment of shell side and tube side fluids, utilization of different shells, over design, and fouling, is retained to appear in the next problem.

1.3.1 Components of STHEs

It is imperative for the designer to have a decent working knowledge of the mechanical components of STHEs and how they impact a thermal design. The basic parts of a STHE are:

- Shell.
- Shell cover.
- Tubes.
- Channel.
- Channel cover.

- Tube sheet.
- Baffles.
- Nozzles.

Other segments incorporate tie-poles and spacers, pass parcel plates, impingement plate, longitudinal baffle, sealing strips, backings, and establishment. The Standards of the Tubular Exchanger Manufacturers Association (TEMA) [29] depict these different parts in detail. An STHE is separated into three sections: the shell, the front head, and the back head. Figure 2.8 illustrates the TEMA classification for the different development conceivable outcomes. Exchangers are described by the letter codes for the three segments — for instance, a BFL exchanger has a hood cover, a two-pass shell comes with a longitudinal baffle and a settled tube sheet raise head.

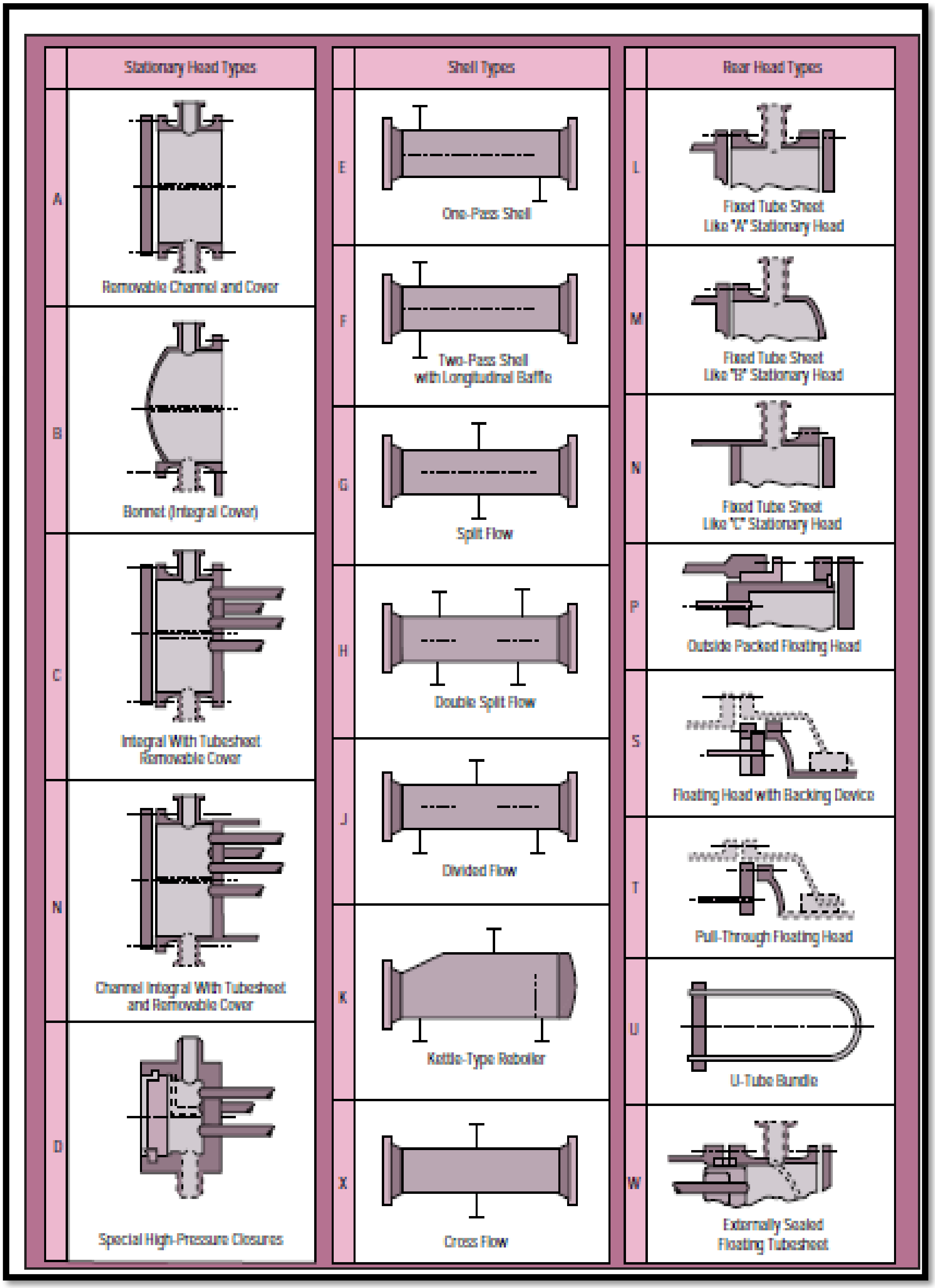

Figure 2.8 Heat exchangers TEMA designations. [28]

2.5.2 Classification

2.5.2.1. Based on construction

Fixed tube sheet: A settled tube sheet heat exchanger (Figure 2.9) has straight tubes that are secured at both ends to tube sheets welded to the shell. The arrangement might have removable channel covers (e.g., AEL), that type of channel covers (e.g., BEM), or necessary tube sheets (e.g., NEN).

The important preferred position of the fixed tube sheet construction is its minimal cost because of its basic construction. Truth to be told, the settled tube sheet is the least expensive arrangement type, as the no measurement of the expansion joint is required. Different preferences are that the tubes can be cleaned mechanically after removal of the channel cover or hood, and that spillage of the shell side fluid is limited since there are no flanged joints.

A disadvantage of this plan is that since the bundle is attached to the shell and can't be removed, the exterior of the tubes can't be cleaned mechanically. Subsequently, its application is limited to cleaning administrations on the shell side.

However, if an attractive chemical cleaning project can be utilized, settled tube sheet construction can be chosen for fouling administrations on the shell side. In case of a substantial difference in temperature between the tubes and the shell, the tube sheets will be unable to handle the differential stresses, thereby making it important to add an extension joint. This takes away the cost of minimal effort to a huge extent.

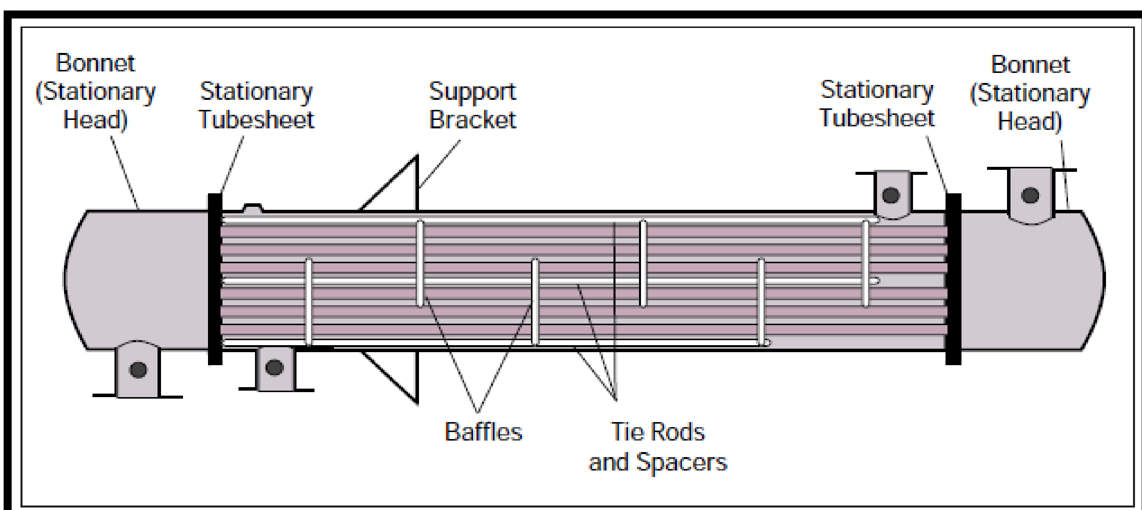

Figure 2.9 Fixed-tube sheet heat exchanger [28]

U-tube: As the name suggests, the containers of a U-tube heat exchanger shown (Figure 2.10) are twisted into the shape of a U. There is just a single tube sheet in a U tube heat exchanger. Nonetheless, the lower cost for the single tube sheet is balanced by the extra expenses triggered for the bowing of the tubes and the somewhat bigger shell measurement (due to the base U-twist range), making the cost of a U-tube heat exchanger tantamount to that of a fixed tube sheet exchanger.

The benefit of a U-tube heat exchanger is that since one end is free, the bundle can expand or contract because of stress differentials. Likewise, the exterior of the tubes can be cleaned, as the tube bundle can be removed.

The disadvantage of the U-tube development is that the internal parts of the tubes can't be cleaned effectively, since the U-curves would require adaptable end penetrating shafts for cleaning. In this way, U-tube heat exchangers should not be utilized for administrations with a grimy fluid inside the tubes.

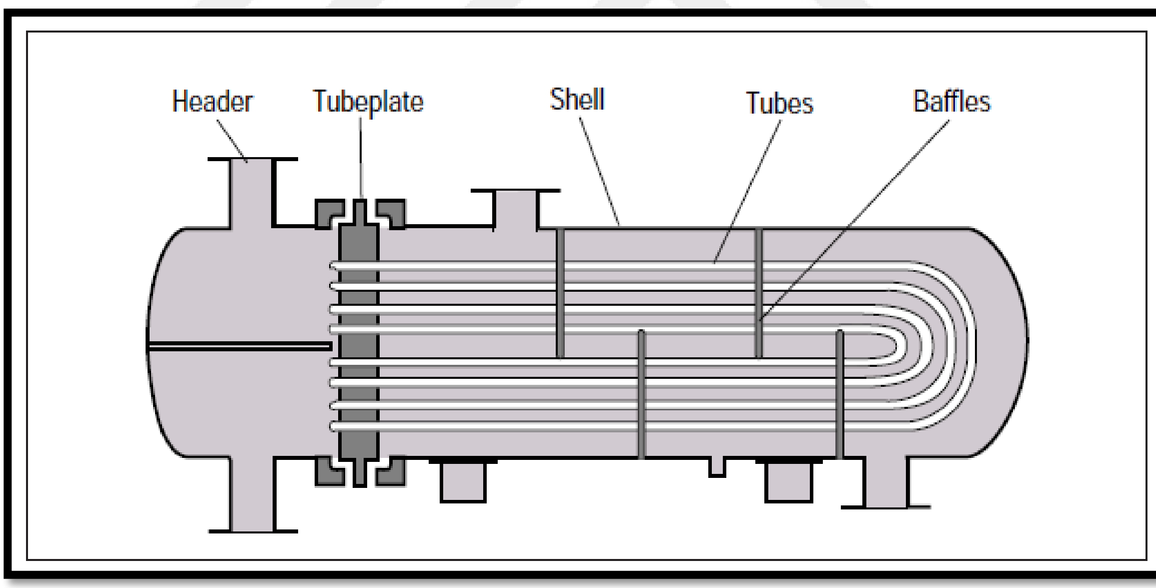

Figure 2.10. U-tube heat exchanger [28]

Floating head: The floating head heat exchanger is the most flexible kind of STHE, and furthermore the costliest. In this plan, one tube sheet is set in respect to the shell, and the other can "float" inside the shell. This allows free development of the tube bundle, and additionally, allows cleaning of both the inner parts and exterior of the tubes.

In this manner, floating head SHTEs can be utilized in cases where both the shell side and the tube side fluids are messy — making this the standard development type utilized as a part of grimy administrations, for example, in petroleum refineries.

There are distinctive kinds of skimming head advancement. The two most typical are the draw through with bolster contraption (TEMA S) and draw through (TEMA T) outlines.

The TEMA S configuration (Figure 2.11) is the most widely recognized design in the chemical procedure industries (CPI). The floating head cover is secured against the gliding tube sheet by shooting it to an astute split support ring. This floating head end is located past the end of the shell and contained by a shell front with a bigger diameter.

To disassemble the heat exchanger, the shell cover is ejected to start with, then the split sponsorship ring, and then the floating head cover, after which the tube bundle can be ejected from the stationary end. In the TEMA T development (Figure 2.12), the whole tube bundle, including the floating head gathering, can be expelled from the stationary end, since the shell width is bigger than the floating-head flange. The floating head cover is attached directly to the floating tube sheet so that a split backing ring is not required.

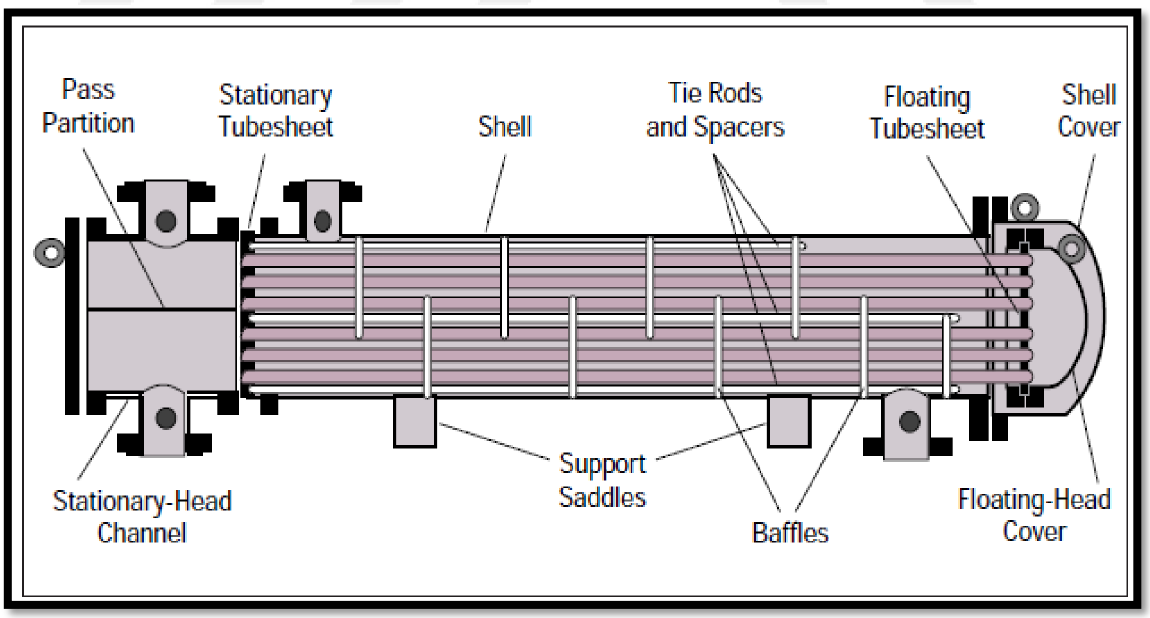

Figure 2.11 Pull-through floating-head exchanger with backing device (TEMAS) [28].

The benefit of this arrangement is that the tube bundle can be ejected from the shell without emptying either the shell or the floating head cover, along these lines lessening support time. This design is especially suited to pot re-boilers having a grimy heating medium where U-tubes cannot be utilized. Because of the enlarged shell, this development is the most expensive of all exchanger types.

There are likewise two types of pressed floating head development — outside packed stuffing-box (TEMA P) and outside-pressed lamp ring (TEMA W) (shown in Figure 2.8). Be that as it may, since they are inclined to leakage, their utilization is restricted to cases with shell side fluids that are nonhazardous and nontoxic and that have direct pressures and temperatures (40 kg/$cm^2$and 300°C).

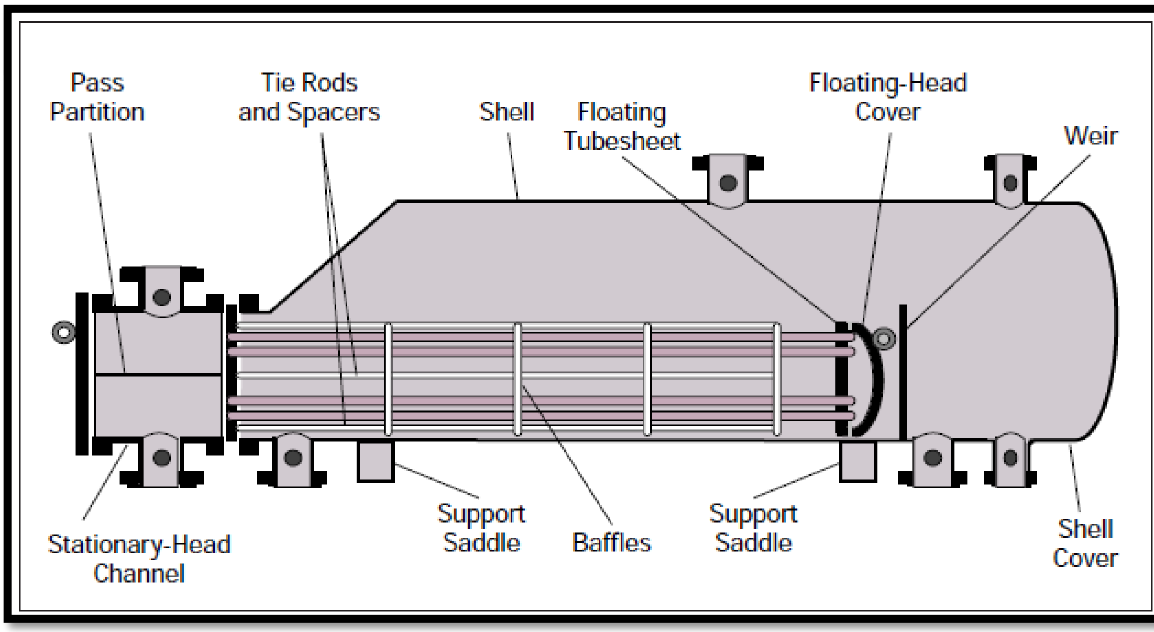

Figure 2.12 Pull-through floating-head exchanger (TEMA T) [28].

2.5.2.2. Based on service

Basically, a case can be single phase, (for example, the cooling or heating of a fluid or gas) or two-phase, (for example, vaporizing or condensing). Since there are two sides to a STHE, this can lead to several mixes of services.

Broadly, examples can be named as follows:

• Single-stage (both shell side and tube side) ;

• Condensing (one side single-phase and the other condensing) ;

• Vaporizing (one side vaporizing and the opposite side single-stage) ;

• Vaporizing/condensing (one side vaporizing and the opposite side condensing).

The accompanying terminology is generally used:

- Heat exchanger: both sides single phase and process streams (that is, not a utility).
- Cooler: one stream is a procedure fluid and the other is air or cooling water.
- Radiator: one stream is a procedure fluid and the other is a hot service, for example, hot oil or steam.
- Condenser: one stream is a condensing vapor and the other is an air or cooling water.
- Chiller: one stream isa handle fluid being condensed at sub-environmental temperatures and the other is a bubbling refrigerant or process stream.
- Re-boiler: one stream is a bottoms stream from a refining section and the other is a hot utility (steam or hot oil) or a procedure stream.

This study will concentrate particularly on single-stage applications

2.5.3 Design Data

Data indicated what should be stable for both the shell side and the tube side. Before examining actual heat exchanger design, let us take a look at the information that must be outfitted by the process licensor before configuration can begin:

1. Stream rates of both streams.
2. Delta temperatures of both streams.
3. Working pressure of both streams. This is required for gasses, particularly if the gas density is not set; it is redundant for fluids, as their properties don't differ with pressure.
4. Reasonable pressure drop for both streams. This is imperative vital parameter for heat exchanger design.

   Generally, for fluids, an estimation of 0.5–0.7 kg/cm^2 is allowed per shell. A higher-pressure drop is generally justified for gooey fluids, particularly in the tube side. For gasses, the permitted esteem is by and large 0.05–0.2 kg/cm^2, with 0.1 kg/cm^2being typical.
5. Fouling resistance for both streams. On the off chance that this is not set, the designer should receive values specified in the TEMA norms or considering experience.

6. Physical properties of both streams. These involve viscosity, thermal conductivity, thickness, and specific heat, ideally at both inlet and outlet temperatures. Viscosity information must be provided at inlet and outlet temperatures, particularly for liquids, since the variety in temperature can be considerable and is sporadic (neither straight nor log-log).

7. Heat duty. The duty specified should be consistent for both the shell and tube side.

8. Kind of heat exchanger. If not outfitted, the designer can pick this based upon the characteristics of the different kinds of development portrayed before. Indeed, the creator is normally in a better position than the procedure architect to do this.

9. Line sizes. It is important to match spout sizes with line sizes to keep away from expanders or reducers. However, estimating criteria for spouts are typically more stringent than for lines, particularly for the shell side inlet. Therefore, spout sizes should at times be one size (or much more in exceptional conditions) bigger than the comparable line sizes, particularly for little lines.

10. Favored tube measure. Tube size is assigned as O.D_thickness_ length. Some plant managers have a preferred O.D_ thickness (generally based upon stock contemplation), and the available design area will decide the largest tube length.

    Many plant managers like to keep every one of the three measurements on hand, again based upon stock considerations.

11. Maximum shell diameter. This is based upon tube bundle evacuation necessities and is restricted by crane limits. Such restrictions apply only to exchangers with removable tube bundles, namely U-tube and floating head.

    For settled tube sheet exchangers, the main constraint is the producer's manufacture capacity and the accessibility of segments, for example, dished finishes and spines. Along these lines, floating head heat exchangers are frequently constrained to a shell I.D. of 1.4–1.5 m and a tube length of 6 m or 9 m, though fixed tube sheet heat exchangers can have shells as expansive as 3 m and tube lengths up to 12 m or more.

12. Materials of development. On the off chance that the tubes and shell are made of indistinguishable materials, all components should be of this material.

    Thus, only the shell and tube materials of development should be indicated. Be that as it may, if the shell and tubes are of various metallurgy, the materials of all essential parts should be specified to maintain a strategic distance from any vagueness.

    The vital parts are shell (and shell cover), tubes, channel (and channel cover), tube sheets, and baffles. Tube sheets can be lined or clad.

13. Special considerations. These incorporate cycling, agitate conditions, elective working scenarios, and whether operation is consistent or intermittent.

2.5.4 Tube Side Design

Tube side computations are very direct, since tube side stream signifies a straight forward instance of flow through a round conductor. Pressure drop and heat-transfer coefficients both change with tube side speed, the last more strongly so.

A decent design will make the best utilization of the permissible pressure drop, as this will yield the highest heat transfer coefficient.

On the off chance that all the tube side fluid was to course through every one of the tubes (one tube pass), it would lead to a certain velocity.

Generally, this speed is inadmissibly low and hence must be increased. By consolidating pass segment plates (with fitting casketing) in the channels, the tube side fluid is made to travel a few passes through a small amount of the aggregate number of tubes. Accordingly, in a heat exchanger with 200 tubes and two passes, the liquid moves through 100 tubes at any given moment, and the velocity will be twice what it would be if there were just a single pass. The quantity of tube passes is usually one, two, four, six, eight, etc.

2.5.4.1. Heat-transfer Coefficient

The tube side heat-transfer coefficients are components of the Prandtl number, Reynolds Number, and the tube diameter. These can be separated into the accompanying basic parameters: physical properties (in viscosity, specific heat and thermal conductivity; tube diameter; furthermore, significantly, mass speed). The variety in fluid consistency is very extensive; along these lines, this physical property has the most sensational impact on heat-transfer coefficient. The central equation for turbulent heat exchange inside tubes is:

$$\mathrm{Nu} = 0.027(R_e)^{0.8}(\,P_r)^{0.33} \tag{2.6}$$

or

$$\left(\frac{hD}{K}\right) = 0.027\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{C\mu}{K}\right)^{0.33} \tag{2.7}$$

Rearranging:

$$h = 0.027\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{C\mu}{K}\right)^{0.33}\left(\frac{K}{D}\right) \tag{2.8}$$

Viscosity impacts the heat-transfer coefficient in two contradicting routes — as a parameter of the Prandtl number, and as a parameter of Reynolds number. Along these lines, from Eq. (2.7):

$h\alpha(\mu)^{0.33-0.8}$ (2a)

$h\alpha(\mu)^{-0.47}$ (2b)

As such, the heat-transfer coefficient is contrarily corresponding to viscosity to the 0.47 power. So, the heat-transfer coefficient is specifically corresponding to thermal conductivity to the 0.67 power. These two truths lead to some fascinating sweeping statements about heat transfer.

A high thermal conductivity leads to a high heat transfer coefficient. In this manner, cooling water (thermal conductivity of around 0.55$(\frac{kcal}{h.m.c^{\circ}})$ has to a great degree high heat-transfer coefficient of regularly 6,000$(\frac{kcal}{h.m^2.c^{\circ}})$, trailed by hydrocarbon fluids (thermal conductivity in the between 0.08 and 0.12 $(\frac{kcal}{h.m.c^{\circ}})$ at 250–1,300 $(\frac{kcal}{h.m^2.c^{\circ}})$,, and after that hydrocarbon gasses (thermal conductivity between 0.02 and 0.03 $(\frac{kcal}{h.m.c^{\circ}})$ at 50–500 $(\frac{kcal}{h.m^2.c^{\circ}})$,,. Hydrogen is an uncommon gas, since it has a particularly high thermal conductivity (more prominent than that of hydrocarbon fluids). Therefore, its heat transfer coefficient is near the furthest reaches of the range for hydrocarbon liquids.

The scope of heat transfer coefficient for hydrocarbon fluids is somewhat expansive because of the vast variation in their viscosity, from under 0.1c_p for ethylene and propylene to more than 1,000 c_p or more for bitumen. The huge variety in the heat transfer coefficient of hydrocarbon gasses is attributable to the extensive variety in working pressure. As working pressure rises, gas density increases. Pressure drop is straight forwardly corresponding to the square of mass speed and contrarily relative to density. In this way, for a similar pressure drop, a higher mass speed can be kept up when the density is higher. This higher mass speed converts into a higher heat transfer coefficient.

2.5.4.2 Pressure Drop

Mass speed unequivocally impacts the heat-transfer coefficient. For turbulent streams, the tube side heat-transfer coefficient shifts to the 0.8 force of tube side mass speed, while tube side pressure drop differs to the square of mass B speed. Along these lines, with increasing mass speed, pressure drop builds more quickly than does the heat-transfer coefficient [30]. So, there will be an ideal mass speed above which it will be inefficient to increase mass speed further.

Furthermore, high speeds lead to disintegration. Be that as it may, the pressure drop weakness generally can be controlled much sooner than erosive speeds occur. The base suggested liquid speed inside tubes is 1.0 m/s, while the greatest speed is 2.5–3.0 m/s. Pressure drop is proportional to the square of speed and the aggregate length of travel.

In this manner, when the quantity of tube passes is expanded for a given number of tubes and a given tube side stream rate, the pressure drop rises to the 3D square of this expansion. In genuine practice, the rise is to some degree less due to lower friction factors at higher Reynolds numbers, so the example should be roughly 2.8 instead of 3. Tube side pressure drop rises steeply with an expansion in the number of tube passes.

Consequently, it regularly happens that for a given number of tubes and two passes, the pressure drop is much lower than reasonably expected, yet with four passes it surpasses the appropriate pressure drop. During such conditions, a standard tube must be used, and the designer can be compelled to accept a lower speed. However, if the tube measurement and length are shifted, the reasonable pressure drop can be better determined and a higher tube side velocity realized.

The accompanying tube diameters are generally utilized as a part of the CPI:$\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$,$\frac{3}{4}$,1, $1\frac{1}{4}$, and $1\frac{1}{5}$ in. Of these, $\frac{3}{4}$in. as well as, 1 in. are the most well-known. Tubes smaller than $\frac{3}{4}$ in. O.D. should not be utilized for fouling services. The use of small diameter tubes, for example, $\frac{1}{2}$in., is justified only for small heat exchangers with heat-transfer ranges under 20–30 m^2.

One should understand that the aggregate pressure drop for a given stream must be met. The circulation of pressure drop in the different heat exchangers for a given stream in a circuit can fluctuate to acquire greater heat transfer in all the heat exchangers. A hot fluid stream coursing through a few preheat exchangers should be considered.

Normally, a pressure drop of 0.7 kg/cm^2 per shell is allowed for fluid streams. On the off chance that there are five such preheat exchangers; an aggregate pressure drop of 3.5 kg/cm^2 for the circuit would be permitted. If the pressure drops through two of these exchangers end up being just 0.8 kg/cm^2, an adjustment of 2.7 kg/cm^2 would be accessible for the other three.

2.5.5 Shell Side Design

The shell side computations are significantly more complex than those for the tube side. This is fundamentally because on the shell side there is not only one stream, but rather one main cross stream and four leakage or sidestep streams. There are different shell side stream arrangements, and in addition, different tube format designs and baffling outlines, which together decide the shell side stream analysis.

2.5.5.1 Shell Setup

TEMA characterizes different shell designs in view of the stream of the shell side fluid through the shell: E, F, G, H, J, K, and X See Figure 2.8.

In a TEMA E single-pass shell, the shell side fluid enters the shell at one end and leaves from the flip side. This is the most well-known shell types — more heat exchangers are operated in this setup than every other arrangement combined.

A TEMA F two-pass shell has a longitudinal baffle that partitions the shell into two passes. The shell side fluid enters toward one side, navigates the whole length of the exchanger through one-half the shell cross-sectional zone, turns around and flows during that time pass, then finally leaves at the end of the second pass. The longitudinal baffle stops well short of the tube sheet, so that the fluid can flow into the second pass. The F shell is utilized for temperature-cross circumstances — that is, the place the cool stream leaves at a temperature higher than the outlet temperature of the hot stream. If a two-pass (F) shell has just two tube passes, this turns into a genuine counter current arrangement where an expansive temperature cross can be achieved.

A TEMA J shell is an isolated stream shell where in the shell side fluid enters the shell at the center and partitions into two parts, one flowing to one side and the other to one side and leaving separately. They are then combined into a single stream. This is recognized as a J 1–2 shell. Alternatively, the stream can separate into two parts that enter the shell at the two closures, flow toward the inside, and leave as a solitary stream, which is distinguished as a J 2–1 shell.

A TEMA G shell is a part stream shell see Figure 2.8. This development is typically utilized for horizontal thermosiphon re-boilers. There is just a focal bolster plate and it has no baffles. A G shell cannot be utilized for heat exchangers with tube lengths larger than 3 m, since this would surpass the constraint on the most extreme unsupported tube length determined by TEMA — commonly 1.5 m, however, it varies with tube O.D., thickness, and material.

When a bigger tube length is required, a TEMA H shell see Figure 2.8 is utilized. A H shell is basically two G shells set next to each other, so that there are two full bolster plates. This is described as a twofold part setup, as the stream is separated twice and recombined twice. This construction, as well, is perpetually utilized for flat thermosiphon re-boilers. The benefit of G and H shells is that the pressure drop is radically less and there are no cross baffles.

A TEMA X shell see Figure 2.8 is a pure cross-stream shell where the shell side liquid enters at the top (or base) of the shell, flows over the tubes, and exits from the inverse side of the shell. The stream can be presented through numerous spouts found deliberately along the length of the shell with a specific end goal to accomplish a superior dispersion. The pressure drop will be to a great degree low — actually, there is not really any pressure drop in the shell, and what pressure drop there is, is for all intents and purposes in the nozzles.

Thus, this setup is utilized for cooling or condensing vapors at low pressure, particularly in a vacuum. Full bolster plates can be found if necessary for basic trustworthiness; they do not interfere with the shell side stream since they are parallel to the stream's direction.

A TEMA K shell (see Figure 2.8) is a unique cross-stream shell utilized for pot re-boilers (in this manner the K). It has a vital vapor-withdrawal space typified in an augmented shell. Here, as well, full bolster plates can be utilized as required.

2.5.5.2 Tube Layout Patterns

There are four tube designs, as shown in Figure 2.13: triangular (30°), turned triangular (60°), square (90°), and pivoted square (45°).

A triangular (or turned triangular) example will suit a greater number of tubes than a square (or rotated square) design. Moreover, a triangular example delivers high turbulence and hence a high heat-transfer coefficient. Notwithstanding, at the run of the model pitch of 1.25 times the tube O.D. does not allow automatic cleaning of tubes, since the paths are not accessible. Thus, a triangular layout is restricted to clean shell side administrations. For administrations that require mechanical cleaning on the shell side, a square design must be utilized. Chemical cleaning does not require access paths, so a triangular format can be utilized for dirty shell side administrations provided synthetic cleaning is appropriate and effective.

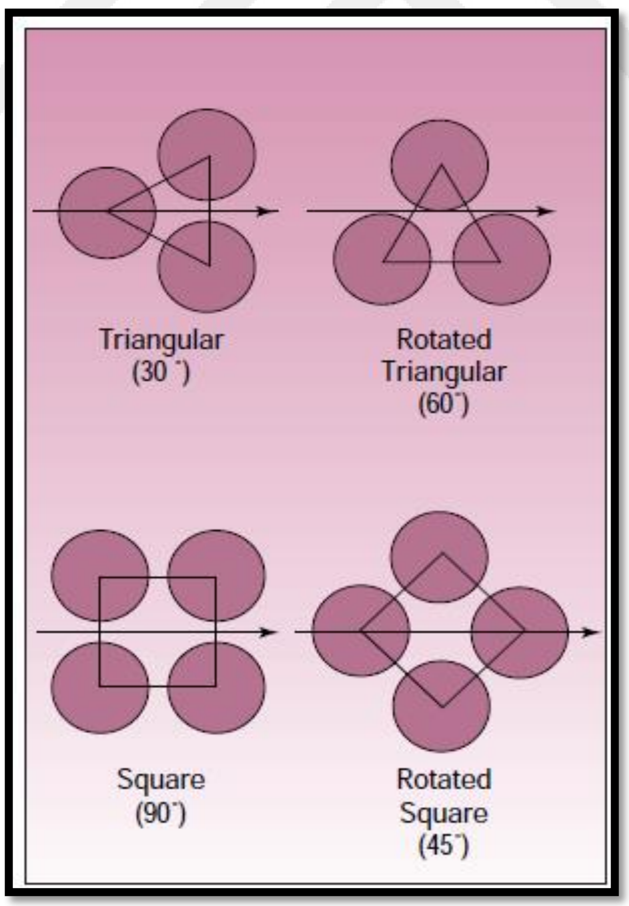

Figure 2.13. Tube layout patterns [28]

A pivoted triangular example at times offers any points of interest over a triangular design, and its utilization is consequently not mainstream. For unclean shell side administrations, a square design is commonly utilized. However, since this is an in-line design, it brings down turbulence. Thus, when the shell side Reynolds number is low (< 2,000), it is generally profitable to utilize a pivoted square design because this produces much higher turbulence, which brings about a higher productivity of transformation of pressure drop to heat transfer.

As noted before, settled tube sheet development is generally utilized for clean administrations on the shell side, U tube development for clean administrations on the tube side, and floating head development for unclean administrations on both the shell side and tube side. (For clean administrations on both shell side and tube side, either a settled tube sheet or U-tube development can be utilized, although a U-tube is preferable since it grants differential extension between the shell and the tubes).

Hence, a triangular tube design can be utilized for settled tube sheet exchangers and a square (or pivoted square) design for floating head exchangers. For U-tube exchangers, a triangular example can be used provided the shell side stream is perfect and a square (or pivoted square) design on the off chance that it is dirty.

2.5.5.3 Tube Pitch

Tube pitch is characterized as the briefest separation between two adjoining tubes (Figure 2.14). For a triangular layout, TEMA determines a base tube pitch of 1.25 times the tube O.D. Therefore, a 25-mm tube pitch is generally utilized for 20-mm O.D. tubes.

For square layout, TEMA furthermore suggests a base cleaning path of 4 in. (then again 6 mm) between adjoining tubes. Subsequently, the base tube pitch for square layout is either 1.25 times the tube O.D. or the tube O.D. itself in addition to 6 mm, whichever is bigger. For instance, 20-mm tubes should be laid on a 26-mm (20 mm + 6 mm) square pitch, yet 25-mm tubes should be laid on a 31.25-mm (25 mm x1.25) square pitch.

Designers want to utilize the base prescribed tube pitch, since it leads to the smallest shell measurement for a given number of tubes. Nonetheless, in excellent conditions, the tube pitch can be expanded to a higher value, for instance, to diminish shell side pressure drop. This is especially valid because of a cross-stream shell.

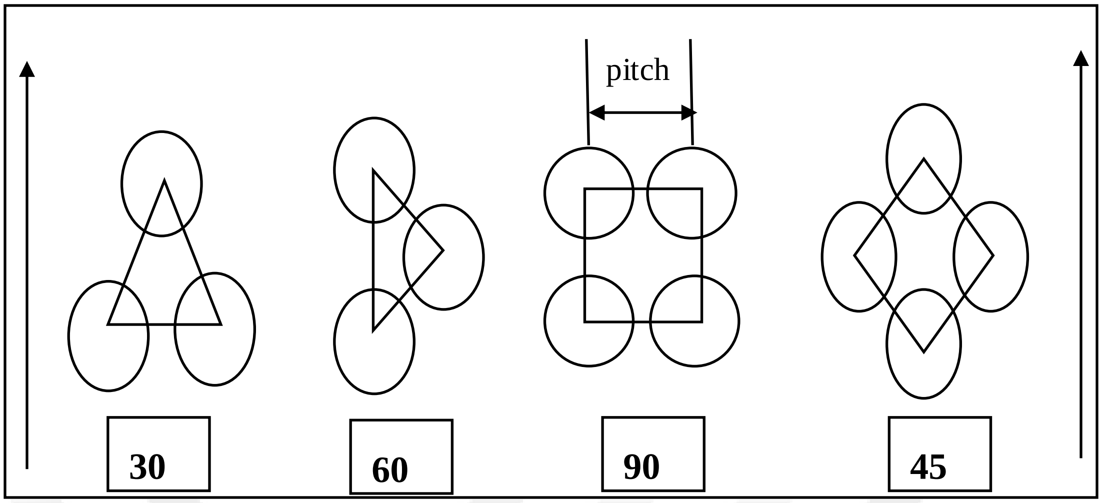

Figure 2.14. Tube pitch (triangular two types and square two types)

2.5.6 Baffling

Type of baffles: Confounds are utilized to bolster the tubes, permit an attractive speed to be kept up for the shell side fluid, and avoid distress of tubes because of stream instigated vibration. There are two kinds of baffles: plate and rod.

Plate baffles can be single-segmental, twofold segmental, or triple-segmental, as appears in Figure 2.15.

Baffle spacing: Baffle dividing is the center line-to-center line separation between adjoining shocks. It is the most vital parameter in STHE design.

The TEMA principles indicate the base shock separating as one-fifth of the shell inside diameter or 2 in., whichever is more notable. Nearer separating will bring about poor bundle infiltration by the shell side fluid and trouble in mechanical cleaning of the exterior of the tubes. Besides, a low confuse separating brings about a poor stream transmission as will be clarified later.

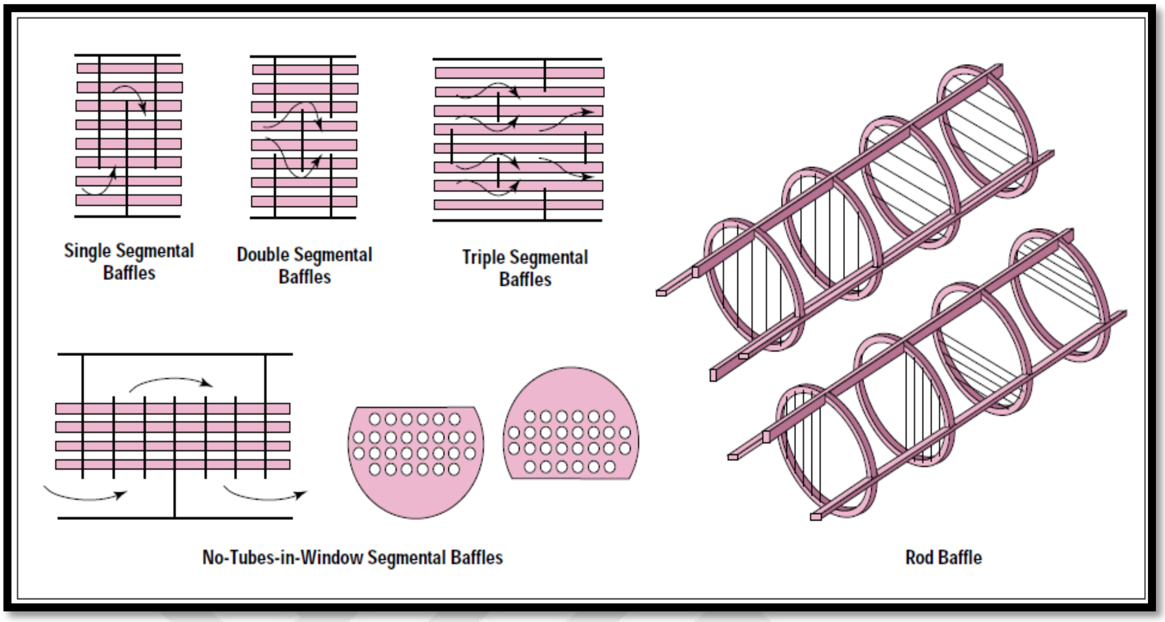

Figure 2.15. Types of baffles [28]

The greatest baffle separation is the shell inside distance across. Higher baffle separation will lead predominantly to a longitudinal stream, which is less productive than a cross-stream, and substantial unsupported tube ranges, which will make the exchanger failure to tube distress because of stream incited vibration. In optimum baffle separation for turbulent flow on the shell side (Re > 1,000), the heat-transfer coefficient differs to the 0.6–0.7 force of velocity; notwithstanding, pressure drop fluctuates to the 1.7–2.0 power. For laminar stream (Re < 100), the exponents are 0.33 for the heat-transfer coefficient and 1.0 for pressure drop. In this manner, as baffle spacing is reduced, pressure drop increments occur at a much quicker rate than does the heat-transfer coefficient.

This implies there will be an ideal proportion of baffle separation of shell inside distance across that will result in the best productivity of transformation of pressure drop to heat transfer. This ideal proportion is ordinarily in the vicinity of 0.3 and 0.6.

Baffle cut: As appears in Figure 2.16, baffle cut is the stature of the fragment that is sliced in each baffle to allow the shell side fluid to flow over the baffle.

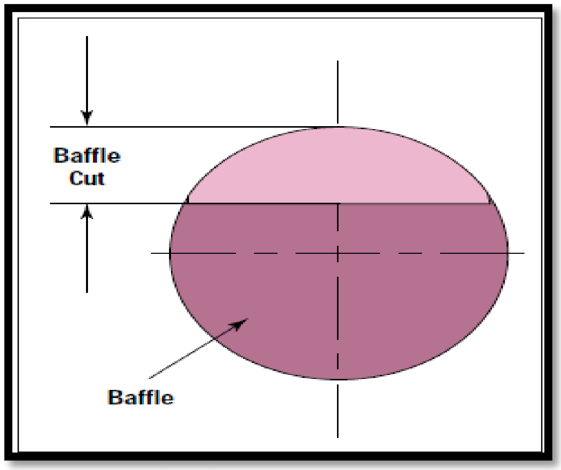

Figure. 2.16 Baffle cut [28]

This is communicated as a rate of the shell inside diameter. Even though this, as well, is an essential parameter for STHE outline, its impact is less significant than that of baffle separation. Baffle cut can vary between 15% and 45% of the shell inside distance across. Both small and large baffle cuts are inconvenient for proficient heat transfer on the shell side because of expansive deviation from a perfect condition, as defined in Figure 2.17. It is unequivocally suggested that single baffle cuts between 20% and 35% be employed. Decreasing baffle cuts under 20% to build the shell side heat transfer coefficient or increasing the baffle cuts past 35% to diminish the shell side pressure drop as a rule leads to poor designs.

Different parts of tube bundle geometry should be changed instead to accomplish those goals. For instance, double segmental baffle or a partitioned stream shell, or even a cross-stream shell, may be utilized to diminish the shell side pressure drop. For single-stage fluids on the shell side, a level baffle cut as shown (Figure 2.18) is suggested, because this limits gathering of stores at the base of the shell and furthermore counteracts stratification. Be that as it may, because a two-pass shell (TEMA F), a vertical cut is favored for the ease of creation and package assembly. Shock is examined in closer detail in [31] and [32].

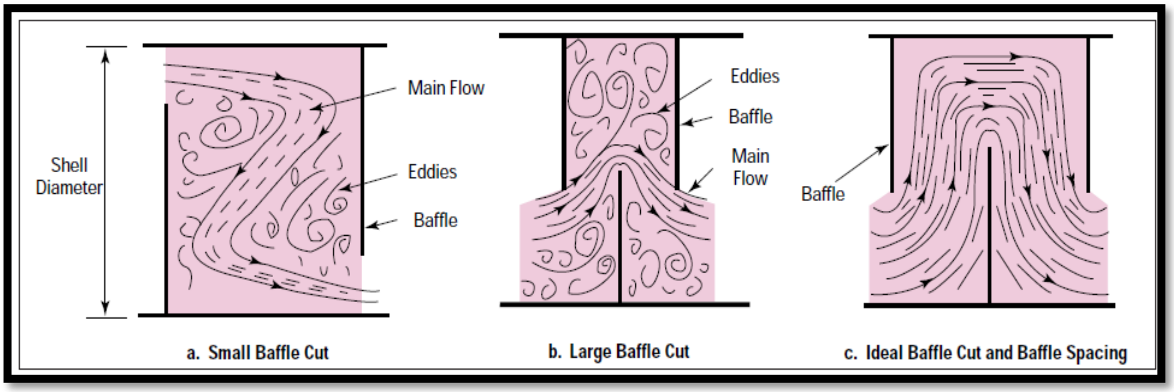

Figure 2.17. Effect of small and large baffle cuts [28]

2.5.7 Equalize Cross-flow and Window Velocities

The stream crosswise over tubes is referred to as cross-stream, while flow through the window zone (that is, through the baffle cut region) is referred to as window flow.

The window speed and the cross-stream speed should be as close as conceivable — ideally within 20% of each other. If they vary by more than that, rehashed speeding up and deceleration occur along the length of the tube bundle, bringing about wasteful transformation of pressure drop to heat transfer.

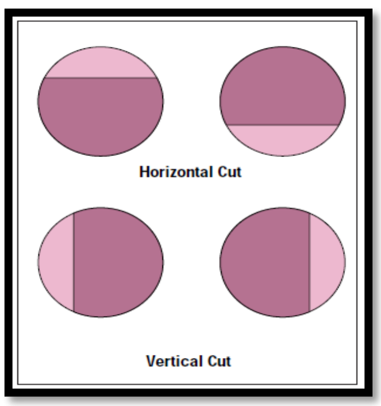

Figure 2.18. Baffle cut orientation [28]

2.5.8 Shell Side Stream Analysis

On the shell side, there is not only one stream, but rather a principle cross stream and four leakage or bypass streams, as outlined in Figure 2.19. Tinker [6] proposed calling these streams the fundamental cross stream (B), a tube-to-baffle gap leakage stream (An), a bundle sidestep stream (C), a pass-segment sidestep stream (F), and a baffle to-shell leakage stream (E). However, the B (fundamental cross-stream) stream is exceptionally powerful for heat.

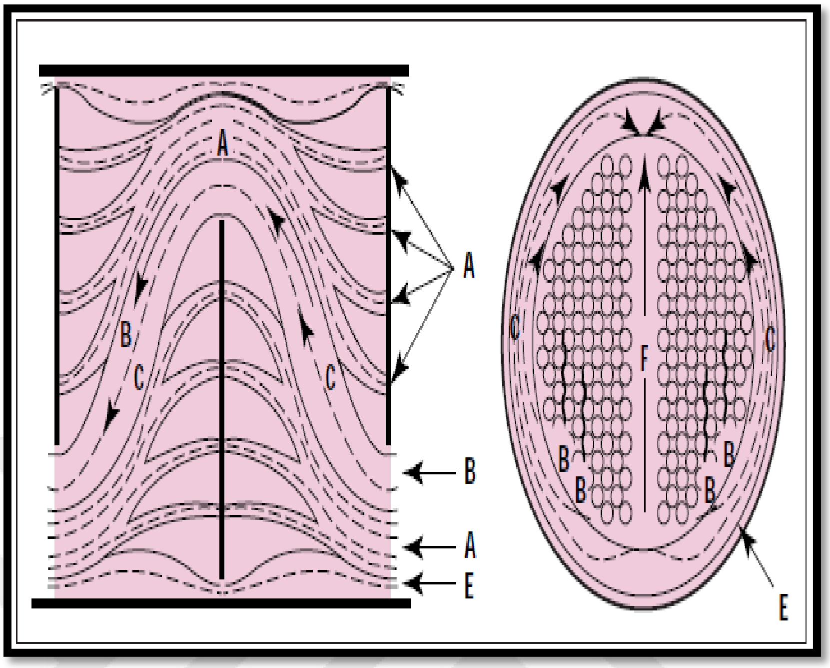

Figure 2.19. Shell side flow distribution [28]

Transferring alternate streams is not as compelling. The A stream is genuinely proficient, because the shell side fluid is in contact with the tubes. Also, the C stream is in contact with the fringe tubes around the bundle, and the F stream is in contact with the tubes along the pass-segment paths. Thus, these streams likewise encounter heat transfer, although at a lower productivity than the B stream.

In any case, since the E stream travels along the shell divider, where there are no tubes, it experiences no heat transfer. The portions of the aggregate stream represented by these five streams can be resolved for a specific arrangement of exchanger geometry and shell side stream conditions by any refined heat exchanger thermal design software.

Basically, the five streams are in parallel and travel along the paths of fluctuating water powered resistances. Along these lines, the stream divisions will be to such an extent that the pressure drop of each stream is indistinguishable, since every one of the streams start and end at the delta and outlet spouts. In this way, based upon the proficiency of each of these streams, the general shell side stream effectiveness and thus the shell side heat-transfer coefficient is set up. Since the stream portions depend emphatically upon the path resistances, varying any of the accompanying development parameters will influence stream examination and along these lines, the shell side execution of an exchanger:

• Baffle spacing and baffle cut;
• Tube layout angle and tube pitch;
• Number of paths in the stream course and path width;
• Clearance between the tube and the baffle hole;
• Clearance between the shell I.D. and the baffle;
• Area of fixing strips and fixing rods.

Using a low baffle spacing tends to expand the spillage and sidestep streams. This is because each of the five shell side streams are in parallel and, in this way, have a similar pressure drop. The leakage path measurements are settled. Thus, when baffle spacing is diminished, the resistance of the primary cross-stream way and in this way its pressure drop increases. Since the pressure drops of every one of the five streams must be equivalent, the spillage and sidestep streams rise until the pressure drops of the considerable number of streams adjust.

The net result is a rise in the pressure drop without a comparing increment in the heat-transfer coefficient. The shell side fluid viscosity additionally influences stream examination profoundly. Notwithstanding impacting the shell side heat transfer and pressure reperformance, the stream investigation likewise influences the mean temperature distinction (MTD) of the exchanger. This will be discussed in detail later. However, in the following chapter a case that exhibits how to streamline baffle outline when there is no huge temperature profile distortion will be examined.

CHAPTER 3

MATERIALS AND METHODS

The heat transfer investigation analysis of a heat exchanger (shell and tube) includes the figuring of the overall heat transfer- coefficient from the individual film coefficients [32]. The shell-side coefficient displays many troubles because of an excessive amount of complex nature of the flow in the shell side. Likewise, the heat exchangers can be outlined by the LMTD at the point when inlet and outlet, conditions are specified.

In the present review inlet temperature of shell and tube sides are taken as information parameters with a given bundle arrangement of square pitch. The thermal analysis is done taking crude oil inside the tube and crude oil products on the shell side. The outline of shell and tube exchanger utilizing Kern strategy for crude oil and unrefined petroleum items is approved by the well-known Dittus-Boelter condition of turbulent stream inside a tube. Parameters, for example, heat transfer coefficient, friction coefficient, length, zone and pressure drop are resolved. "MATLAB R2014a" Program has been developed to assess the parameters above. Tables are to depict the behavior for various fluid combinations such as the heat transfer analysis of heat exchangers.

3.1 Design Procedure of Shell and Tube Heat Exchanger

3.1.1 Stepwise Procedure for Calculation

A heat exchanger can be designed by the LMTD when inlet and outlet conditions are specified.

When the problem is to determine the inlet and outlet temperatures for a heat exchanger, the analysis is performed more easily by using a method based on effectiveness of the heat exchanger and number of transfer units (NTU).The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$€ = \frac{\text{actual heat transfer}}{\text{maximume pssiple heat transfer}} = \frac{Q}{Q\text{max}} \tag{3.1}$$

The actual heat transfer rate Q can be determined by energy balance equation,

$$Q = \dot{m_h} c_{ph}(t_{h1} - t_{h2}) = \dot{m_c} c_{pc}(t_{c2} - t_{c1}) \tag{3.2}$$

The fluid capacity rate C:

$\dot{m_h} c_{ph} = c_h = \text{hot fluid capacity rate}$

$\dot{m_c} c_{pc} = c_c = \text{cold fluid capacity rate}$

$c_{min} = \text{the minimum fluid capacity rate}(c_h \text{ or } c_c)$

$c_{max} = \text{the maximum fluid capacity rate}(c_h \text{ or } c_c)$

The number of transfer units (NTU) = $\frac{UA}{C_{min}}$ (3.3)

Where;

U = overall heat transfer coefficient in W/m^2. K

A = surface area in m^2

The effectiveness $= \frac{c_h(t_{hi} - t_{ho})}{c_{min}(t_{hi} - t_{ci})} = \frac{c_c(t_{co} - t_{ci})}{c_{min}(t_{hi} - t_{ci})}$ (3.4)

The governing equations for design problem are usually given as follows:

Heat rate

$$Q = C_h(T_{hi} - T_{ho}) = C_c(T_{ci} - T_{co}) \tag{3.5}$$

Where:

Q= heat duty of heat exchanger, W

Ch= specific heat of the hot fluid, J/kgK

Cc = specific heat of the cold fluid, J/kgK

Thi= temperature of the hot fluid inside, K

Tho= temperature of the hot fluid outside, K

Tci= temperature of the cold fluid inside, K

Tco= temperature of the cold fluid outside, K

Where heat capacity rate for hot or cold fluid

$C = \dot{m} C_p$
Where;

$\dot{m}$= mass flow rate, kg/sec

C= heat capacity

Log mean temperature difference for pure counter flow

$$\Delta T_{lm,cf} = \frac{(T_{hi}-T_{co})-(T_{ho}-T_{ci})}{\ln\left[{(T_{hi}-T_{co})}/{(T_{ho}-T_{ci})}\right]} \tag{3.6}$$

The effective mean temperature difference for cross flow

$$\Delta T_m = F\Delta T_{lm,cf}$$

Where F=correction factor

Shell-side area is calculated by: $A_s = \frac{D_s BC}{P_t}$ (3.7)

Where;

Ds=shell diameter, m

C= clearance, m

B= baffle spacing, m

$$G_s = \frac{\dot{m}_s}{A_s} \tag{3.8}$$

Where, $\dot{m}_s$ = mass flow rate of shell side, kg/s

As= area of the shell, m^2

$$D_e = 4\frac{\left(P_t^2 - \pi\frac{D_0^2}{4}\right)}{\pi D_0} \tag{3.9}$$

Where Pt= tube pitch, m

Do= outer Diameter of the tube, m

$$Re_s = \frac{D_e G_s}{\mu} \tag{3.10}$$

Where De= equivalent diameter, m

μ= dynamic viscosity, Ns/ m^2

Shell side Nusselt number is given by Kern

$$Nu = 0.36\left[\frac{D_e G_s}{\mu_b}\right]^{0.55}\left[\frac{C_P \mu_b}{k}\right]^{0.33}\left[\frac{\mu_b}{\mu_w}\right]^{0.14} \tag{3.11}$$

Where;

k= thermal conductivity, W/m. K

μw= dynamic viscosity of water fluid, Ns/ m^2

μb= shell fluid dynamic viscosity at average temperature, Ns/ m^2

Gs= mass velocity of shell side, kg/ m^2. S

De= equivalent diameter of shell side, m

The shell-side heat transfer coefficient, ho, is then calculated as:

$$h_o = \frac{Nu.k}{D_e} \tag{3.12}$$

Where;

ho= heat transfer coefficient, W/ m^2. k

k= thermal conductivity, W/mK

Tube-side heat transfer coefficient by:

$$A_t = \pi \frac{D_i^2}{4} \tag{3.13}$$

Where Di= tube inner diameter, m

$$A_{tp} = \frac{A_t N_t}{no.of\ passes}$$

Where Nt= number of tubes

$$G_t = \frac{\dot{m}_t}{A_{tp}} \tag{3.14}$$

Where Gt= mass velocity of tube, kg/ m^2. s

Atp= heat transfer area based on tube surface, m^2

$$u_t = \frac{G_t}{\rho} \tag{3.15}$$

Where;

ρ= density of fluid at average temperature, kg/ m^3

$$Re_t = \frac{u_t \rho\, d_i}{\mu} \tag{3.16}$$

Where;

di= inner diameter of tube, m

Using the petukhov and kirillov correlation:

$$\mathrm{Nu} = \frac{\left(\frac{f}{2}\right)\mathrm{Re\ Pr}}{1.07+12.7\left(\frac{f}{2}\right)^{1/2}\left(\frac{\mathrm{Pr}^2}{3-1}\right)} \tag{3.17}$$

Where;

f= friction factor of flow

Re= Reynolds number

Pr= prandtl number

Where: $f = (1.58 \ln \mathrm{Re} - 3.28)^{-2}$ (3.18)

The tube-side heat transfer coefficient, hi, is then found as:

$$h_i = \frac{\mathrm{Nu.k}}{d_i} \tag{3.19}$$

The shell-side pressure drop can be calculated from equations

$$\Delta P = f\frac{G_s^2(N_b+1)D_s}{2\rho D_e \phi_s} \tag{3.20}$$

Where;

ΔP= pressure drop for shell side, Pa

Nb= number of baffles

Ǿs= viscosity correction factor for shell side fluid

$$N_b = \frac{L}{B}$$

$$f = \exp(0.576 - 0.19 \ln R_s) \tag{3.21}$$

The tube-side pressure drop can be calculated from Equation:

$$\Delta P_t = \left(4f\frac{LN_P}{D_i} + 4N_b\right)\frac{\rho u_m^2}{2} \tag{3.22}$$

Where Np= number of passes

f= friction factor of tube side

3.1.2 Stepwise Procedure for calculation:

The following steps are adopted for the calculation of parameters of shell and tube heat exchanger

1. The outlet temperatures of shell and tube heat exchanger are computed by equations (3.4) and (3.5)
2. The log mean temperature difference to the shell and tube are computed using equations (3.6)
3. Reynolds numbers on shell side using the following equations (3.10) is calculated
4. Nusselt number is computed on the shell side using the equation (3.11) by using Macadam's correlation
5. Heat transfer coefficient on shell side is calculated using the equation (3.12).
6. Pressure drop on shell side is calculated using the following equation (3.20)
7. Tube side pressure drop is calculated by using equations (3.16), (3.18), (3.22)

CHAPTER 4

RESULTS AND DISCUSSION

The program was evaluated with data obtained from four industrial heat exchangers. The program was written and implemented in MATLAB. It is essential to heat one fluid (either in the shell or tube side) by transfer heat with the other. Tables 4. (1, 3, 5, 7) demonstrates the working fluids, fluid physical properties and the fluid performance information for the four heat exchangers obtained from the refinery. It was essential to evaluate the heat exchangers assuming thermally and hydraulically realistic for the service they are being utilized for. The results were received after application of the program for the four heat exchangers that appear in Tables 4. (2, 4, 6, 8).

The results in Tables 4. (2,4,6,8) shows that the clean and design overall coefficients are greater than the required overall coefficient for the four heat exchangers. This implies that the heat exchangers are thermally suitable for the service they are being used for. Also, since the shell and tube side pressure drops are greater than the allowable pressure drop, the heat exchangers are hydraulically suitable for the service they are being used for.

Table.4.1 – Experimental (REAL) results E01

HEAT EXCHANGER					
Service: Crude Oil / Kerosene Exchanger				Item: E01	
	TEMA Type AES Horizontal Area 144 m^2				
1	Fluid Allocation	**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name	*Kerosene*		*Crude Oil*	
3	Total Fluid Entering kg/hr.	*42 350*		*392 480*	
4	Liquid	*42 350*		*392 480*	
5	Liquid Density kg/ m^3	*666.3*	*757.3*	*840.5*	*828.9*
6	Viscosity cP	*0.368*		*5.103*	
7	Specific Heat kcal/kg. °C	*0.561*		*0.457*	
8	Thermal Conductivity kcal/hr. m °C	*0.104*		*0.117*	
9	Temperature °C	*183 / 68*		*25 / 40*	
10	Operating Pressure kg/ cm^2 g	*4.50*		*18.40*	
11	Velocity m/sec	*0.30*		*2.75*	
12	Pressure Drop kg/ cm^2	*0.200*		*0.900*	
13	Heat Exchanged MM kcal/hr.	*2.753*			
14	Transfer Rate kcal/hr. m^2 °C	*385.2*			
15	O.D mm			*20.00*	
16	I.D mm	*750.0*		15.4	
17	Thk mm			*2.3*	
18	Length mm	*4 500*		*4 500*	
19	Pitch /angle mm/ Deg.	*26 / 45 °*			
20	Spacing center / Cut mm/ %	*151 / 22*			
21	MATERIALS	*SA 516 Gr.70*		*SA 179*	
25	No. Passes per Shell	*1*		*2*	

Table 4.2 – Numerical (MATLAB) results E01

<table>
<tr><td colspan="6">HEAT EXCHANGER</td></tr>
<tr><td colspan="4">Service: Crude Oil / Kerosene Exchanger</td><td colspan="2">Item: E01</td></tr>
<tr><td></td><td colspan="5">TEMA Type AES Horizontal Area 144 m^2</td></tr>
<tr><td></td><td colspan="5"></td></tr>
<tr><td>1</td><td>Fluid Allocation</td><td colspan="2">Shell side (In./Out.)</td><td colspan="2">Tube side (In./Out.)</td></tr>
<tr><td>2</td><td>Fluid Name</td><td colspan="2">Kerosene</td><td colspan="2">Crude Oil</td></tr>
<tr><td>3</td><td>Total Fluid Entering kg/hr.</td><td colspan="2">42 350</td><td colspan="2">392 480</td></tr>
<tr><td>4</td><td>Temperature °C</td><td>183</td><td>69.75</td><td>25</td><td>40</td></tr>
<tr><td>5</td><td>Correction Factor</td><td colspan="2">0.956031</td><td colspan="2">0.009135</td></tr>
<tr><td>6</td><td>Heat Transfer Coefficient</td><td colspan="2">602.813215</td><td colspan="2">978.453806</td></tr>
<tr><td>7</td><td>Reynold number</td><td colspan="2">28220.079610</td><td colspan="2">5990.953945</td></tr>
<tr><td>8</td><td>Nussle number</td><td colspan="2">133.646732</td><td colspan="2">128.787937</td></tr>
<tr><td>9</td><td>Friction factor</td><td colspan="2">0.253833</td><td colspan="2">0.009135</td></tr>
<tr><td>10</td><td>Overall Heat Transfer Co-efficient kcal/hr. m2 °C</td><td colspan="4">329.533908</td></tr>
<tr><td>11</td><td>Pressure drop kg/ cm^2</td><td colspan="2">0.168</td><td colspan="2">0.78</td></tr>
<tr><td>12</td><td>LMTD is</td><td colspan="4">84.576350</td></tr>
<tr><td></td><td>Heat Transferred MM kcal/hr.</td><td colspan="4">2.69</td></tr>
</table>

Table 4.3 Experimental (REAL) results E02

HEAT EXCHANGER						
Service: Crude Oil / Kerosene Exchanger					Item: E02	
	TEMA Type AES Horizontal Area *166.0*m²					
1	Fluid Allocation		**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name		*Light Gas Oil*		*Crude Oil*	
3	Total Fluid Entering	kg/hr.	*70 290*		*397 540*	
4	Liquid		*70 290*		*397 540*	
5	Liquid Density	kg/ m^3	*704.8*	*773.5*	*840.5*	*825.5*
6	Viscosity	cP	*0.531*		*4.709*	
7	Specific Heat	kcal/kg °C	*0.576*		*0.473*	
8	Thermal Conductivity	kcal/hr. m °C	*0.110*		*0.114*	
9	Temperature	°C	*196*	*104*	*39*	*59*
10	Operating Pressure	kg/ cm^2 g	*4.80*		*17.40*	
11	Velocity	m/sec	*0.43*		*2.41*	
12	Pressure Drop	kg/ cm^2	*0.300*		*0.700*	
13	Heat Exchanged	MM kcal/hr.	*3.732*			
14	Transfer Rate	kcal/hr. m^2 °C	*397.1*			
15	O.D	mm			20.00	
16	I.D	mm	*800.0*		15.4	
17	Thk	mm			*2.3*	
18	Length	mm	*4 500*		*4 500*	
19	Pitch / angle	mm /Deg.	*26 / 45 °*			
20	Spacing cent mm/ %Cut		*160 / 23*			
21	MATERIALS		*SA 515 Gr.70*		*SA 213 T2*	
25	No. Passes per Shell		*1*		*2*	

Table 4.4 Numerical (MATLAB) results EO2

HEAT EXCHANGER					
Service: Crude Oil / *Light Gas Oil* Exchanger				Item: E02	
	TEMA Type AES Horizontal Area 166 m^2				
1	Fluid Allocation	**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name	*Light Gas Oil*		*Crude Oil*	
3	Total Fluid Entering kg/hr.	*70 290*		*397 540*	
4	Temperature °C	196	103	39	59
5	Correction Factor	0.964374		0.008883	
6	Heat Transfer Coefficient	765.922798		1023.895899	
7	Reynold number	28702.357373		6575.628190	
8	Nusslte number	160.54664		138.315762	
9	Friction factor	0.253017		0.008883	
10	Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	381.328999			
11	Pressure drop kg/cm2	0.264		0.632	
12	LMTD is	95.987951			
13	Heat Transferred MM kcal/hr.	3.760			

Table 4.5 Experimental (REAL) results E03

<table>
<tr><td colspan="6">HEAT EXCHANGER</td></tr>
<tr><td colspan="4">Service: Crude Oil / Heavy Gas Oil Exchanger</td><td colspan="2">Item: E03</td></tr>
<tr><td></td><td colspan="5">TEMA Type AES Horizontal Area 96 m²</td></tr>
<tr><td></td><td colspan="5"></td></tr>
<tr><td>1</td><td>Fluid Allocation</td><td colspan="2">Shell side
(In./Out.)</td><td colspan="2">Tube side
(In./Out.)</td></tr>
<tr><td>2</td><td>Fluid Name</td><td colspan="2">Heavy Gas Oil</td><td colspan="2">Crude Oil</td></tr>
<tr><td>3</td><td>Total Fluid Entering
kg/hr.</td><td colspan="2">14 430</td><td colspan="2">392 480</td></tr>
<tr><td>4</td><td>Liquid</td><td colspan="2">14 430</td><td colspan="2">392 480</td></tr>
<tr><td>5</td><td>Liquid Density
kg/ m³</td><td>739.4</td><td>818.9</td><td>812.7</td><td>808.8</td></tr>
<tr><td>6</td><td>Viscosity
cP</td><td colspan="2">0.368</td><td colspan="2">5.103</td></tr>
<tr><td>7</td><td>Specific Heat
kcal/kg. °C</td><td colspan="2">0.565</td><td colspan="2">0.490</td></tr>
<tr><td>8</td><td>Thermal Conductivity
kcal/hr. m °C</td><td colspan="2">0.120</td><td colspan="2">0.111</td></tr>
<tr><td>9</td><td>Temperature
°C</td><td colspan="2">205 /90</td><td colspan="2">62 / 67</td></tr>
<tr><td>10</td><td>Operating Pressure
kg/ cm² g</td><td colspan="2">4.00</td><td colspan="2">16.6</td></tr>
<tr><td>11</td><td>Velocity
m/sec</td><td colspan="2">0.09</td><td colspan="2">2.83</td></tr>
<tr><td>12</td><td>Pressure Drop
kg/ cm²</td><td colspan="2">0.200</td><td colspan="2">0.800</td></tr>
<tr><td>13</td><td>Heat Exchanged
MM kcal/hr.</td><td colspan="4">0.941</td></tr>
<tr><td>14</td><td>Transfer Rate
kcal/hr. m² °C</td><td colspan="4">243.6</td></tr>
<tr><td>15</td><td>O.D
mm</td><td colspan="2"></td><td colspan="2">20.00</td></tr>
<tr><td>16</td><td>I.D
mm</td><td colspan="2">750.0</td><td colspan="2">15.4</td></tr>
<tr><td>17</td><td>Thk
mm</td><td colspan="2"></td><td colspan="2">2.3</td></tr>
<tr><td>18</td><td>Length mm</td><td colspan="2">3000</td><td colspan="2">3000</td></tr>
<tr><td>19</td><td>Pitch /angle
mm/ Deg.</td><td colspan="4">26 / 45 °</td></tr>
<tr><td>20</td><td>Spacing center / Cut
mm/ %</td><td colspan="4">150 / 22</td></tr>
<tr><td>21</td><td>MATERIALS</td><td colspan="2">SA 387 Gr.2</td><td colspan="2">SA 213 T2</td></tr>
<tr><td>25</td><td>No. Passes per Shell</td><td colspan="2">1</td><td colspan="2">2</td></tr>
</table>

Table 4.6 Numerical (MATLAB) results E03

HEAT EXCHANGER					
Service: Crude Oil / *Heavy Gas Oil* Exchanger				Item: E03	
	TEMA Type AES Horizontal Area 96 m^2				
1	Fluid Allocation	**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name	*Heavy Gas Oil*		*Crude Oil*	
3	Total Fluid Entering kg/hr.	*14 430*		*392 480*	
4	Temperature °C	205	87	62	67
5	Correction Factor	0.973438		0.007124	
6	Heat Transfer Coefficient	379.978403		1578.209536	
7	Reynold number	3861.103137		14393.927656	
8	Nussle number	73.010712		218.958800	
9	friction factor	0.370406		0.007124	
10	Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	285.467580			
11	Pressure drop kg/ cm^2	0.155		0.67	
12	LMTD is	66.200373			
	Heat Transferred MM kcal/hr.	0.961576000000			

Table 4.7 Experimental (REAL) results E04

	HEAT EXCHANGER				
	Service: Crude Oil / Naphtha Exchanger			Item: E04	
	TEMA Type AES Horizontal Area 505 m^2				
1	Fluid Allocation	**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name	*Naphtha*		*Crude Oil*	
3	Total Fluid Entering kg/hr.	*332 720*		*392 480*	
4	Liquid	*332 720*		*392 480*	
5	Liquid Density kg/ m^3	*664.7*	*697.3*	*808.3*	*764.9*
6	Viscosity cP	*0.25*		*1.414*	
7	Specific Heat kcal/kg. °C	*0.583*		*0.521*	
8	Thermal Conductivity kcal/hr. m °C	*0.092*		*0.105*	
9	Temperature °C	*161 / 102*		*67 / 122*	
10	Operating Pressure kg/ cm^2 g	*6.1*		*15.7*	
11	Velocity m/sec	*1.03*		*2.4*	
12	Pressure Drop kg/ cm^2	*1.6*		*2.6*	
13	Heat Exchanged MM kcal/hr.	*11.398*			
14	Transfer Rate kcal/hr. m^2 °C	*668.4*			
15	O.D mm			*20.00*	
16	I.D mm	*1200.0*		14.8	
17	Thk mm			*2.6*	
18	Length mm	*6000*		*6000*	
19	Pitch /angle mm/ Deg.	*26 / 45 °*			
20	Spacing center / Cut mm/ %	*240 / 23*			
21	MATERIALS	*SA 516 Gr.70*		*SA 213 T2*	
25	No. Passes per Shell	*1*		*2*	

Table 4.8 Numerical (MATLAB) results E04

HEAT EXCHANGER					
Service: Crude Oil / *Naphtha* Exchanger				Item: E04	
	TEMA Type AES Horizontal Area 505 m^2				
1	Fluid Allocation	**Shell side** (In./Out.)		**Tube side** (In./Out.)	
2	Fluid Name	Naphtha		*Crude Oil*	
3	Total Fluid Entering kg/hr.	*332 720*		*392 480*	
4	Temperature °C	161	103.02	67	122
5	Correction Factor	0.337731		0.007980	
6	Heat Transfer Coefficient	1383.964995		902.378315	
7	Reynold number	128255.183350		9519.199755	
8	Nussle number	346.853592		132.348820	
9	Friction factor	0.190379		0.007980	
10	Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	452.455774			
11	Pressure drop kg/ cm^2	1.32		2.101	
12	LMTD is	37.490793			
	Heat Transferred MM kcal/hr.	11.2465144			

Table 4.9 A comparison of numerical and experimental results E0-1

	Experimental Results	MATLAB Results	%Difference
Heat Transferred MM kcal/hr.	2.753	2.69	2.18
Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	385.2	329.533	14.45
Pressure Drop kg/ cm^2 (shell / tub	*0.2 /0.9*	*0.168/ 0.78*	*16 /13.3*

Table 4.10 A comparison of numerical and experimental results E0-2

	Experimental Results	MATLAB Results	%Difference
Heat Transferred MM kcal/hr	3.732	3.76	-0.8
Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	397.1	381.328	3.97
Pressure Drop kg/ cm^2 (shell / tube)	0.3 /0.7	0.264/ 0.632	12 /9.7

Table 4.11 A comparison of numerical and experimental results E0-3

	Experimental Results	MATLAB Results	%Difference
Heat Transferred MM kcal/hr	*0.941*	0.961576	-2.1866
Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	*243.6*	285.467580	-1718
Pressure Drop kg/ cm^2 (shell / tube)	*0.200* /0.8	0.155/ 0.67	22.5/16.25

Table 4.12 A comparison of numerical and experimental results E0-4

	Experimental Results	MATLAB Results	%Difference
Heat Transferred MM kcal/hr	11.398	11.2465144	1.329
Overall Heat Transfer Co-efficient kcal/hr. m^2 °C	668.4	452.455774	32.37
Pressure Drop kg/ cm^2 (shell / tube)	1.6 /2.6	1.32/ 2.101	17.5 /19.19

CHAPTER 5

CONCLUSION AND FUTURE WORK

5.1 Conclusion

A PC program in MATLAB was created to assess the performance of shell and tube heat exchangers for their thermal and hydraulic suitability.

The Program which was tested using field data from four industrial exchangers (Ex. 1, Ex. 2, Ex. 3, and Ex.4) showed that the result obtained, compares reasonably with the actual performance data, in this manner, exhibiting that the program is dependable and can be connected in the performance analysis of shell and tube heat exchangers.

5.2 Future Work

- This work can be branched out for various bundle tube setups, for example, triangular pitch, and for various tube formats for heat transfer analysis on shell and tube heat exchanger for specific applications.
- The setup can be tried with totally different designs, for instance altering the baffle layout or type of shell and tube tool in CFD too. The device style qualities can be changed like tube arrangement, tube length, etc.
- Use the CFD programming taking an extra exact prompt to least time.

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APPENDIX-A

MATLAB PROGRAM

The following "**MATLAB**" program has been developed to evaluate output parameters for four different heat exchangers in general:

- The program to evaluate the Res no & ΔPs in the case of shell and tube heat exchanger
- Initial temperatures & fluid properties are to be defined initially.

```
close all, clear all,
clc


tci=input('Enter the Cold Fluid Inlet Temp :tci \n tci= ');
sprintf('%f', tci);
%tci=25;tco=40;thi=183;
tco=input('Enter Cold Fluid Outlet Temp : tco \n tco= ');
sprintf('%f', tco);
thi=input('Enter Hot Fluid Inlet Temp: thi \n thi= ');
sprintf('%f', thi);
%printf("Enter Tubeside Specifications\n");
%printf("Enter Outer Dia of Tube: do \n");
%scanf("%f",&d0);
d0=0.02;
%printf("Enter Inner Dia of Tube:di\n");
%scanf("%f",&di);
di=0.0154 ;
%printf("Enter Flow Area of Tube-side: ac \n");
%scanf("%f",&ac);
ac=0.0001862;
%printf("Enter Wall Thickness of Tube-side: tw \n");
%scanf("%f",&tw);
tw=0.0023;
%printf("Enter Mass Tube Side fluid mass flow rate: mt
\n");
%scanf('%f', &mt);
```

```
mt=392480;
input('Enter the Tube Side Fluid Properties\n');
cpt=input('Enter the specific heat of tubeside fluid:cpt\n
cpt= '); %%% its cp and I'm convert to cpt
sprintf('%f', cpt);
%cpt=0.457;
ubt=input('Enter Viscosity :ubt \n ubt= ');
sprintf('%f', ubt);
%ubt=18.37;
kt=input('Enter Thermal Conductivity : kt \n kt= ');
sprintf('%f',kt);
%kt=0.117;
pt=input('Enter Density : pt \n pt= ');
sprintf('%f',pt);
%pt=834.7;
prt=input('Enter Prandtl Number : prt \n prt= ');
sprintf('%f',prt);

%prt=77.75;
%printf("Enter Shell side specifications");
%printf("Enter Pitch Size :pst \n");
%scanf("%f", &pst);
pst=0.026;
%printf("Enter Clearance: c \n");
%scanf("%f", &c);
c=0.006;
%printf("Enter Baffle spacing: b \n");
%scanf("%f", &b);
b=0.151;
%printf("Enter Shell side Diameter:ds ");
%scanf("%f", &ds);
ds=0.75;
%printf("Enter Mass Shell Side fluid mass flow rate: ms
\n");
%scanf("%f", &ms);
ms=42350;
input('Enter the Shell Side Fluid Properties\n');
cps=input('Enter Specific Heat : cps \n cps= ');
sprintf('%f', cps);
%cps=0.561;
ubs=input('Enter Viscosity : ubs\n ubs= ');
sprintf('%f', ubs);
%ubs=1.324;
ks=input('Enter Thermal Conductivity : ks \n ks= ');
sprintf('%f', ks);
%ks=0.104;
ps=input('Enter Density : ps \n ps= ');
sprintf('%f', ps);
%ps=711.8;
prs=input('Enter Prandtl Number: prs \n prs= ');
sprintf('%f', prs);
```

```
%prs=7.1;
disp('Calculations of LMTD and Mass Flow Rates');
q =mt*cpt*(tco-tci);
tho=thi - (q/(ms*cps));
fprintf('Heat Transferred = %f\n',q);  %%%%% here I write
title I think don't need
fprintf('Hot Fluid inlet Temp is %f\n',thi);   %%
fprintf('Hot Fluid Outlet Temp is %f\n',tho);%%
fprintf('Cold Fluid inlet Temp is %f\n',tci);%%
fprintf('Cold Fluid Outlet Temp is %f\n',tco);%%   the same
% dt1=0.000;
% dt2=0.000;
% dt3=0.000;
% dt4=0.000;
% dt5=0.000;
% dt6=0.000;
dt1=thi - tho;
sprintf('dt1 is %f\n',dt1);     %%%% the same
dt2=tho - tci;
sprintf('dt2 is %f\n',dt2);       %%%%% same
dt3=thi - tco;
sprintf('dt3 is %f\n',dt3);        %% same
dt4=tco - tci;
sprintf('dt4 %f\n',dt4);     %% same
dt5=thi - tci;
input('Enter the type of Flow: 1. Parallel Flow   2.
Counter Flow  ');
% sprintf('%d',ch);  its error
% ch=sprintf('ch is %d \n',ch);
ch=2;
switch ch

case 2
lmtd =((thi - tco)-(tho - tci))/(log((thi - tco)/(tho -
tci)));
fprintf('LMTD is %f\n',lmtd);
%%break;
case 1
lmtd =((thi - tci)-(tho - tco))/(log((thi - tci)/(tho -
tco)));
fprintf('LMTD is %f\n',lmtd);
%%break;
end

r=dt1/dt4;
fprintf('r value %f\n',r);
p=dt4/dt5;
fprintf('p value %f\n',p);
a1=sqrt((r*r)+1);
fprintf('a1 value %f\n',a1);
a2=(1 - p);
```

```
fprintf('a2 value %f\n',a2);
a3=(1-(p*r));
fprintf('a3 value %f\n',a3);
a4=p*((r+1)- a1);
fprintf('a4 value %f\n',a4);
a5=p*((r+1)+a1);
fprintf('a5 value %f\n',a5);
a6=log((2 - a4)/(2 - a5));
fprintf('a6 value %f\n',a6);
a7=log(a2/a3);
fprintf('a7 value %f\n',a7);
f=(a1*a7)/((r - 1)*a6);
fprintf('Correction Factor %f\n', f);
fprintf('Calculation of Shell Side Heat Transfer Co-
efficient\n');
as=(ds*c*b)/pst;
fprintf('as%f\n',as);
gs=ms/as;
fprintf('gs %f\n',gs);
de=((4*pst*pst)-(3.14*d0*d0))/(3.14*d0);
fprintf('de %f\n',de);
x1=(de*gs)/ubs;
fprintf('x1 %f\n',x1);
x2=(cps*ubs)/ks;
fprintf('x2 %f\n',x2);
%scanf('%f', &ut);
uw=18.37;
x3=ubs/uw;
fprintf('x3 %f\n',x3);
t1=x1 ^ 0.55;
fprintf('t1 %f\n',t1);
t2=x2 ^ 0.33;
fprintf('t2 %f\n',t2);
t3=x3 ^ 0.14;


fprintf('t3 %f\n',t3);
nus=0.36*t1*t2*t3;
fprintf('nus %f\n',nus);
ho=(nus*ks)/de;
fprintf('ho %f\n',ho);
fprintf('Calculation of Tube side Heat Transfer Co-
efficient');   %%%% may disp
at=(3.14/4)*di*di;
%scanf("%d", &nop);
%scanf("%d", &nt);
nop=2;nt=590;
atp=(nt*at)/nop;
fprintf('atp %f\n',atp);
gt=mt/atp;
ut=gt/pt;
```

```
ret=(gt*di)/ubt;
fprintf('ret %f\n',ret);
f1=((1.58*log(ret))-3.28);
fprintf('f1 %f\n',f1);
tf=f1 ^ -2;
fprintf('f %f\n',tf);
e4=tf/2;
e1=sqrt(e4);
e2=prt ^ 0.67;
e3=12.7*e1*(e2-1);
nut=(e4*ret*prt)/(1.07+e3);
fprintf('nut %f\n',nut);
hi=(nut*kt)/di;
fprintf('hi %f\n',hi);
fprintf('Calculation of Overall Heat Transfer
Coefficient');
m1=d0/(di*hi);
m2=log(d0/di);
m3=1/ho;
%scanf('%f',&kw);
kw=54;
m4=(d0*m2)/(2*kw) ;
u1=m1+m4+m3;
fprintf('1 %f\n',u1);
uu=1/u1;
fprintf('uu %f\n',uu);
area=144;
len=4.5;
nb=26;
k1=log(x1);
fp=exp(0.576 - 0.19*k1);
fprintf('fpis %f\n',fp);
fprintf('Heat Transferred = %f\n',q);
len=4.5;
fprintf('len %f\n',len);
nb=26;
fprintf('nb %f\n',nb);
fprintf('res = %f\n',x1);
k1=log(x1);


fp=exp(0.576 - 0.19*k1);
fprintf('fpis %f\n',fp);
dp=(fp*gs*gs*(nb+1)*ds)/(2*ps*de*x3);
fprintf('pr drop = %f\n',dp);
fprintf('-------------------------------------------------
-----------');
fprintf('\n');
fprintf('Heat Transfered = %f\n',q);
fprintf('Hot Fluid inlet Temp is %f\n',thi);
fprintf('Hot Fluid Outlet Temp is %f\n',tho);
```

```
fprintf('Cold Fluid inlet Temp is %f\n',tci);
fprintf('Cold Fluid Outlet Temp is %f\n',tco);
fprintf('LMTD is %f\n',lmtd);
fprintf('Correction Factor-Shell Side %f\n', f);
fprintf('Heat Transfer Coefficient -Shell Side %f\n',ho);
fprintf('Renold number-Shell Side %f\n',x1) ;
fprintf('Nusselt number-Shell Side %f\n',nus) ;
fprintf('Correction Factor-Tube Side %f\n',tf);
fprintf('Heat Transfer Coefficient -Tube Side %f\n',hi);
fprintf('Renold number-Tube Side %f\n',ret) ;
fprintf('Nusselt number-Tube Side %f\n',nut) ;
fprintf('Overall Heat Transfer Co-efficient = %f\n',uu);
fprintf('shell side friction factor = %f\n',fp);
fprintf('tube side friction factor = %f\n',tf);
fprintf('Area = %f\n',area);
fprintf('Length = %f\n',len);
fprintf('Number of baffles = %f\n',nb);
fprintf('pressure drop = %f\n',dp);
fprintf('-------------------------------------------------
-----------');
fprintf('\n');
```

APPENDIX-B

FLOW DIAGRAM

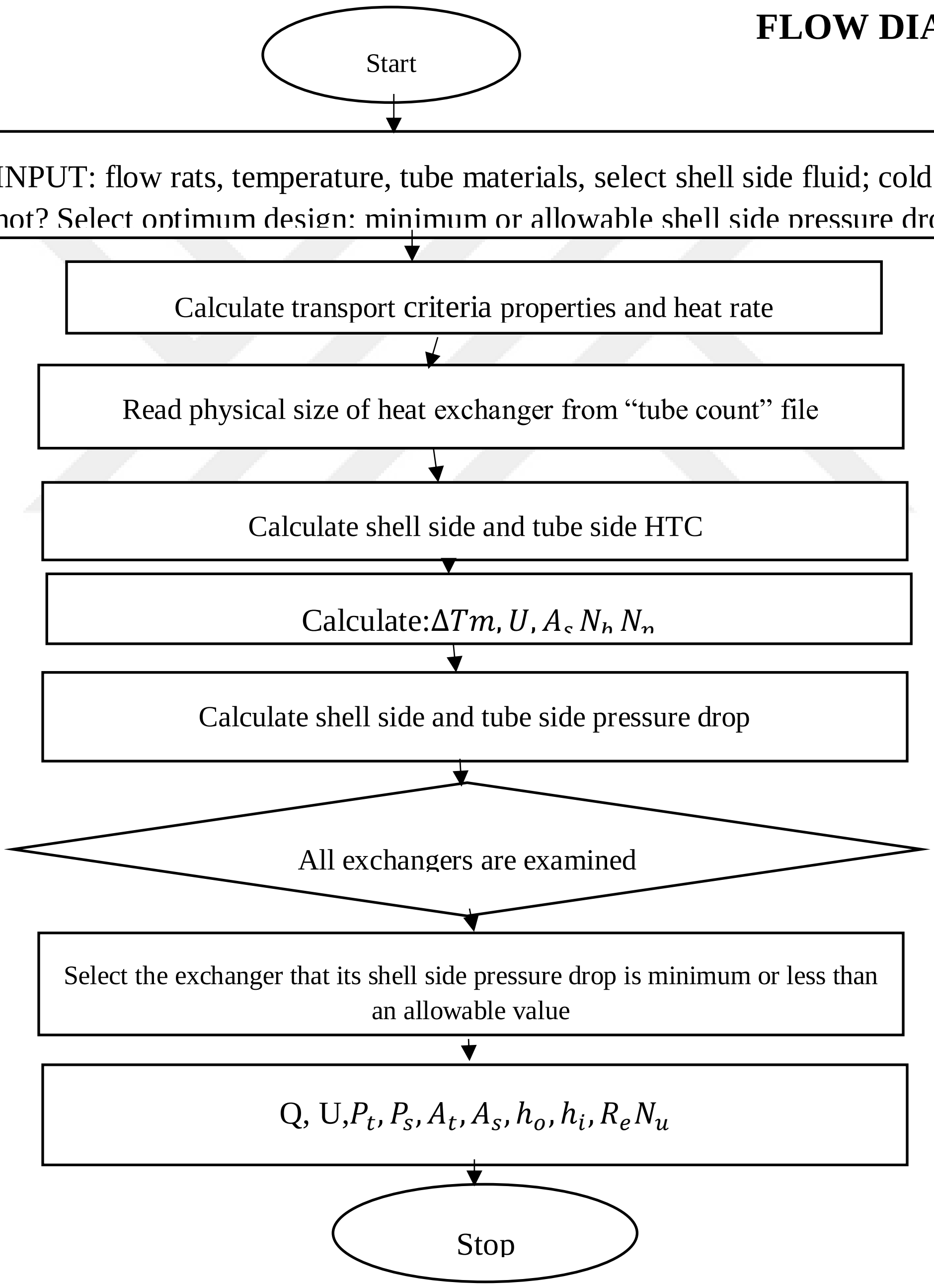

Fig.B.1. Flow Diagram of the Design Program

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1. AL Nidawi M. B .and Heperkan. H. (2017). “Heat Transfer Analysis on Crude Oil and Petroleum Products Using the Heat Exchangers in Petroleum Industries”, Proceeding of International Conference on Progressing Applied Science 2017 (ICPAS 2017), 04-06 January 2017, Istanbul, Turkey. ISBN: 978-605-9546-02-7.