SHELL AND TUBE HEAT EXCHANGER DESIGN USING CFD TOOLS

Submitted by

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ABSTRACT

In present day shell and tube heat exchanger is the most common type heat exchanger widely use in oil refinery and other large chemical process, because it suits high pressure application. The process in solving simulation consists of modeling and meshing the basic geometry of shell and tube heat exchanger using CFD package SOLIDWORKS. The objective of the project is design of shell and tube heat exchanger with helical baffle and study the flow and temperature field inside the shell using SOLIDWORKS software tools. The heat exchanger contains 7 tubes and 600 mm length shell diameter 90 mm. The flow pattern in the shell side of the heat exchanger with continuous helical baffles was forced to be rotational and helical due to the geometry of the continuous helical baffles, which results in a significant increase in heat transfer coefficient per unit pressure drop in the heat exchanger.

CONTENTS

Nomenclature

L	Heat exchanger length
Di	Shell inner diameter,
do	Tube outer diameter
Nt	Number of tubes,
Nb	Number of baffles.
B	Central baffle spacing,
Bc	Baffle cut

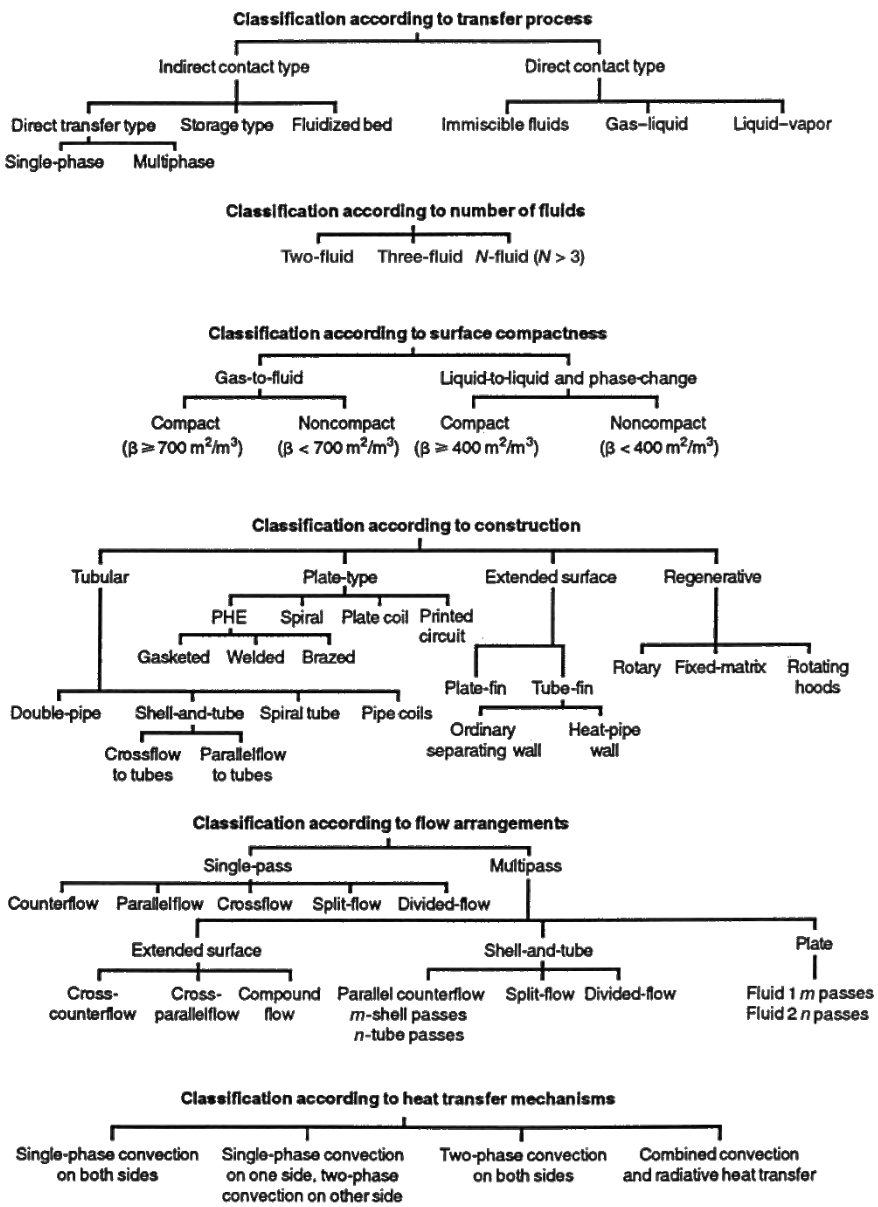
Chapter 1

Introduction

1.INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single- or multicomponent fluid streams. In other applications, the objective may be to recover or reject heat, or sterilize, pasteurize, fractionate, distill, concentrate, crystallize, or con-trol a process fluid. In a few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak. Such exchangers are referred to as direct transfer type, or simply recuperators. In con-trast, exchangers in which there is intermittent heat exchange between the hot and cold fluids—via thermal energy storage and release through the exchanger surface or matrix— are referred to as indirect transfer type, or simply regenerators. Such exchangers usually have fluid leakage from one fluid stream to the other, due to pressure differences and matrix rotation/valve switching. Common examples of heat exchangers are shell-and-tube exchangers, automobile radiators, condensers, evaporators, air preheaters, and cooling towers. If no phase change occurs in any of the fluids in the exchanger, it is sometimes referred to as a sensible heat exchanger. There could be internal thermal energy sources in the exchangers, such as in electric heaters and nuclear fuel elements. Combustion and chemical reaction may take place within the exchanger, such as in boilers, fired heaters, and fluidized-bed exchangers. Mechanical devices may be used in some exchangers such as in scraped surface exchangers, agitated vessels, and stirred tank reactors. Heat transfer in the separating wall of a recuperator generally takes place by conduction. However, in a heat pipe heat exchanger, the heat pipe not only acts as a separating wall, but also facilitates the transfer of heat by condensation, evaporation, and conduction of the working fluid inside the heat pipe. In general, if the fluids are immiscible, the separating wall may be eliminated, and the interface between the fluids replaces a heat transfer surface, as in a direct-contact heat exchanger.

FIGURE 1.1 Classification of heat exchangers (Shah, 1981).



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

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

Not only are heat exchangers often used in the process, power, petroleum, transpor-tation, air-conditioning, refrigeration, cryogenic, heat recovery, alternative fuel, and manufacturing industries, they also serve as key components of many industrial products available in the marketplace. These exchangers can be classified in many different ways. We will classify them according to transfer processes, number of fluids, and heat transfer mechanisms. Conventional heat exchangers are further classified according to construc-tion type and flow arrangements. Another arbitrary classification can be made, based on the heat transfer surface area/volume ratio, into compact and noncompact heat exchan-gers. This classification is made because the type of equipment, fields of applications, and design techniques generally differ. All these classifications are summarized in Fig. 1.1 and discussed further in this chapter. Heat exchangers can also be classified according to the process function, as outlined in Fig. 1.2. However, they are not discussed here and the reader may refer to Shah and Mueller (1988). Additional ways to classify heat exchangers are by fluid type (gas–gas, gas–liquid, liquid–liquid, gas two-phase, liquid two-phase, etc.), industry, and so on.[1]

1.1.Classification according to transfer processes

Heat exchangers are classified according to transfer processes into indirect- and direct-contact types.[2]

1.1.1.Indirect-Contact Heat Exchangers:

In an indirect-contact heat exchanger, the fluid streams remain separate and the heat transfers continuously through an impervious dividing wall or into and out of a wall in a transient manner. Thus, ideally, there is no direct contact between thermally interacting fluids. This type of heat exchanger, also referred to as a surface heat exchanger, can be further classified into direct-transfer type, storage type, and fluidized-bed exchangers.[2]

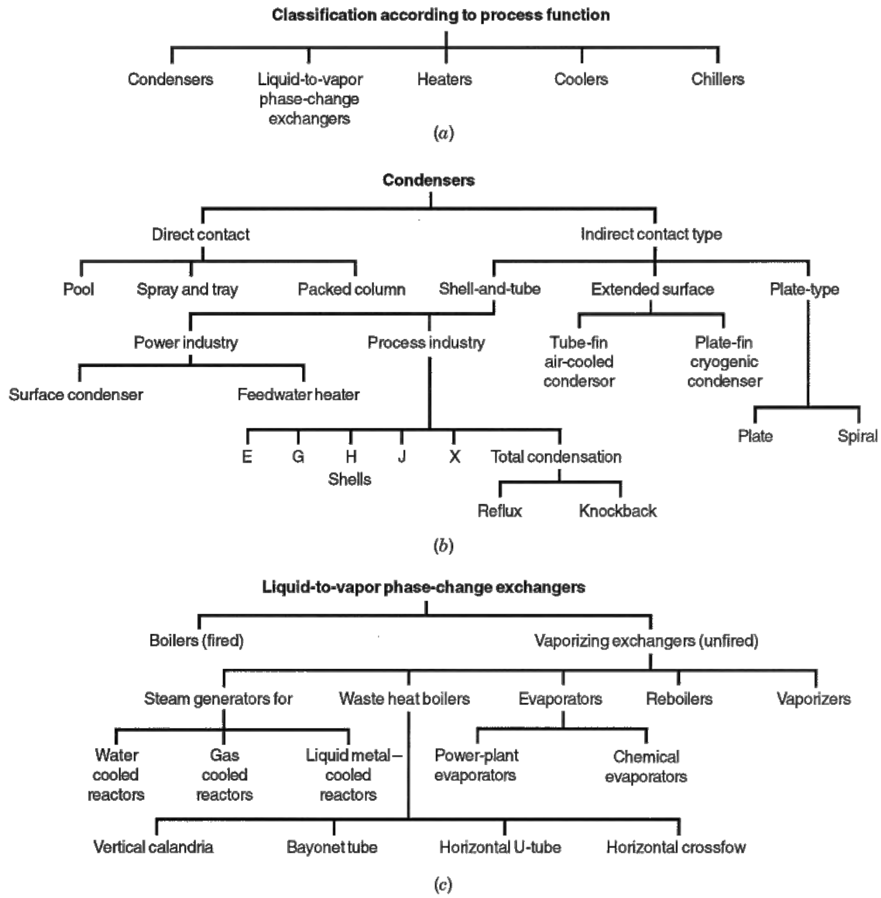


FIGURE 1.2 (a) Classification according to process function; (b) classification of condensers; (c) classification of liquid-to-vapor phase-change exchangers.

1.1.2.Direct-Transfer Type Exchangers:

In this type, heat transfers continuously from the hot fluid to the cold fluid through a dividing wall. Although a simultaneous flow of two (or more) fluids is required in the exchanger, there is no direct mixing of the two (or more) fluids because each fluid flows in separate fluid passages. In general, there are no moving parts in most such heat exchangers. This type of exchanger is designated as a recuperative heat exchanger or simply as a recuperator.{ Some examples of direct-transfer type heat exchangers are tubular, plate-type, and extended surface exchangers.

Note that the term recuperator is not commonly used in the

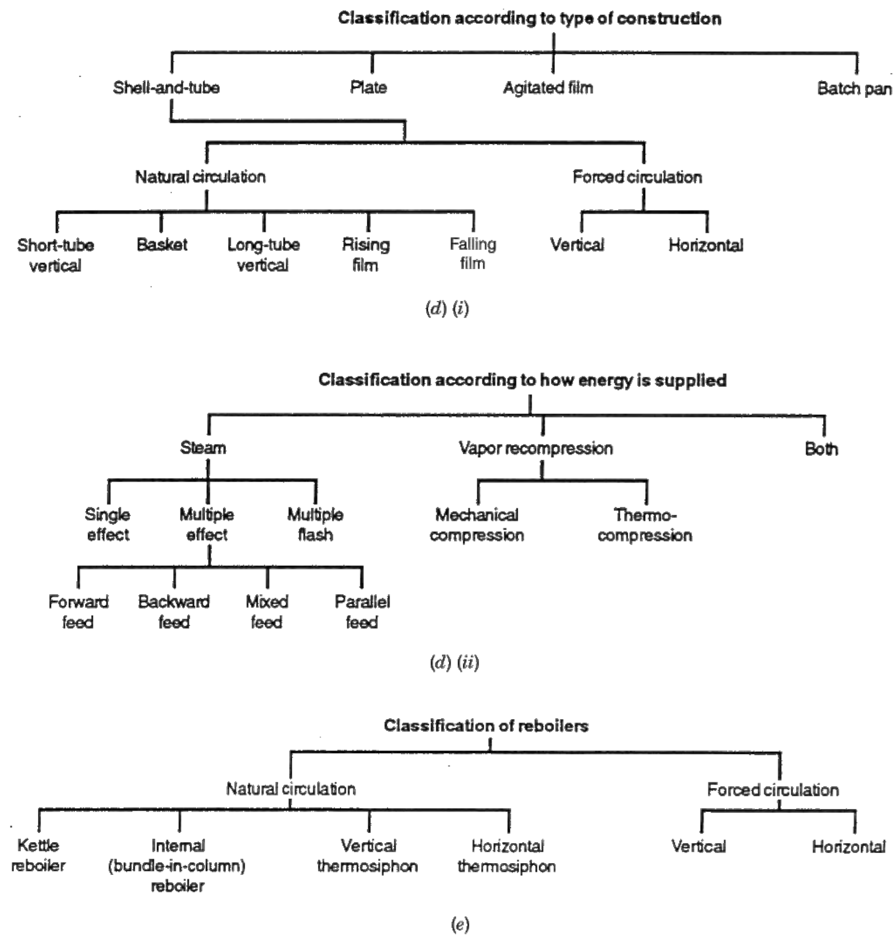


FIGURE 1.2 (d) classification of chemical evaporators according to (i) the type of construction, and (ii) how energy is supplied (Shah and Mueller, 1988); (e) classification of reboilers.

process industry for shell-and-tube and plate heat exchangers, although they are also considered as recuperators. Recuperators are further subclassified as prime surface exchangers and extended-surface exchangers. Prime surface exchangers do not employ fins or extended surfaces on any fluid side. Plain tubular exchangers, shell-and-tube exchangers with plain tubes, and plate exchangers are good examples of prime surface exchangers. Recuperators consti-tute a vast majority of all heat exchangers.[2]

1.1.3.Storage Type Exchangers:

In a storage type exchanger, both fluids flow alter-natively through the same flow passages, and hence heat transfer is intermittent. The heat transfer surface (or flow passages) is generally cellular in structure and is referred to as a matrix, or it is a permeable (porous) solid material, referred to as a packed bed. When hot gas flows over the heat transfer surface (through flow passages), the thermal energy from the hot gas is stored in the matrix wall, and thus the hot gas is being cooled during the matrix heating period. As cold gas flows through the same passages later (i.e., during the matrix cooling period), the matrix wall gives up thermal energy, which is absorbed by the cold fluid. Thus, heat is not transferred continuously through the wall as in a direct-transfer type exchanger (recuperator), but the corre-sponding thermal energy is alternately stored and released by the matrix wall. This storage type heat exchanger is also referred to as a regenerative heat exchanger, or simply as a regenerator. To operate continuously and within a desired temperature range, the gases, headers, or matrices are switched periodically (i.e., rotated), so that the same passage is occupied periodically by hot and cold gases. The actual time that hot gas takes to flow through a cold regenerator matrix is called the hot period or hot blow, and the time that cold gas flows through the hot regenerator matrix is called the cold period or cold blow. For successful operation, it is not necessary to have hot- and cold-gas flow periods of equal duration. There is some unavoidable carryover of a small fraction of the fluid trapped in the passage to the other fluid stream just after switching of the fluids; this is referred to as carryover leakage. In addition, if the hot and cold fluids are at different pressures, there will be leakage from the high-pressure fluid to the low-pressure fluid past the radial, peripheral, and axial seals, or across the valves. This leakage is referred to as pressure leakage. Since these leaks are unavoidable, regenerators are used exclusively in gas-to-gas heat (and mass) transfer applications with sensible heat transfer; in some applications, regenerators may transfer moisture from humid air to dry air up to about 5%.[2]

1.1.4.Fluidized-Bed Heat Exchangers:

In a fluidized-bed heat exchanger, one side of a two-fluid exchanger is immersed in a bed of finely divided solid material, such as a tube bundle immersed in a bed of sand or coal particles. If the upward fluid velocity on the bed side is low, the solid particles will remain fixed in position in the bed and the fluid will flow through the interstices of the bed. If the upward fluid velocity is high, the solid particles will be carried away with the fluid. At a ‘‘proper’’ value of the fluid velocity, the upward drag force is slightly higher than the weight of the bed particles. As a result, the solid particles will float with an increase in bed volume, and the bed behaves as a liquid. This characteristic of the bed is referred to as a fluidized condition. Under this condition, the fluid pressure drop through the bed remains almost constant, independent of the flow rate, and a strong mixing of the solid particles occurs. This results in a uniform temperature for the total bed (gas and par-ticles) with an apparent thermal conductivity of the solid particles as infinity. Very high heat transfer coefficients are achieved on the fluidized side compared to particle-free or dilute-phase particle gas flows. Chemical reaction is common on the fluidized side in many process applications, and combustion takes place in coal combustion fluidized beds. The common applications of the fluidized-bed heat exchanger are drying, mixing, adsorption, reactor engineering, coal combustion, and waste heat recovery. Since

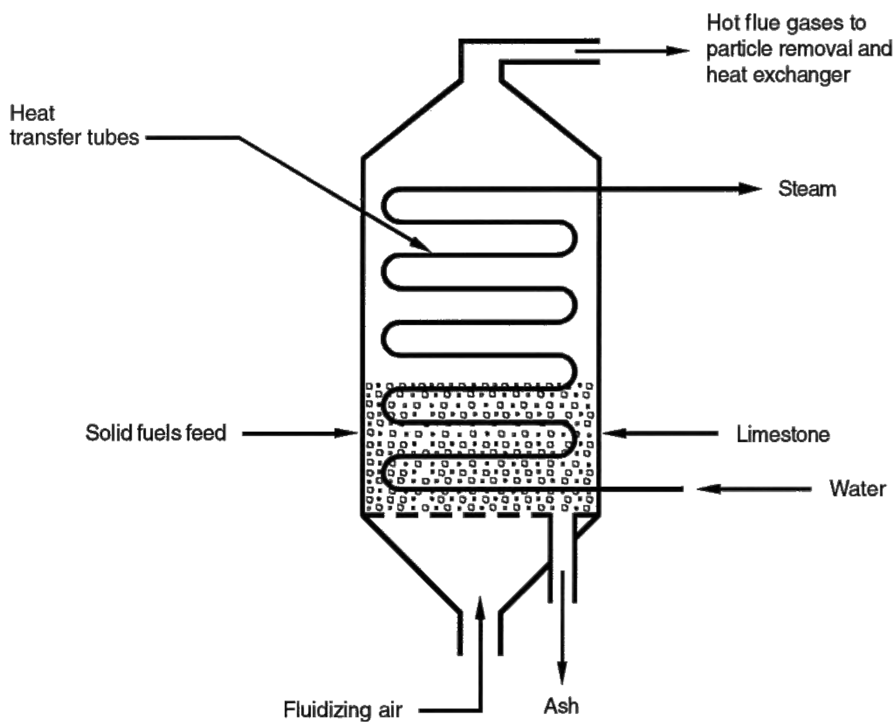


FIGURE 1.3 Fluidized-bed heat exchanger.

the initial temperature difference ($T_{h;i}$ _ $T_{f\ ;i}$) is reduced due to fluidization, the exchanger effectiveness is lower, and hence "-NTU theory for a fluidized-bed exchanger needs to be modified (Suo, 1976). [2]

1.1.5.Direct-Contact Heat Exchangers:

In a direct-contact exchanger, two fluid streams come into direct contact, exchange heat, and are then separated. Common applications of a direct-contact exchanger involve mass transfer in addition to heat transfer, such as in evaporative cooling and rectification; applications involving only sensible heat transfer are rare. The enthalpy of phase change in such an exchanger generally represents a significant portion of the total energy trans-fer. The phase change generally enhances the heat transfer rate. Compared to indirect-contact recuperators and regenerators, in direct-contact heat exchangers, (1) very high heat transfer rates are achievable, (2) the exchanger construction is relatively inexpensive, and (3) the fouling problem is generally nonexistent, due to the absence of a heat transfer surface (wall) between the two fluids. However, the applications are limited to those cases where a direct contact of two fluid streams is permissible. The design theory for these exchangers is beyond the scope of this book and is not covered. [2]

1.1.6.Immiscible Fluid Exchangers:

In this type, two immiscible fluid streams are brought into direct contact. These fluids may be single-phase fluids, or they may involve condensation or vaporization. Condensation of organic vapors and oil vapors with water or air are typical examples. [2]

1.1.7.Gas–Liquid Exchangers:

In this type, one fluid is a gas (more commonly, air) and the other a low-pressure liquid (more commonly, water) and are readily separable after the energy exchange. In either cooling of liquid (water) or humidification of gas (air) applications, liquid partially evaporates and the vapor is carried away with the gas. In these exchangers, more than 90% of the energy transfer is by virtue of mass transfer (due to the evaporation of the liquid), and convective heat transfer is a minor mechan-ism. A ‘‘wet’’ (water) cooling tower with forced- or natural-draft airflow is the most common application. Other applications are the air-conditioning spray chamber, spray drier, spray tower, and spray pond. [2]

1.1.8.Liquid–Vapor Exchangers:

In this type, typically steam is partially or fully condensed using cooling water, or water is heated with waste steam through direct contact in the exchanger.

Noncondensables and residual steam and hot water are the outlet streams. Common examples are desuperheaters and open feedwater heaters (also known as deaeraters) in power plants. [2]

1.2.Classificatıon According To Number Of Fluids

Most processes of heating, cooling, heat recovery, and heat rejection involve transfer of heat between two fluids. Hence, two-fluid heat exchangers are the most common. Three-fluid heat exchangers are widely used in cryogenics and some chemical processes (e.g., air separation systems, a helium–air separation unit, purification and liquefaction of hydro-gen, ammonia gas synthesis). Heat exchangers with as many as 12 fluid streams have been used in some chemical process applications. The design theory of three- and multifluid heat exchangers is algebraically very complex.[1]

1.3.Classification According To Construction Features

Heat exchangers are frequently characterized by construction features. Four major con-struction types are tubular, plate-type, extended surface, and regenerative exchangers. Heat exchangers with other constructions are also available, such as scraped surface exchanger, tank heater, cooler cartridge exchanger, and others (Walker, 1990). Some of these may be classified as tubular exchangers, but they have some unique features compared to conventional tubular exchangers. Since the applications of these exchangers are specialized, we concentrate here only on the four major construction types noted above.

Although the "-NTU and MTD methods are identical for tubular, plate-type, and extended-surface exchangers, the influence of the following factors must be taken into account in exchanger design: corrections due to leakage and bypass streams in a shell-and-tube exchanger, effects due to a few plates in a plate exchanger, and fin efficiency in an extended-surface exchanger. Similarly, the "-NTU method must be modified to take into account the heat capacity of the matrix in a regenerator. Thus, the detailed design theory differs for each construction type. Let us first discuss the construction features of the four major types.[1]

1.3.1.Tubular Heat Exchangers

These exchangers are generally built of circular tubes, although elliptical, rectangular, or round/flat twisted tubes have also been used in some applications. There is considerable flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length, and arrangement. Tubular exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids. Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications. They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and/ or pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchangers would work. These exchangers may be classified as shell-and-tube, double-pipe, and spiral tube exchangers. They are all prime surface exchangers except for exchangers having fins outside/inside tubes.[3]

1.3.2.Shell-and-Tube Exchangers:

This exchanger, shown in Fig., is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes. The major components of this exchanger are tubes (or tube bundle), shell, front-end head, rear-end head, baffles, and tubesheets, and are described briefly later in this subsection.

A variety of different internal constructions are used in shell-and-tube exchangers, depending on the desired heat transfer and pressure drop performance and the methods employed to reduce thermal stresses, to prevent leakages, to provide for ease of cleaning, to contain operating pressures and temperatures, to control corrosion, to accommodate highly asymmetric flows, and so on. Shell-and-tube exchangers are classified and con-structed in accordance with the widely used TEMA (Tubular Exchanger Manufacturers Association) standards (TEMA, 1999), DIN and other standards in Europe and else-where, and ASME (American Society of Mechanical Engineers) boiler and pressure vessel codes. TEMA has developed a notation system to designate major types of shell-and-tube exchangers. In this system, each exchanger is designated by a three-letter combination, the first letter indicating the front-end head type, the second the shell type, and the third the rear-end head type. These are identified in Fig. Some common shell-and-tube exchangers are AES, BEM, AEP, CFU, AKT, and AJW. It should be emphasized that there are other special types of shell-and-tube exchangers commercially available that have front- and rear-end heads different from those in Fig. Those exchangers may not be identifiable by the TEMA letter designation.

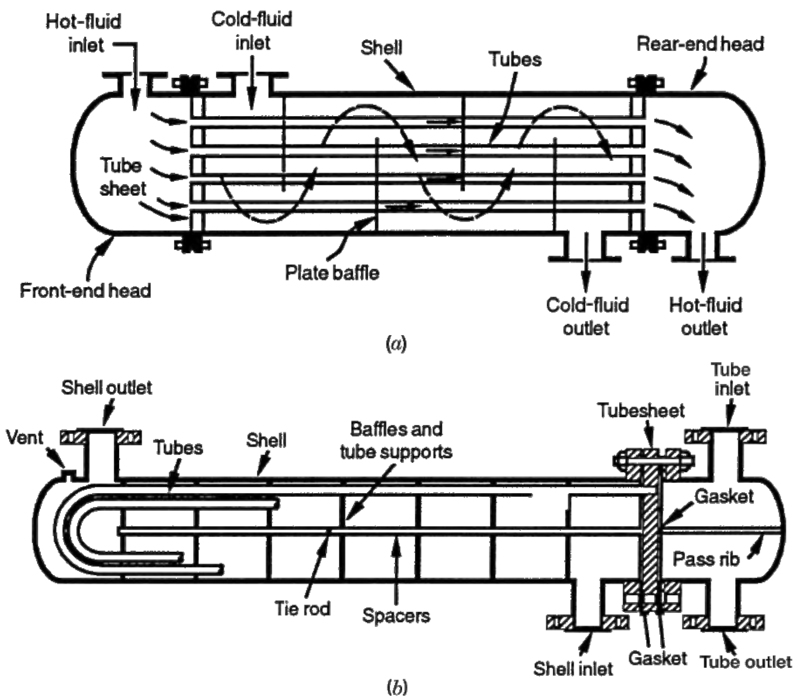


FIGURE 1.5 (a) Shell-and-tube exchanger (BEM) with one shell pass and one tube pass; (b) shell-and-tube exchanger (BEU) with one shell pass and two tube passes.

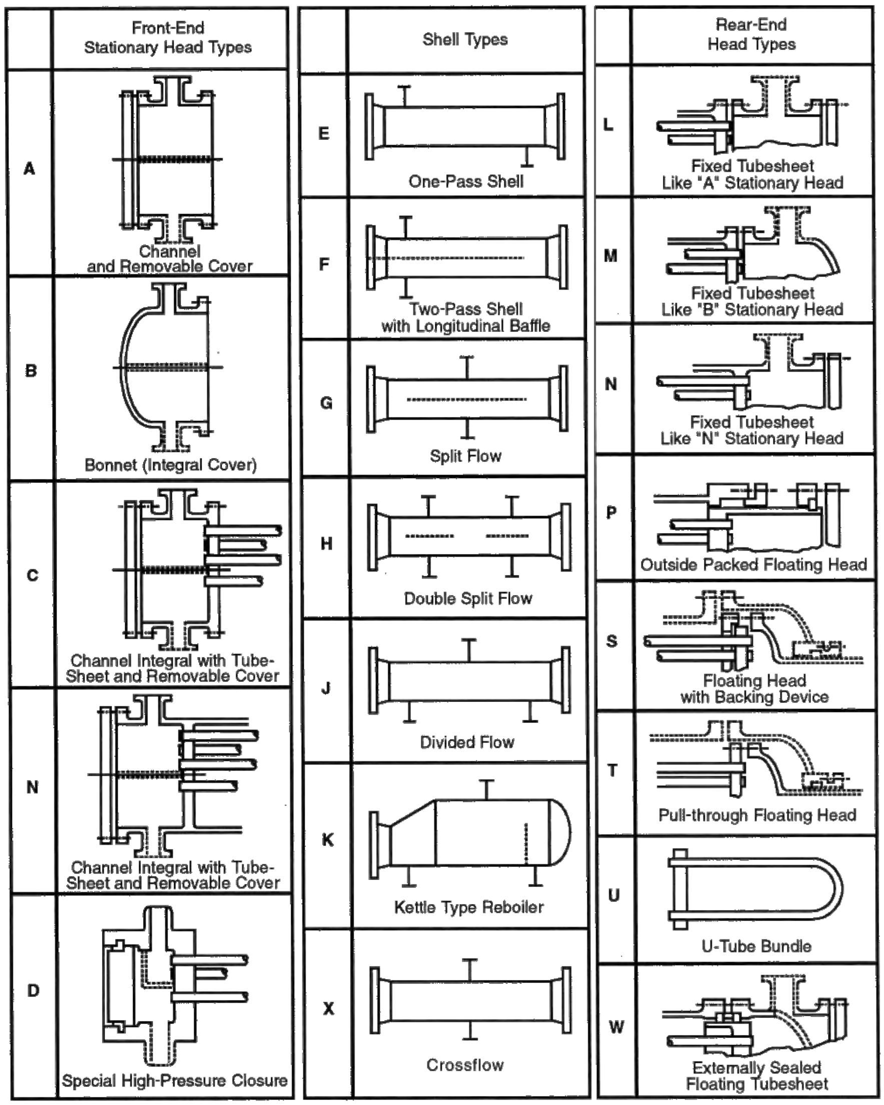
The three most common types of shell-and-tube exchangers are (1) fixed tubesheet design, (2) U-tube design, and (3) floating-head type. In all three types, the front-end head is stationary while the rear-end head can be either stationary or floating, depending on the thermal stresses in the shell, tube, or tubesheet, due to temperature differences as a result of heat transfer.

The exchangers are built in accordance with three mechanical standards that specify design, fabrication, and materials of unfired shell-and-tube heat exchangers. Class R is for the generally severe requirements of petroleum and related processing applications. Class C is for generally moderate requirements for commercial and general process applications. Class B is for chemical process service. The exchangers are built to comply with the applicable ASME Boiler and Pressure Vessel Code, Section VIII (1998), and other pertinent codes and/or standards. The TEMA standards supplement and define the ASME code for heat exchanger applications. In addition, state and local codes applicable to the plant location must also be met.

The TEMA standards specify the manufacturing tolerances for various mechanical classes, the range of tube sizes and pitches, baffling and support plates, pressure classification, tubesheet thickness formulas, and so on, and must be consulted for all these details. In this book, we consider only the TEMA standards where appropriate, but there are other standards, such as DIN 28 008.

Tubular exchangers are widely used in industry for the following reasons. They are custom designed for virtually any capacity and operating conditions, such as from high

FIGURE 1.6 Standard shell types and front- and rear-end head types (From TEMA, 1999).



vacuum to ultrahigh pressure [over 100 MPa (15,000 psig)], from cryogenics to high temperatures [about 11008C (20008F)] and any temperature and pressure differences between the fluids, limited only by the materials of construction. They can be designed for special operating conditions: vibration, heavy fouling, highly viscous fluids, erosion, corrosion, toxicity, radioactivity, multicomponent mixtures, and so on. They are the most versatile exchangers, made from a variety of metal and nonmetal materials (such as graphite, glass, and Teflon) and range in size from small [0.1 m^2 (1 ft^2)] to supergiant [over 10^5 m^2 (10^6 ft^2)] surface area. They are used extensively as process heat exchangers in the petroleum-refining and chemical industries; as steam generators, condensers, boiler feedwater heaters, and oil coolers in power plants; as condensers and evaporators in some air-conditioning and refrigeration applications; in waste heat recovery applications with heat recovery from liquids and condensing fluids; and in environmental control.

Round tubes in various shapes are used in shell-and-tube exchangers. Most common are the tube bundles[{] with straight and U-tubes used in process and power industry exchangers. However, sine-wave bend, J-shape, L-shape or hockey sticks, and inverted hockey sticks are used in advanced nuclear exchangers to accom-modate large thermal expansion of the tubes. Some of the enhanced tube geometries used in shell-and-tube exchangers are shown in Fig1.7. Serpentine, helical, and bay-onet are other tube shapes that are used in shell-and-tube exchan-gers. In most applications, tubes have single walls, but when working with radioactive,

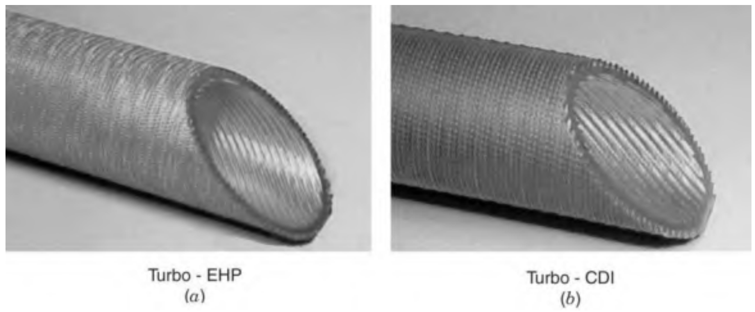


FIGURE 1.7 Some enhanced tube geometries used in shell-and-tube exchangers: (a) internally and externally enhanced evaporator tube; (b) internally and externally enhanced condenser tube. (Courtesy of Wolverine Tube, Inc., Decatur, AL.)

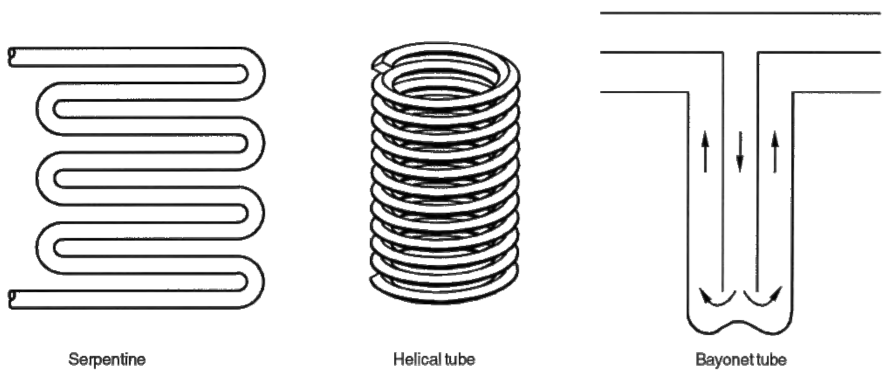


FIGURE 1.8 Additional tube configurations used in shell-and-tube exchangers.

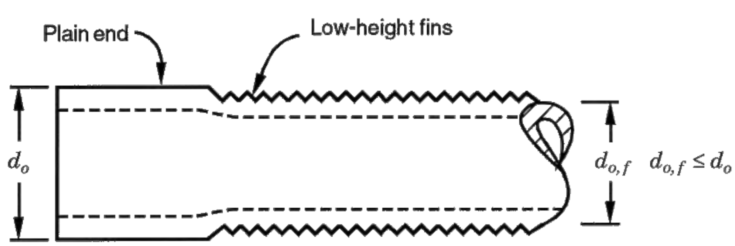


FIGURE 1.9 Low-finned tubing. The plain end goes into the tubesheet.

reactive, or toxic fluids and potable water, double-wall tubing is used. In most applica-tions, tubes are bare, but when gas or low-heat-transfer coefficient liquid is used on the shell side, low-height fins (low fins) are used on the shell side. Also, special high-flux-boiling surfaces employ modified low-fin tubing. These are usually integral fins made from a thick-walled tube, shown in Fig. 1.9. Tubes are drawn, extruded, or welded, and they are made from metals, plastics, and ceramics, depending on the applications.[3]

1.3.3.Shells:

The shell is a container for the shell fluid. Usually, it is cylindrical in shape with a circular cross section, although shells of different shapes are used in specific applica-tions and in nuclear heat exchangers to conform to the tube bundle shape. The shell is made from a circular pipe if the shell diameter is less than about 0.6 m (2 ft) and is made from a metal plate rolled and welded longitudinally for shell diameters greater than 0.6 m (2 ft). Seven types of shell configurations, standardized by TEMA (1999), are E, F, G, H, J, K, and X, shown in Fig. The E shell is the most common, due to its low cost and simplicity, and has the highest log-mean temperature-difference correction factor F. Although the tubes may have single or multiple passes, there is one pass on the shell side. To increase the mean temperature difference and hence exchanger effectiveness, a pure counterflow arrangement is desir-able for a two-tube-pass exchanger. This is achieved by use of an F shell having a longitudinal baffle and resulting in two shell passes. Split- and divided-flow shells, such as G, H, and J (see Fig. 1.6), are used for specific applications, such as thermosiphon boiler, condenser, and shell-side low pressure drops. The K shell is a kettle reboiler used for pool boiling applications. The X shell is a crossflow exchanger and is used for low pressure drop on the shell side and/or to eliminate the possibility of flow-induced vibrations. [4]

1.3.4.Nozzles:

The entrance and exit ports for the shell and tube fluids, referred to as nozzles, are pipes of constant cross section welded to the shell and channels. They are used to distribute or collect the fluid uniformly on the shell and tube sides. Note that they differ from the nozzle used as a fluid metering device or in jet engines, which has a variable flow area along the flow length.[4]

1.3.5.Front- and Rear-End Heads:

These are used for entrance and exit of the tube fluid; in many rear-end heads, a provision has been made to take care of tube thermal expan-sion. The front-end head is stationary, while the rear-end head could be either station-ary (allowing for no tube thermal expansion) or floating, depending on the thermal stresses between the tubes and shell. The major criteria for selection of the front-end head are cost, maintenance and inspection, hazard due to mixing of shell and tube fluids, and leakage to ambient and operating pressures. The major criteria for selection of the rear-end head are the allowance for thermal stresses, a provision to remove the tube bundle for cleaning the shell side, prevention of mixing of tube and shell fluids, and sealing any leakage path for the shell fluid to ambient. The design selection criteria for the front- and rear-end heads of Fig. 1.10[4]

1.3.6.Baffles:

Baffles may be classified as transverse and longitudinal types. The purpose of longitudinal baffles is to control the overall flow direction of the shell fluid such that a desired overall flow arrangement of the two fluid streams is achieved. For example, F, G, and H shells have longitudinal baffles. Transverse baffles may be classified as plate baffles and grid (rod, strip, and other axial-flow) baffles. Plate baffles are used to support the tubes during assembly and operation and to direct the fluid in the tube bundle approximately at right angles to the tubes to achieve higher heat transfer coefficients. Plate baffles increase the turbulence of the shell fluid and minimize tube-to-tube temperature differences and thermal stresses due to the crossflow. Shown in Fig. are single- and multisegmental baffles and disk and doughnut baffles. Single- and double-segmental baffles are used most frequently due to their ability to assist maximum heat transfer (due to a high-shell-side heat transfer coefficient) for a given pressure drop in a minimum amount of space. Triple and no-tubes-in-window segmental baffles are used for low-pressure-drop applications. The choice of baffle type, spacing, and cut is determined largely by flow rate, desired heat transfer rate, allowable pressure drop, tube support, and flow-induced vibrations. Disk and doughnut baffles/ support plates are used primarily in nuclear heat exchangers. These baffles for nuclear exchangers have small perforations between tube holes to allow a combination of crossflow and longitudinal flow for lower shell-side pressure drop. The combined flow results in a slightly higher heat transfer coefficient than that for pure longitudinal flow and minimizes tube-to-tube temperature differences. Rod (or bar) baffles, the most common type of grid baffle, used to support the tubes and increase the turbulence of the shell fluid, are shown in Fig. The flow in a rod baffle heat exchanger is parallel to the tubes, and flow-induced vibrations are virtually eliminated by the baffle support of the tubes. One alternative to a rod baffle heat exchanger is the use of twisted tubes (after flattening the circular tubes, they are twisted), shown in Fig. Twisted tubes pro-vide rigidity and eliminate flow-induced tube vibrations, can be cleaned easily on the shell side with hydrojets, and can be cleaned easily inside the tubes, but cannot be retubed. Low-finned tubes are also available in a twisted-tube configuration. A helical baffle shell-and-tube exchanger with baffles as shown in Fig. also has the following advantages: a lower shell-side pressure drop while maintaining the high heat transfer coefficient of a segmental exchanger, reduced leakage streams, and elimination of dead spots and recirculation zones (thus reducing fouling). Every shell-and-tube exchanger has transverse baffles except for X and K shells, which have support plates because the sole purpose of these transverse baffles is to support the tubes. [5]

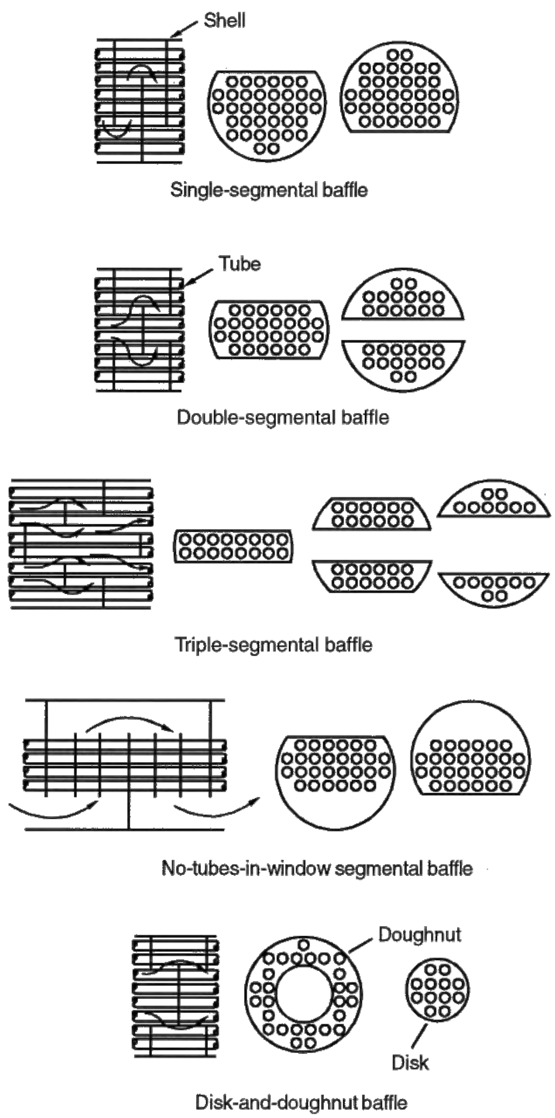


FIGURE 1.10 Plate baffle types, modified from Mueller (1973).

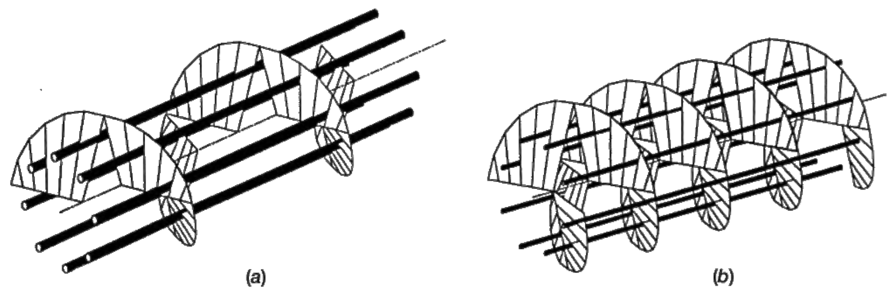


FIGURE 1.11 Helical baffle shell-and-tube exchanger: (a) single helix; (b) double helix. (Courtesy of ABB Lumus Heat Transfer, Bloomfield, NJ.)

1.3.7.Double-Pipe Heat Exchangers:

This exchanger usually consists of two con-centric pipes with the inner pipe plain or finned, as shown in Fig. One fluid flows in the inner pipe and the other fluid flows in the annulus between pipes in a counterflow direction for the ideal highest performance for the given surface area. However, if the application requires an almost constant wall temperature, the fluids may flow in a parallelflow direction. This is perhaps the simplest heat exchanger. Flow distribution is no problem, and cleaning is done very easily by disassembly. This con-figuration is also suitable where one or both of the fluids is at very high pressure,

Comparison of Various Types of Shell-and-Tube Heat Exchangers

Characteristic	Segmental Baffle	Rod Baffle	Twisted Tube	Helical Baffle
Good heat transfer per unit pressure drop	No	Yes	Yes	Yes
High shell-side heat transfer coefficient	Yes	No	No	Yes
Tube-side enhancement	With inserts	With inserts	Included	With inserts
Suitable for very high exchanger effectiveness	No	Yes	Yes	No
Tends to have low fouling	No	Yes	Yes	Yes
Can be cleaned mechanically	Yes, with square pitch	Yes	Yes	Yes, with square pitch
Low flow-induced tube vibration	With special designs	Yes	Yes	With double helix
Can have low-finned tubes	Yes	Yes	Yes	Yes

because containment in the small-diameter pipe or tubing is less costly than containment in a large-diameter shell. Double-pipe exchangers are generally used for small-capacity applications where the total heat transfer surface area required is 50 m^2 (500 ft^2) or less because it is expensive on a cost per unit surface area basis. Stacks of double-pipe or multitube heat exchangers are also used in some process applications with radial or longitudinal fins. The exchanger with a bundle of U tubes in a pipe (shell) of 150 mm (6 in.) diameter and above uses segmental baffles and is referred to variously as a hairpin or jacketed U-tube exchanger.[6]

<u>1.3.8.Spiral Tube Heat Exchangers:</u>

These consist of one or more spirally wound coils fitted in a shell. Heat transfer rate associated with a spiral tube is higher than that for a straight tube. In addition, a considerable amount of surface can be accommodated in a given space by spiraling. Thermal expansion is no problem, but cleaning is almost impossible.[6]

1.3.9.Plate-Type Heat Exchangers

Plate-type heat exchangers are usually built of thin plates (all prime surface). The plates are either smooth or have some form of corrugation, and they are either flat or wound in an exchanger. Generally, these exchangers cannot accommodate very high pressures, temperatures, or pressure and temperature differences. Plate heat exchangers (PHEs) can be classified as gasketed, welded (one or both fluid passages), or brazed, depending on the leak tightness required. Other plate-type exchangers are spiral plate, lamella, and platecoil exchangers. [6]

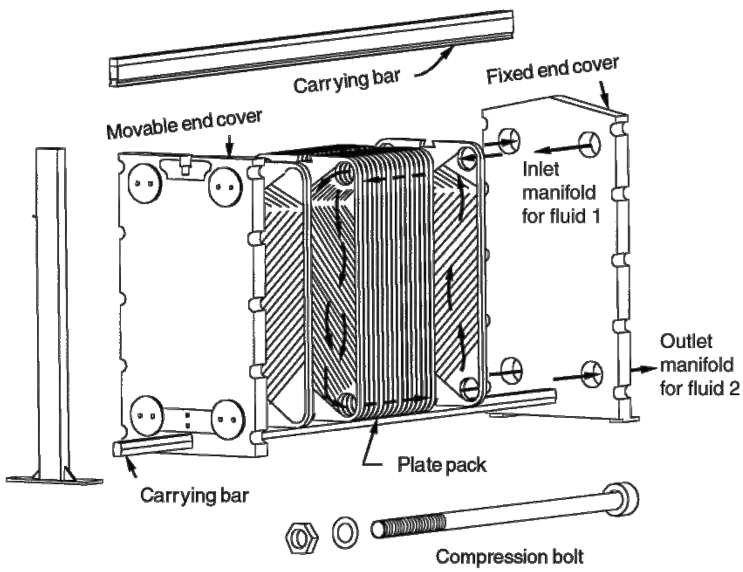
1.3.10.Gasketed Plate Heat Exchangers

1.3.10.1.Basic Construction:

The plate-and-frame or gasketed plate heat exchanger (PHE) con-sists of a number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame as shown in Fig. The frame usually has a fixed end cover (headpiece) fitted with connecting ports and a movable end cover (pressure plate, follower, or tailpiece). In the frame, the plates are suspended from an upper carrying bar and guided by a bottom carrying bar to ensure proper alignment. For this purpose, each plate is notched at the center of its top and bottom edges. The plate pack with fixed and movable end covers is clamped together by long bolts, thus compressing the gaskets and forming a seal. For later discussion, we designate the resulting length of the plate pack as L_{pack}. The carrying bars are longer than the compressed stack, so that when the movable end cover is removed, plates may be slid along the support bars for inspection and cleaning.

Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on sheet metal. On one side of each plate, special grooves are provided along the per-iphery of the plate and around the ports for a gasket, as indicated by the dark lines in Fig. Typical plate geometries (corrugated patterns) are shown in Fig., and over 60 different patterns have been developed worldwide. Alternate plates are assembled such

FIGURE 1.12 Gasketed Plate-and-Frame Heat Exchanger



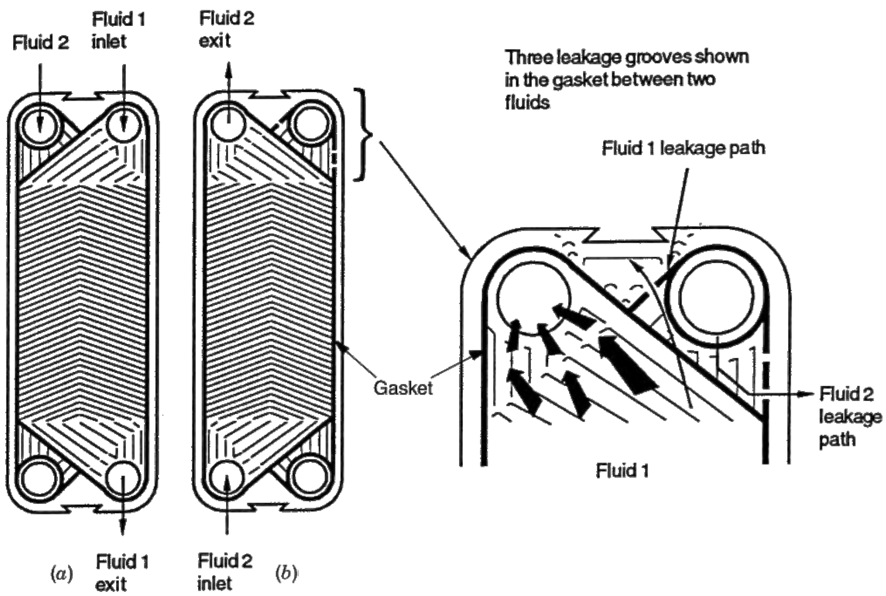


FIGURE 1.13 Plates showing gaskets around the ports (Shah and Focke, 1988).

that the corrugations on successive plates contact or cross each other to provide mechan-ical support to the plate pack through a large number of contact points. The resulting flow passages are narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing. Plates are designated as hard or soft, depending on whether they generate a high or low intensity of turbulence.

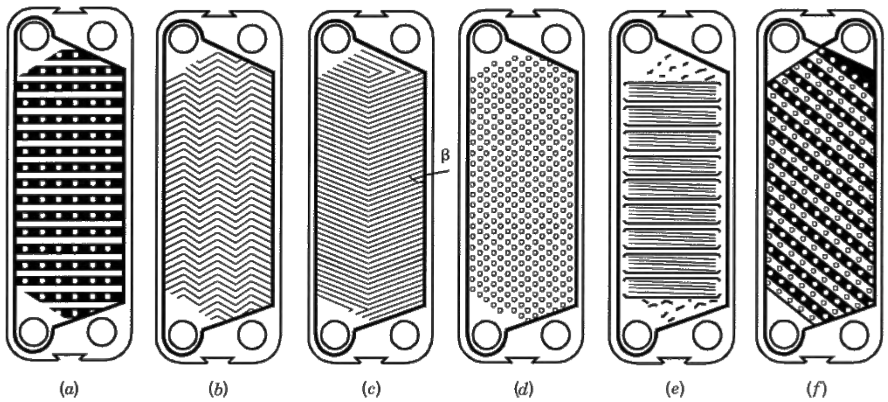


Figure 1.14 Plate patterns: (a) washboard; (b) zigzag; (c) chevron or herringbone; (d) protrusions and depressions; (e) washboard with secondary corrugations; (f) oblique washboard (Shah and Focke, 1988).

Sealing between the two fluids is accomplished by elastomeric molded gaskets [typically, 5 mm (0.2 in.) thick] that are fitted in peripheral grooves mentioned. Gaskets are designed such that they compress about 25% of thickness in a bolted plate exchanger to provide a leaktight joint without distorting the thin plates. In the past, the gaskets were cemented in the grooves, but now, snap-on gaskets, which do not require cementing, are common. Some manufacturers offer special interlocking types to prevent gasket blowout at high pressure differences. Use of a double seal around the port sections, shown in Fig., prevents fluid intermixing in the rare event of gasket failure. The interspace between the seals is also vented to the atmosphere to facilitate visual indication of leakage. Typical gasket materials and their range of applications are listed in Table, with butyl and nitrile rubber being most common. PTFE (polytetrafluoroethylene) is not used because of its viscoelastic proper-ties.

Each plate has four corner ports. In pairs, they provide access to the flow passages on either side of the plate. When the plates are assembled, the corner ports line up to form distribution headers for the two fluids. Inlet and outlet nozzles for the fluids, provided in the end covers, line up with the ports in the plates (distribution headers) and are con-nected to external piping carrying the two fluids. A fluid enters at a corner of one end of the compressed stack of plates through the inlet nozzle. It passes through alternate channels[{] in either series or parallel passages. In one set of channels, the gasket does not surround the inlet port between two plates; fluid enters through that port, flows between plates, and exits through a port at the other end. On the same side of the plates, the other two ports are blocked by a gasket with a double seal, as shown in Fig.a, so that the other fluid cannot enter the plate on that side.[{] In a 1 pass–1 pass[}] two-fluid counterflow PHE, the next channel has gaskets covering the ports just opposite the preceding plate. Incidentally, each plate has gaskets on only one side, and they sit in grooves on the back side of the neighboring plate. In Fig., each fluid makes a single pass through the exchanger because of alternate gasketed and ungasketed ports in each corner opening. The most conventional flow arrangement is 1 pass–1 pass counterflow, with all inlet and outlet connections on the fixed end cover. By blocking flow through some ports with proper gasketing, either one or both fluids could have more than one pass. Also, more than one exchanger can be accommodated in a single frame. In cases with more than two simple 1-pass–1-pass heat exchangers, it is necessary to insert one or more intermediate headers or connector plates in the plate pack at appropriate places. In milk pasteurization applications, there are as many as five exchangers or sections to heat, cool, and regen-erate heat between raw milk and pasteurized milk.

Typical plate heat exchanger dimensions and performance parameters are given in Table. Any metal that can be cold-worked is suitable for PHE applications. The most

TABLE 1.2 Gasket Materials Used in Plate Heat Exchangers

Gasket Material	Generic Name	Maximum Operating Temperature (8C)	Applications	Comments
Natural rubber	cis-1,4-polyisoprene	70	Oxygenated solvents, acids, alcohols	
SBR (styrene butadiene)		80	General-purpose aqueous, alkalies, acids, and oxygenated solvents	Has poor fat resistance
Neoprene	trans-1,4-polychloroprene	70	Alcohols, alkalies, acids, aliphatic hydrocarbon solvents	
Nitrile		100–140	Dairy, fruit juices, beverage, pharmaceutical and biochemical applications, oil, gasoline, animal and vegetable oils, alkalies, aliphatic organic solvents	Is resistant to fatty materials; particularly suitable for cream
Butyl (resin cured)		120–150	Alkalies, acids, animal and vegetable oils, aldehydes, ketones, phenols, and some esters	Has poor fat resistance; suitable for UHT milk duties; resists inorganic chemical solutions up to 1508C
Ethylene propylene (EDPM) rubber		140	Alkalies, oxygenated solvents	Unsuitable for fatty liquids
Silicone rubber	Polydimethyl-siloxane	140	General low-temperature use, alcohols, sodium hypochlorite	
Fluorinated rubber		175	High-temperature aqueous solutions, mineral oils and gasoline, organic solvents, animal and vegetable oils	
Compressed asbestos fiber		200–260	Organic solvents, high-operating-temperature applications	

common plate materials are stainless steel (AISI 304 or 316) and titanium. Plates made from Incoloy 825, Inconel 625, and Hastelloy C-276 are also available. Nickel, cupro-nickel, and monel are rarely used. Carbon steel is not used, due to low corrosion resis-tance for thin plates.

Graphite and polymer plates are used with corrosive fluids. The heat transfer surface area per unit volume for plate exchangers ranges from 120 to 660 m^2/m^3 (37 to 200 ft^2/ft^3).

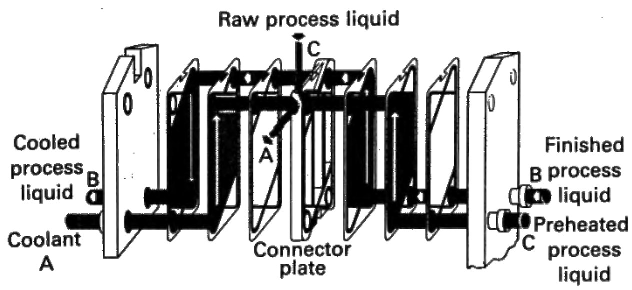


Figure 1.15 A three-fluid plate heat exchanger. (Courtesy of Alfa Laval Thermal, Inc., Lund, Sweden.)

Flow Arrangements. A large number of flow arrangements are possible in a plate heat exchanger, depending on the required heat transfer duty, available pressure drops, minimum and maximum velocities allowed, and the flow rate ratio of the two fluid streams. In each pass there can be an equal or unequal number of thermal plates. Whether the plate exchanger is a single- or multipass unit, whenever possible, the thermodynamically superior counterflow or overall counterflow arrange-ment is used exclusively.

One of the most common flow arrangements in a PHE is a 1-pass–1-pass U config-uration. This is because this design allows all fluid ports to be located on the fixed end cover,

Table 1.3 Some Geometrical and Operating Condition Characteristics of Plate-and-Frame Heat Exchangers

Unit		Operation	
Maximum surface area	2500 m^2	Pressure	0.1 to 3.0 MPa
Number of plates	3 to 700	Temperature	–40 to 2608C
Port size	Up to 400 mm (for liquids)	Maximum port velocity	6 m/s (for liquids)
		Channel flow rates	0.05 to 12.5 m^3/h
		Maximum unit flow rate	2500 m^3/h
Plates		**Performance**	
Thickness	0.5 to 1.2 mm	Temperature approach	As low as 18C
Size	0.03 to 3.6 m^2	Heat exchanger efficiency	Up to 93%
Spacing	1.5 to 7 mm	Heat transfer coefficients for water–water duties	3000 to 8000 $W/m^2 \cdot K$
Width	70 to 1200 mm		
Length	0.4 to 5 m		
Hydraulic diameter	2 to 10 mm		
Surface area per plate	0.02 to 5 m^2		

permitting easy disassembly and cleaning/repair of a PHE without disconnecting any piping. In a multipass arrangement, the ports and fluid connections are located on both fixed and movable end covers. A multipass arrangement is generally used when the flow rates are considerably different or when one would like to use up the available pressure drop by multipassing and hence getting a higher heat transfer coefficient.[6]

<u>1.3.10.2.Advantages and Limitations:</u>

Some advantages of plate heat exchangers are as follows. They can easily be taken apart into their individual components for cleaning, inspec-tion, and maintenance. The heat transfer surface area can readily be changed or re-arranged for a different task or for anticipated changing loads, through the flexibility of plate size, corrugation patterns, and pass arrangements. High shear rates and shear stresses, secondary flow, high turbulence, and mixing due to plate corrugation patterns reduce fouling to about 10 to 25% of that of a shell-and-tube exchanger, and enhance heat transfer. Very high heat transfer coefficients are achieved due to the breakup and reattachment of boundary layers, swirl or vortex flow generation, and small hydraulic diameter flow passages. Because of high heat transfer coefficients, reduced fouling, the absence of bypass and leakage streams, and pure counterflow arrangements, the surface area required for a plate exchanger is one-half to one-third that of a shell-and-tube exchanger for a given heat duty, thus reducing the cost, overall volume, and space requirement for the exchanger. Also, the gross weight of a plate exchanger is about one-sixth that of an equivalent shell-and-tube exchanger. Leakage from one fluid to the other cannot take place unless a plate develops a hole. Since the gasket is between the plates, any leakage from the gaskets is to the outside of the exchanger. The residence time (time to travel from the inlet to the outlet of the exchanger) for different fluid particles or flow paths on a given side is approximately the same. This parity is desir-able for uniformity of heat treatment in applications such as sterilizing, pasteurizing, and cooking. There are no significant hot or cold spots in the exchanger that could lead to the deterioration of heat-sensitive fluids. The volume of fluid held up in the exchan-ger is small; this feature is important with expensive fluids, for faster transient response, and for better process control. Finally, high thermal performance can be achieved in plate exchangers. The high degree of counterflow in PHEs makes temperature approaches of up to 18C (28F) possible. The high thermal effectiveness (up to about 93%) facilitates economical low-grade heat recovery. The flow-induced vibrations, noise, thermal stresses, and entry impingement problems of shell-and-tube exchangers do not exist for plate heat exchangers.

Some inherent limitations of the plate heat exchangers are caused by plates and gaskets as follows. The plate exchanger is capable of handling up to a maximum pressure of about 3 MPa gauge (435 psig) but is usually operated below 1.0 MPa gauge (150 psig). The gasket materials (except for the PTFE-coated type) restrict the use of PHEs in highly corrosive applications; they also limit the maximum operating temperature to 2608C (5008F) but are usually operated below 1508C (3008F) to avoid the use of expensive gasket materials. Gasket life is sometimes limited. Frequent gasket replacement may be needed in some applications. Pinhole leaks are hard to detect. For equivalent flow velocities, pressure drop in a plate exchanger is very high compared to that of a shell-and-tube exchanger. However, the flow velocities are usually low and plate lengths are ‘‘short,’’ so the resulting pressure drops are generally acceptable. The normal symmetry of PHEs may make phase-change applications[{] more difficult, due to large differences in volumetric flows. For some cases, heat exchanger duties with widely different fluid flow rates and depending on the allowed pressure drops of the two fluids, an arrangement of a different number of passes for the two fluids may make a PHE advantageous. However, care must be exercised to take full advantage of available pressure drop while multi-passing one or both fluids.

Because of the long gasket periphery, PHEs are not suited for high-vacuum applica-tions. PHEs are not suitable for erosive duties or for fluids containing fibrous materials. In certain cases, suspensions can be handled; but to avoid clogging, the largest suspended particle should be at most one-third the size of the average channel gap. Viscous fluids can be handled, but extremely viscous fluids lead to flow maldistribution problems, especially on cooling. Plate exchangers should not be used for toxic fluids, due to poten-tial gasket leakage. Some of the largest units have a total surface area of about 2500 m^2 (27,000 ft^2) per frame. Some of the limitations of gasketed PHEs have been addressed by the new designs of PHEs described in the next subsection.

Major Applications. Plate heat exchangers were introduced in 1923 for milk pasteur-ization applications and now find major applications in liquid–liquid (viscosities up to 10 Pa _ s) heat transfer duties. They are most common in the dairy, juice, beverage, alcoholic drink, general food processing, and pharmaceutical industries, where their ease of cleaning and the thermal control required for sterilization/pasteurization make them ideal. They are also used in the synthetic rubber industry, paper mills, and in the process heaters, coolers, and closed-circuit cooling systems of large petrochemical and power plants. Here heat rejection to seawater or brackish water is common in many applications, and titanium plates are then used.

Plate heat exchangers are not well suited for lower-density gas-to-gas applications. They are used for condensation or evaporation of non-low-vapor densities. Lower vapor densities limit evaporation to lower outlet vapor fractions. Specially designed plates are now available for condensing as well as evaporation of high-density vapors such as ammonia, propylene, and other common refrigerants, as well as for combined evapora-tion/condensation duties, also at fairly low vapor densities.[6]

1.4 OBJECTIVE

The main objective of this project is designing and simulation of shell and tube heat exchanger with helical baffle using Solidworks tools.

Chapter 2

Literature Review

2.LITERATURE REVIEW

2.1 Introduction

The purpose of this chapter is to provide a literature review of past research effort such as journals or articles related to shell and tube heat exchanger and computational fluid dynamics (CFD) analysis whether on two dimension and three dimension modelling. Moreover, review of other relevant research studies are made to provide more information in order to understand more on this research.

2.2 Purpose of Use of Helical Baffle:

A new type of baffle, called the helical baffle, provides further improvement. This type of baffle was first developed by Lutcha and Nemcansky. They investigated the flow field patterns produced by such helical baffle geometry with different helix angles. They found that these flow patterns were very close to the plug flow condition, which was expected to reduce shell-side pressure drop and to improve heat transfer performance. Stehlik et al. compared heat transfer and pressure drop correction factors for a heat exchanger with an optimized segmental baffle based on the Bell–Delaware method, with those for a heat exchanger with helical baffles. Kral et al. discussed the performance of heat exchangers with helical baffles based on test results of various baffles geometries. One of the most important Geometric factors of the STHXHB is the helix angle. Recently a comprehensive comparison between the test data of shell-side heat transfer coefficient versus shell-side pressure drop was provided for five helical baffles and one segmental baffle measured for oil-water heat exchanger. It is found that based on the heat transfer per unit shell-side fluid pumping power or unit shell-side fluid pressured drop, the case of 40^0 helix angle behaves the best. The flow pattern in the shell side of the heat exchanger with continuous helical baffles was forced to be rotational and helical due to the geometry of the continuous helical baffles, which results in a significant increase in heat transfer coefficient per unit pressure drop in the heat exchanger. Properly designed continuous helical baffles can reduce fouling in the shell side and prevent the flow-induced vibration as well. The performance of the proposed STHXs was studied experimentally in this work. The use of continuous helical baffles results in nearly 10% increase in heat transfer coefficient compared with that of conventional segmental baffles for the same shell-side pressure drop. Based on the experimental data, the non dimensional correlations for heat transfer coefficient and pressure drop were developed for the proposed continuous helical baffle heat exchangers with different shell configurations, which might be useful for industrial applications and further study of continuous helical baffle heat exchangers.[7]

2.3 Computational Fluid Dynamics (CFD):

CFD is a sophisticated computationally-based design and analysis technique. CFD software gives you the power to simulate flows of gases and liquids, heat and mass transfer, moving bodies, multiphase physics, chemical reaction, fluid-structure interaction and acoustics through computer modelling. This software can also build a virtual prototype of the system or device before can be apply to real-world physics and chemistry to the model, and the software will provide with images and data, which predict the performance of that design. Computational fluid dynamics (CFD) is useful in a wide variety of applications and use in industry. CFD is one of the branches of fluid mechanics that uses numerical methods and algorithm can be used to solve and analyse problems that involve fluid flows and also simulate the flow over a piping, vehicle or machinery. Computers are used to perform the millions of calculations required to simulate the interaction of fluids and gases with the complex surfaces used in engineering. More accurate codes that can accurately and quickly simulate even complex scenarios such as supersonic and turbulent flows are on going research. Onwards the aerospace industry has integrated CFD techniques into the design, R & D and manufacture of aircraft and jet engines. More recently the methods have been applied to the design of internal combustion engine, combustion chambers of gas turbine and furnaces also fluid flows and heat transfer in heat exchanger (Figure 1). Furthermore, motor vehicle manufactures now routinely predict drag forces, underbonnet air flows and surrounding car environment with CFD. Increasingly CFD is becoming a vital component in the design of industrial products and processes.[8]

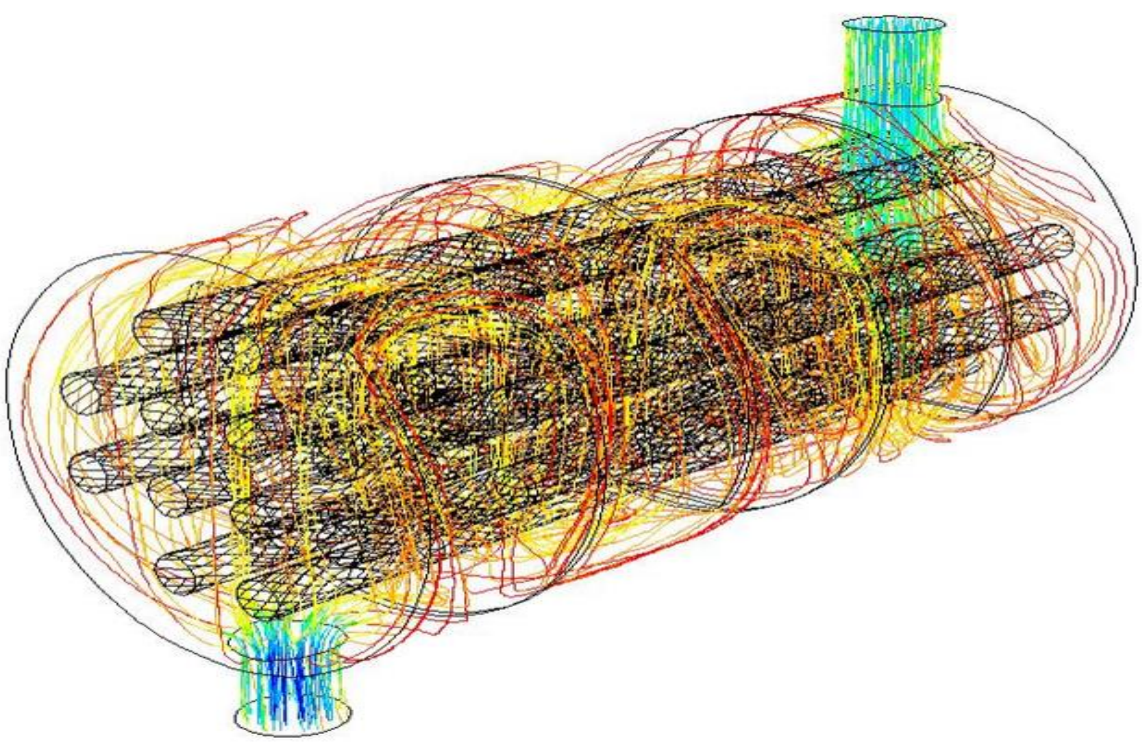


Figure 1.16 Fluid flow simulation for a shell and tube exchanger.

2.4 APPLICATION OF CFD:

CFD not just spans on chemical industry, but a wide range of industrial and nonindustrial application areas which is in below :[8]

- Aerodynamics of aircraft and vehicle.
- Combustion in IC engines and gas turbine in power plant.
- Loads on offshore structure in marine engineering.
- Blood flows through arteries and vein in biomedical engineering.
- Weather prediction in meteorology.
- Flow inside rotating passages and diffusers in turbo-machinery.
- External and internal environment of buildings like wind loading and heating or Ventilation system.
- Mixing and separation or polymer moldings in chemical process engineering.
- Distribution of pollutants and effluent in environmental engineering.

2.5 SOLIDWORKS CFD:

SOLIDWORKS Flow Simulation uses Computational Fluid Dynamics (CFD) analysis to enable quick, efficient simulation of fluid flow and heat transfer. You can easily calculate fluid forces and understand the impact of a liquid or gas on product performance.

Tightly integrated with SOLIDWORKS CAD, CFD analysis using SOLIDWORKS Simulation takes the complexity out of flow analysis and can be a regular part of your design process, reducing the need for costly prototypes, eliminating rework and delays, and saving time and development costs.

CFD simulates fluid (either liquid or gas) passing through or around an object. The analysis can be very complex—for example, containing in one calculation heat transfer, mixing, and unsteady and compressible flows. The ability to predict the impact of such flows on your product performance is time consuming and costly without some form of simulation tool.

SOLIDWORKS Flow Simulation offers a wide range of physical models and fluid flow capabilities so you can obtain better insight into product behavior that is critical to your design success covering a broad range of applications:

- Liquid and gas flow with heat transfer
- External and internal fluid flows
- Laminar, turbulent, and transitional flows
- Time-dependent flow
- Subsonic, transonic, and supersonic regimes
- Gas mixture, liquid mixture
- Conjugate heat transfer
- Heat transfer in solids
- Incompressible and compressible liquid
- Compressible gas
- Real gases
- Water vapor (steam)
- Non-Newtonian liquids (to simulate blood, honey, molten plastics)

Engineers across a wide range of industries can benefit from CFD—such as automotive, aerospace, defense, life science, machinery, and high tech. Indeed, almost every design encounters fluid dynamics at some point, whether heat or liquids, internal or external.[9]

Chapter 3

Computational Model for Heat Exchanger

3. COMPUTATIONAL MODEL FOR HEAT EXCHANGER

3.1 Problem Description:

Design of shell and tube heat exchanger with helical baffle using CFD.

3.2 Computational Model:

The computational model of an experimental tested Shell and Tube Heat Exchanger(STHX) with 0 helix angle is shown in figure, and the geometry parameters are listed in Table. As can be seen from figure ,the simulated STHX has six cycles of baffles in the shell side direction with total number of tube 7 .The whole computation domain is bounded by the inner side of the shell and everything in the shell contained in the domain. The inlet and out let of the domain are connected with the corresponding tubes.

To simplify numerical simulation, some basic characteristics of the process following assumption are made :

1. The shell side fluid is constant thermal properties
2. The fluid flow and heat transfer processes are turbulent and in steady state
3. The leak flows between tube and baffle and that between baffles and shell are neglected
4. The natural convection induced by the fluid density variation is neglected
5. The tube wall temperature kept constant in the whole shell side
6. The heat exchanger is well insulated hence the heat loss to the environment is totally neglected .

3.3 Navier-Stokes Equation:

It is named after Claude-Louis Navier and Gabriel Stokes , He described the motion of fluid substances. These equation arise from applying second law of newton to fluid motion, together with the assumption that the fluid stress is sum of a diffusing viscous term ,plus a pressure term. The derivation of the Navier Stokes equation begins with an application of second law of newton i.e conservation of momentum. In an inertial frame of reference, the general form of the equations of fluid motion is :-[10]

Coordinates: (x,y,z)
Velocity Components: (u,v,w)
Time : t
Density: ρ
Total Energy: Et
Pressure: p
Stress: τ
Heat Flux: q
Reynolds Number: Re
Prandtl Number: Pr

Continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0$$

X – Momentum:

$$\frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u^2)}{\partial x} + \frac{\partial (\rho u v)}{\partial y} + \frac{\partial (\rho u w)}{\partial z} = -\frac{\partial p}{\partial x} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z}\right]$$

Y – Momentum:

$$\frac{\partial (\rho v)}{\partial t} + \frac{\partial (\rho u v)}{\partial x} + \frac{\partial (\rho v^2)}{\partial y} + \frac{\partial (\rho v w)}{\partial z} = -\frac{\partial p}{\partial y} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z}\right]$$

Z – Momentum

$$\frac{\partial (\rho w)}{\partial t} + \frac{\partial (\rho u w)}{\partial x} + \frac{\partial (\rho v w)}{\partial y} + \frac{\partial (\rho w^2)}{\partial z} = -\frac{\partial p}{\partial z} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z}\right]$$

Energy:

$$\frac{\partial (E_T)}{\partial t} + \frac{\partial (u E_T)}{\partial x} + \frac{\partial (v E_T)}{\partial y} + \frac{\partial (w E_T)}{\partial z} = -\frac{\partial (up)}{\partial x} - \frac{\partial (vp)}{\partial y} - \frac{\partial (wp)}{\partial z} - \frac{1}{Re_r Pr_r}\left[\frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y} + \frac{\partial q_z}{\partial z}\right]$$
$$+ \frac{1}{Re_r}\left[\frac{\partial}{\partial x}(u\tau_{xx} + v\tau_{xy} + w\tau_{xz}) + \frac{\partial}{\partial y}(u\tau_{xy} + v\tau_{yy} + w\tau_{yz}) + \frac{\partial}{\partial z}(u\tau_{xz} + v\tau_{yz} + w\tau_{zz})\right]$$

3.4 Geometry of Heat Exchanger:

The model is designed according to TEMA (Tubular Exchanger Manufacturers Association) Standards Gaddis (2007).[11]

Shell-and-tube heat exchangers in various sizes are widely used in industrial operations and energy conversion systems. Tubular Exchanger Manufacturers Association (TEMA) regularly publishes standards and design recommendations (9th edition is published in 2007 [10]). Shell-and-tube heat exchangers have been very successfully designed according to TEMA standards and using recommended correlation based analytical approaches. These approaches have constantly improved since the early days due to accumulating industrial experience and operational data, and improving instrumentation. The correlation based approaches can be used for sizing and can also be used iteratively to obtain general performance parameters (rating) of a heat exchanger. At a given iteration, if the performance of the considered design is calculated to be unsatisfactory, a better performing design can be obtained by changing the design parameters in the right direction. An experienced heat exchanger designer knows what to change in which direction.

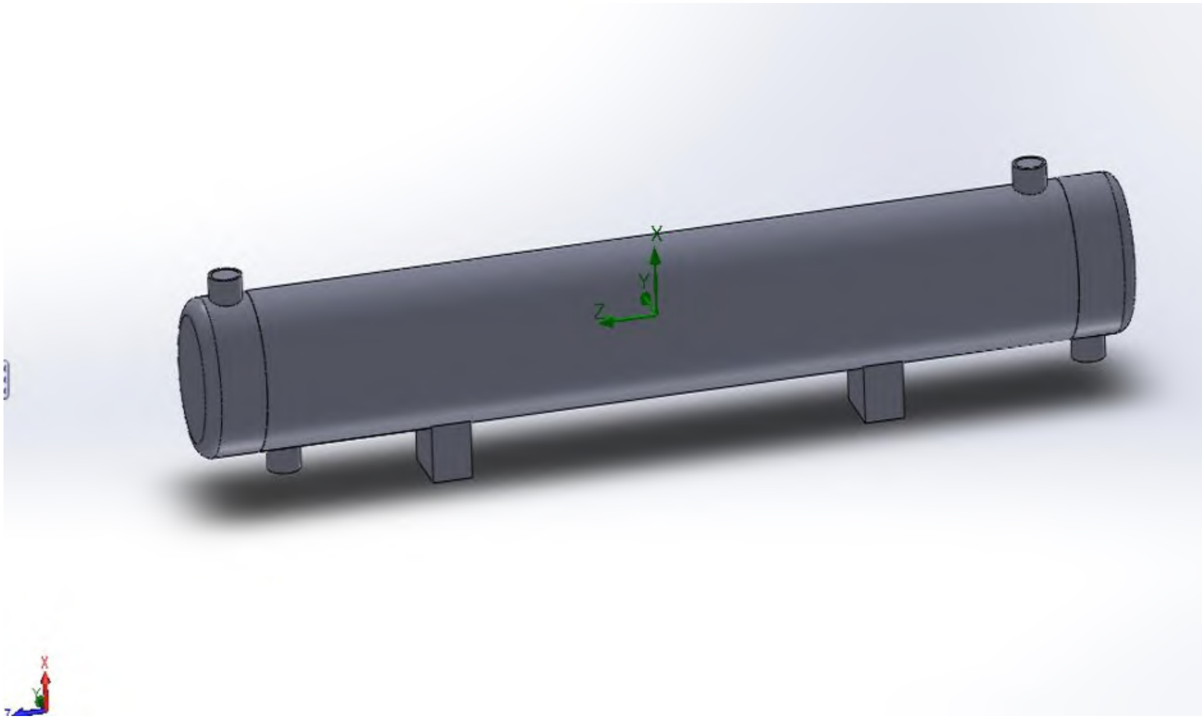


Figure 1.17 The Heat Exchanger Design with Helical Baffle

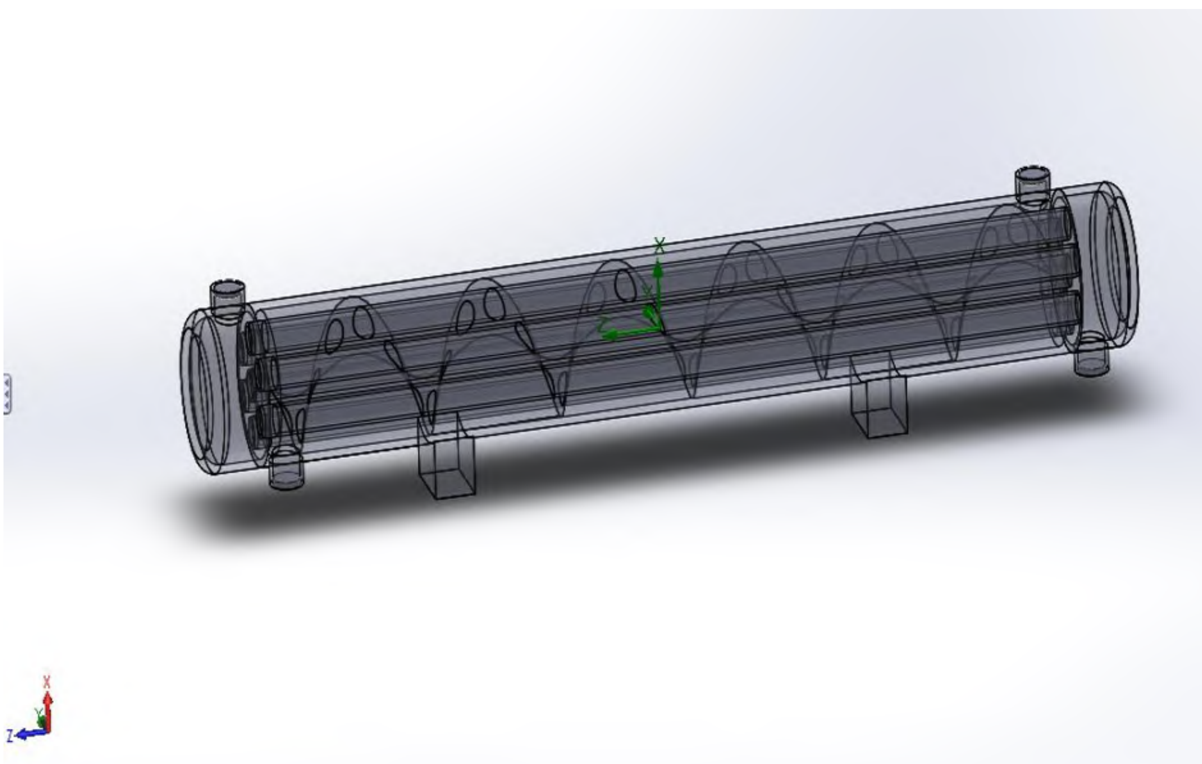


Figure 1.18 Isometric view of arrangement of baffles and tubes of shell and tube heat exchanger with baffle inclination.

Geometric dimensions of shell and tube heat exchanger

The Heat Exchanger was designed according to TEMA Standards and dimensions are in the below table.

Heat exchanger length, *L*	600mm
Shell inner diameter, *Di*	90mm
Tube outer diameter, do	20mm
Tube bundle geometry and pitch Triangular	30mm
Number of tubes, *Nt*	7
Number of baffles, *Nb*	6

Table 1.4 The Heat Exchanger Dimensions[11]

3.5 Meshing

The partial differential equations that govern fluid flow and heat transfer are not usually amenable to analytical solutions, except for very simple cases. Therefore, in order to analyze fluid flows, flow domains are split into smaller subdomains (made up of geometric primitives like hexahedra and tetrahedra in 3D and quadrilaterals and triangles in 2D). The governing equations are then discretized and solved inside each of these subdomains. Typically, one of three methods is used to solve the approximate version of the system of equations: finite volumes, finite elements, or finite differences. Care must be taken to ensure proper continuity of solution across the common interfaces between two subdomains, so that the approximate solutions inside various portions can be put together to give a complete picture of fluid flow in the entire domain. The subdomains are often called elements or cells, and the collection of all elements or cells is called a mesh or grid. The origin of the term mesh (or grid) goes back to early days of CFD when most analyses were 2D in nature. For 2D analyses, a domain split into elements resembles a wire mesh, hence the name.

The process of obtaining an appropriate mesh (or grid) is termed mesh generation (or grid generation), and has long been considered a bottleneck in the analysis process due to the lack of a fully automatic mesh generation procedure. Specialized software progams have been developed for the purpose of mesh and grid generation, and access to a good software package and expertise in using this software are vital to the success of a modeling effort.[9]

Number Of Cells

To get better results, number of cells are created as much as possible.

Total cells	581736
Fluid cells	304631
Solid cells	44447
Partial cells	232658
Irregular cells	0
Trimmed cells	0

Table 1.5 Mesh Informations

3.6 Problem Setup

Simulation was carried out in Solidworks Fluid Simulation. In the Fluid Simulation solver Pressure Based type was selected, absolute velocity formation and steady time was selected for the simulation. In the model option energy calculation was on and the viscous was set as standard k-e, standard wall function(k-epsilon 2 eqn).

In cell zone fluid water-liquid was selected. Water-liquid and steel, aluminum was selected as materials for simulation. Boundary condition was selected for inlet, outlet as seen in the below tables.

Fluid Properties

Property	Symbol	Unit	Water
Specific heat	C_p	kj/kg.K	4.176
Thermal conductivity	K	W/m.K	0.615
Viscosity	μ	kg/m.s	0.001
Prandtl number	Pr	-	5.42
Density	ρ	kg/m^3	996

Table 1.6 Fluid Properties

Shell Side Properties

Quantity	Symbol	Value
Shell side fluid		water
Mass flow rate	m_s	0.03kg/s
Shell ID	Di_s	0.09m
Shell length	L_s	0.6m
Tube pitch	p_t	0.03m
No. of baffles	N_b	6
No. of passes		1

Table 1.7 Shell Side Properties

Tube Side Properties

Quantity	Symbol	Value
Tube side fluid		water
Mass flow rate	m_t	0.01kg/s
Tube OD	Do_t	0.02m
Tube thickness	t_t	0.002m
No. of tubes		7

Table 1.8 Tube Side Properties

Chapter 4

Results

4.RESULTS

Under the above boundary condition and solution initialize condition simulation was set for 272 iterations.

4.1. Temperature Distrubition across the tube and Shell

Temperature distibutions across the tube and shell can be observed in the below tables. There are 7 tubes in the shell. Cut plots are taken right on the middle of the shell to observe more transfer area. Fluid in the middle tube became cold faster than other tubes because the cold fluid that contacts to the tube in the middle is much more than the others.

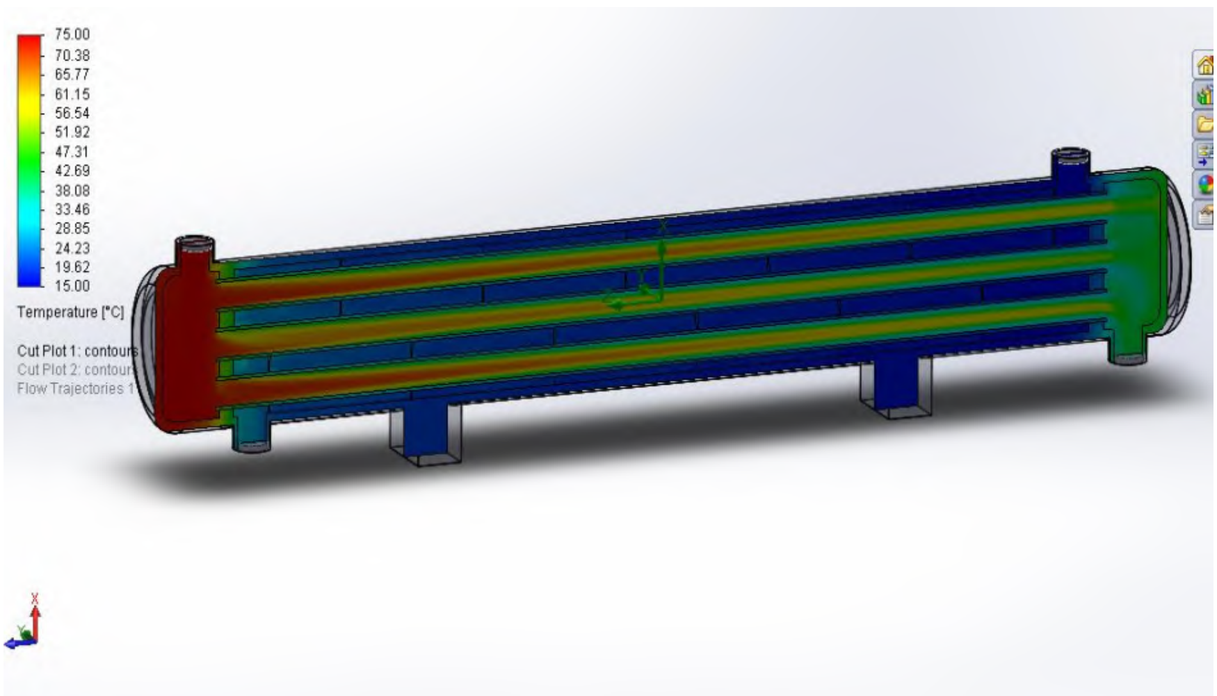


Figure 1.19 Temperature Distrubition Front Cut Plot

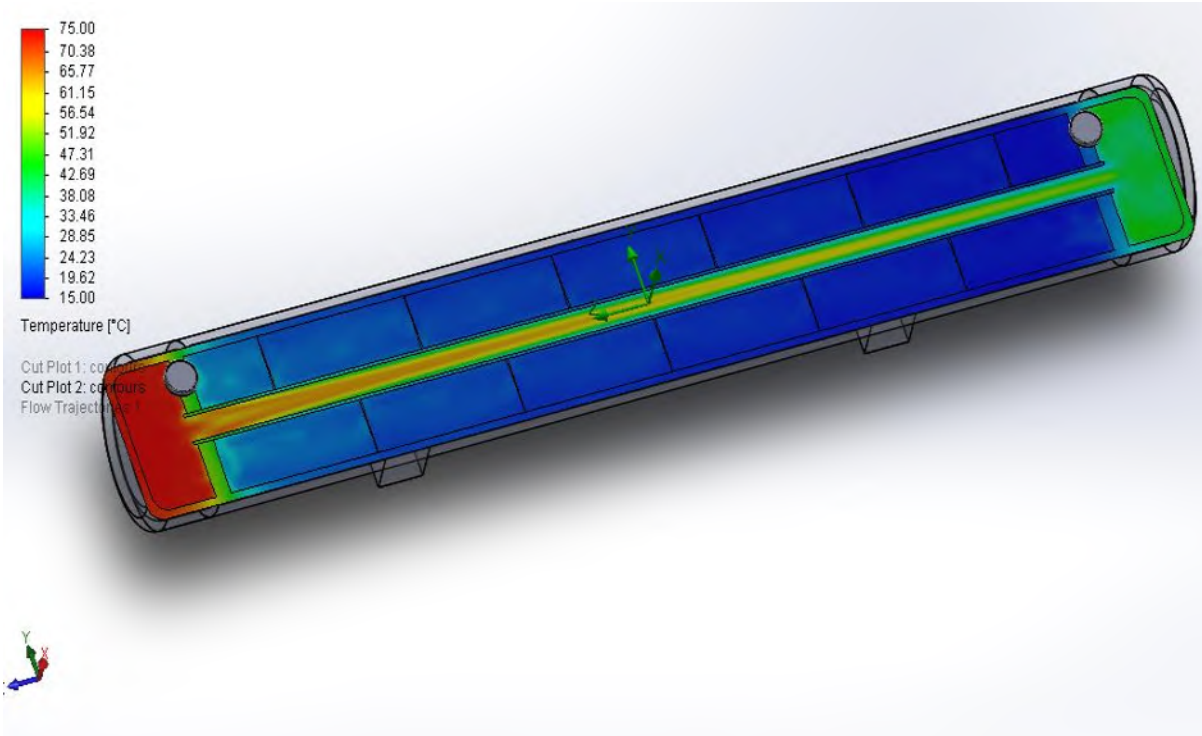


Figure 1.20 Temperature Distrubition Top Cut Plot

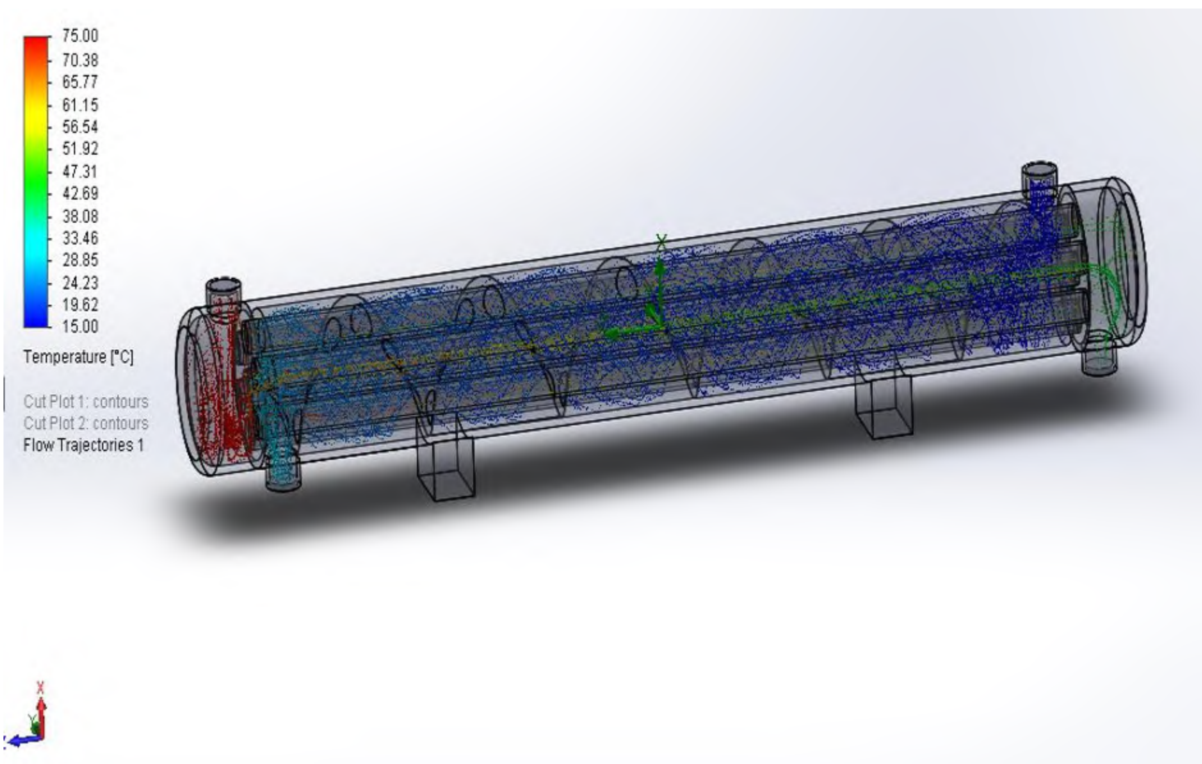


Figure 1.21 Temperature Distrubition Flow Trajectory

Medium - Fluid/Solid; Iteration = 272

	x [m]	y [m]	z [m]	Temperature (Fluid) [°C]
Inlet temperature of tube side	0,061355513	-1,37914E-08	0,327500075	75
Outlet temperature of tube side	-0,061355513	-1,37914E-08	-0,327500075	42,29
Inlet temperature of shell side	0,061355513	-1,37914E-08	-0,286999941	15
Outlet temperature of shell side	-0,061355513	-1,37914E-08	0,286999941	26,56

Table 1.9 Temperature values on the monitoring points

4.2. Velocity Distrubition across the tube and shell

Velocity Distrubition across the tube and shell can be observed in the below tables. By dint of baffles, inlet and outlet velocities are diffirent. Therefore, travel time of fluids are longer and the amount of heat transfer is more than normally.

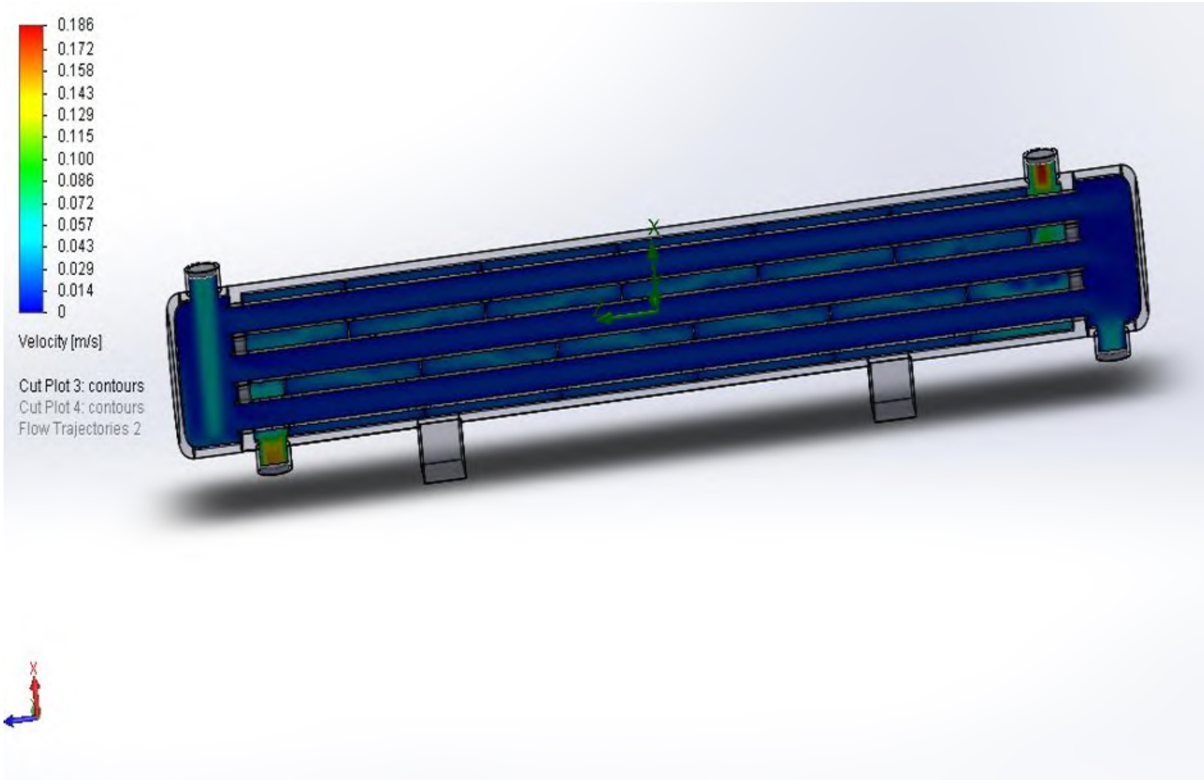


Figure 1.22 Velocity Distrubition on the Front Cut Plot

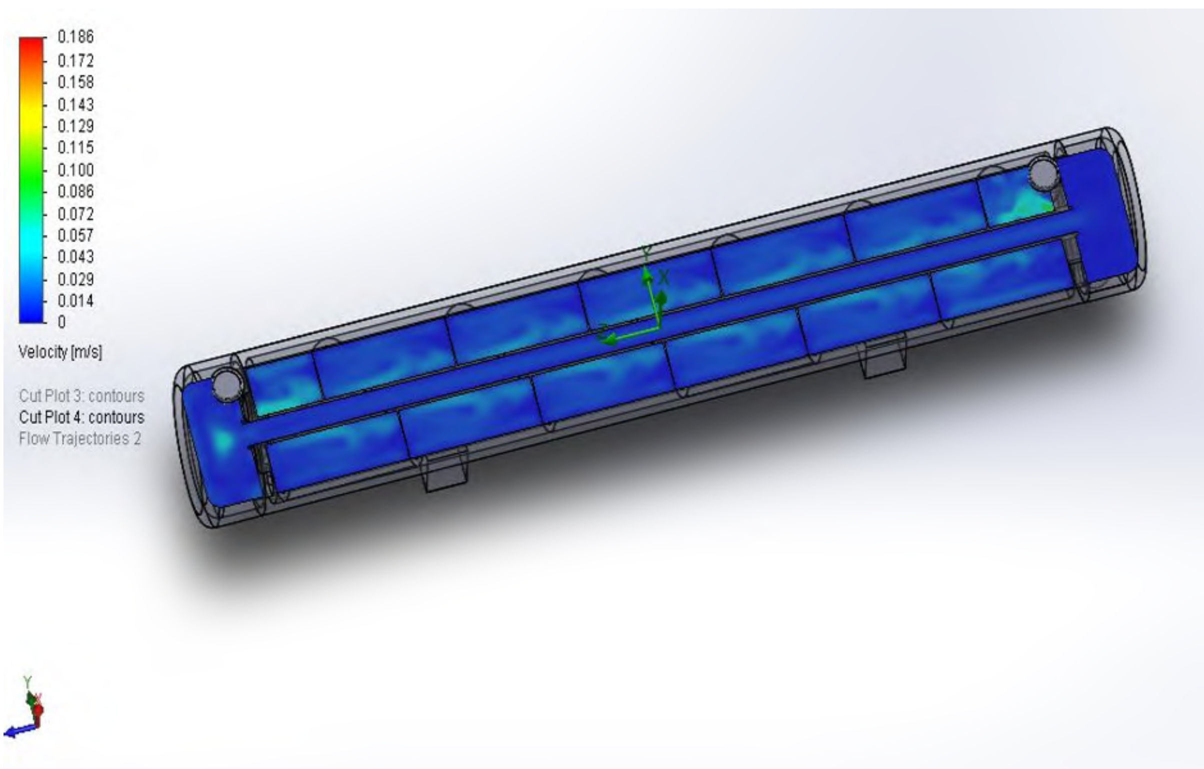


Figure 1.23 Velocity Distrubition on the Top Cut Plot

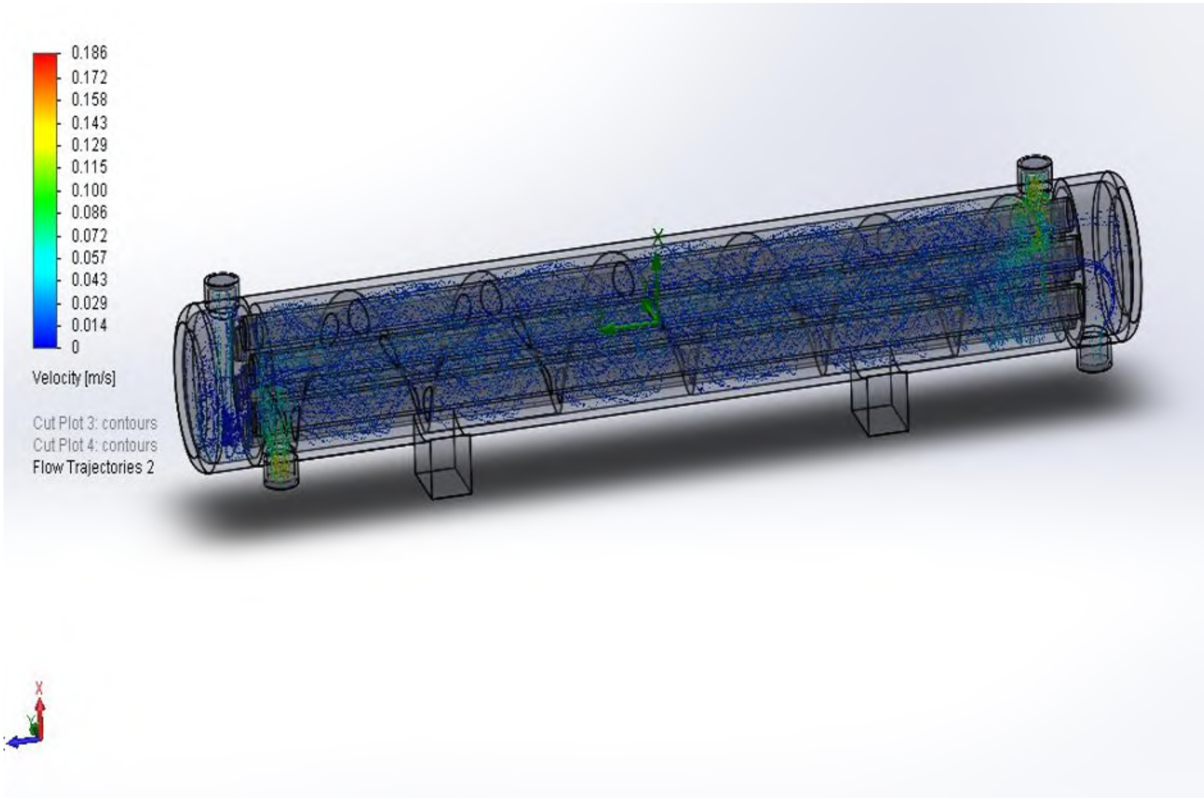


Figure 1.24 Velocity Distrubition Flow Trajectory

Medium - Fluid/Solid; Iteration = 272

	x [m]	y [m]	z [m]	Velocity [m/s]
Inlet velocity of tube side	0,061355513	-1,37914E-08	0,327500075	0,063
Outlet velocity of tube side	-0,061355513	-1,37914E-08	-0,327500075	0,049
Inlet velocity of shell side	0,061355513	-1,37914E-08	-0,286999941	0,184
Outlet velocity of shell side	-0,061355513	-1,37914E-08	0,286999941	0,154

Table 1.10 Velocity Values on the Monitoring Points

Chapter 5

Static Analysis of the Heat Exchanger

5.Static Analysis of the Heat Exchanger

After designing and heat transfer analysis, we checked how much pressure can handle our heat exchanger. By using our design dimensions we find the below result:

ID = 0.09m

OD = 0.1m

$t_s = \frac{pxDs}{fxj - 0.6p} + c$

t_s = the shell thickness

p= design pressure

D_s= the shell ID

f=max. Allowable stress

j= joint efficency

c= corrosion allowence (for stainless steel 321 is 0.7)

In our design, the shell thickness is 5 mm and for static analyze we need to know how much pressure we can use for max deformation.

5 mm $= \frac{px90mm}{234.422 \frac{N}{mm^2 (0.7)} - 0.6p} + c$

After calculations P_{max}=7.421 N/mm^2 which is 74.21 Bar

So, we calculated our max allowable pressure for static analysis. To use the max pressure we should multiply the pressure with safety factor and it is 1.5.

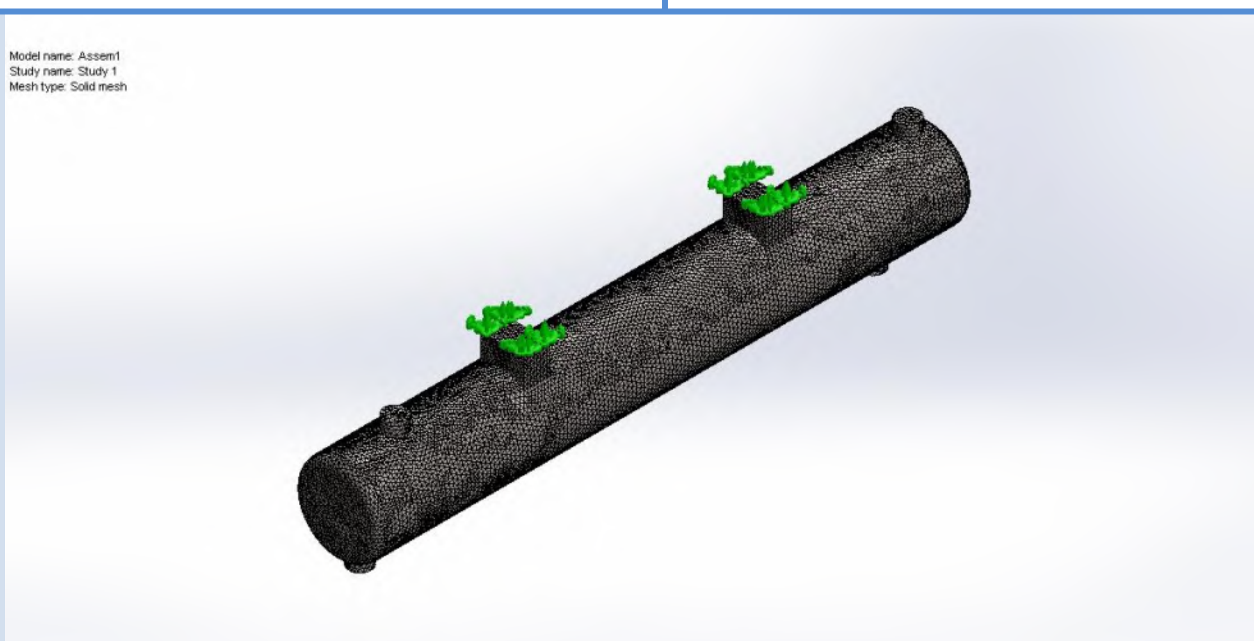
And now our pressure for analysis is 115 bar.

5.1.Mesh Information

In the below table, our mesh sizes are shown.

Mesh type	Solid Mesh
Mesher Used:	Curvature based mesh
Jacobian points	4 Points
Maximum element size	4.04656 mm
Minimum element size	1.34884 mm
Mesh Quality	High
Remesh failed parts with incompatible mesh	On

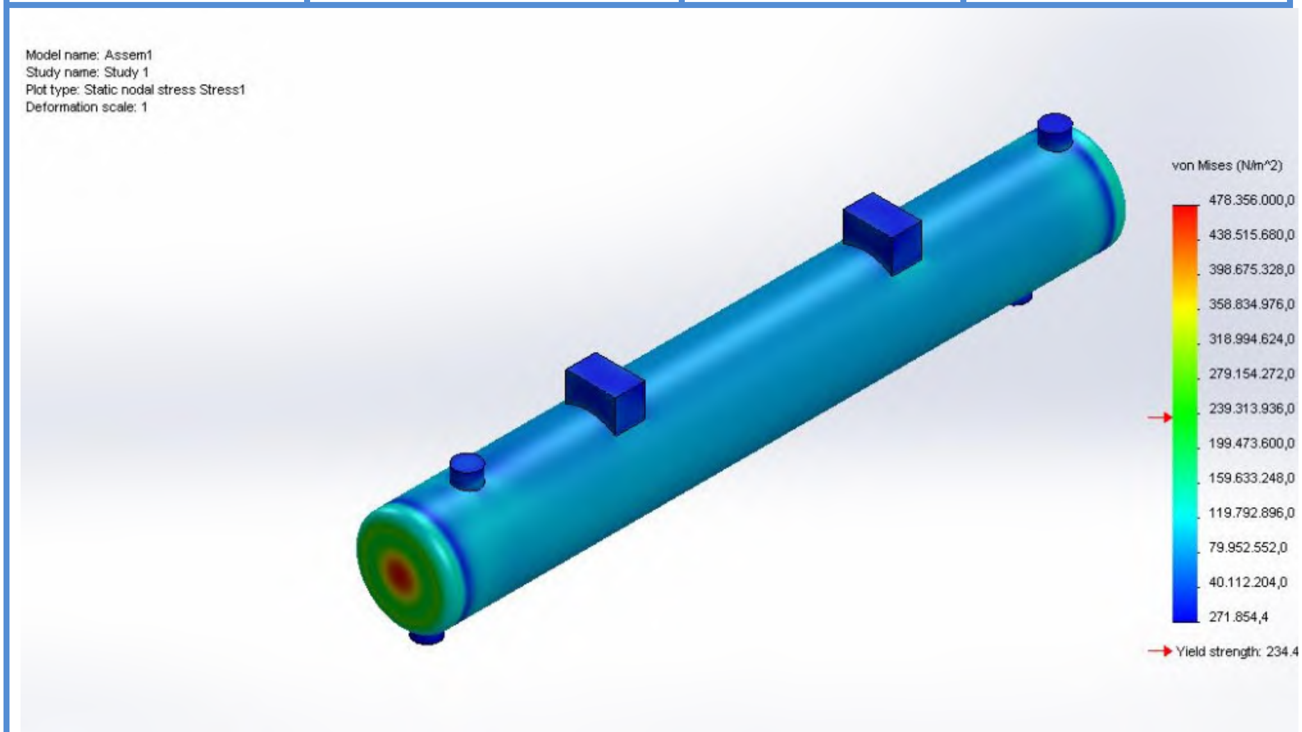
Total Nodes	277689
Total Elements	164601
Maximum Aspect Ratio	4.5999
% of elements with Aspect Ratio < 3	99.8
% of elements with Aspect Ratio > 10	0
% of distorted elements(Jacobian)	0



5.2.Study Results

Below our von Mises Stress results are shown. Under 11.5 MPa min and max stress on the shell that is made of stainless steel 321.

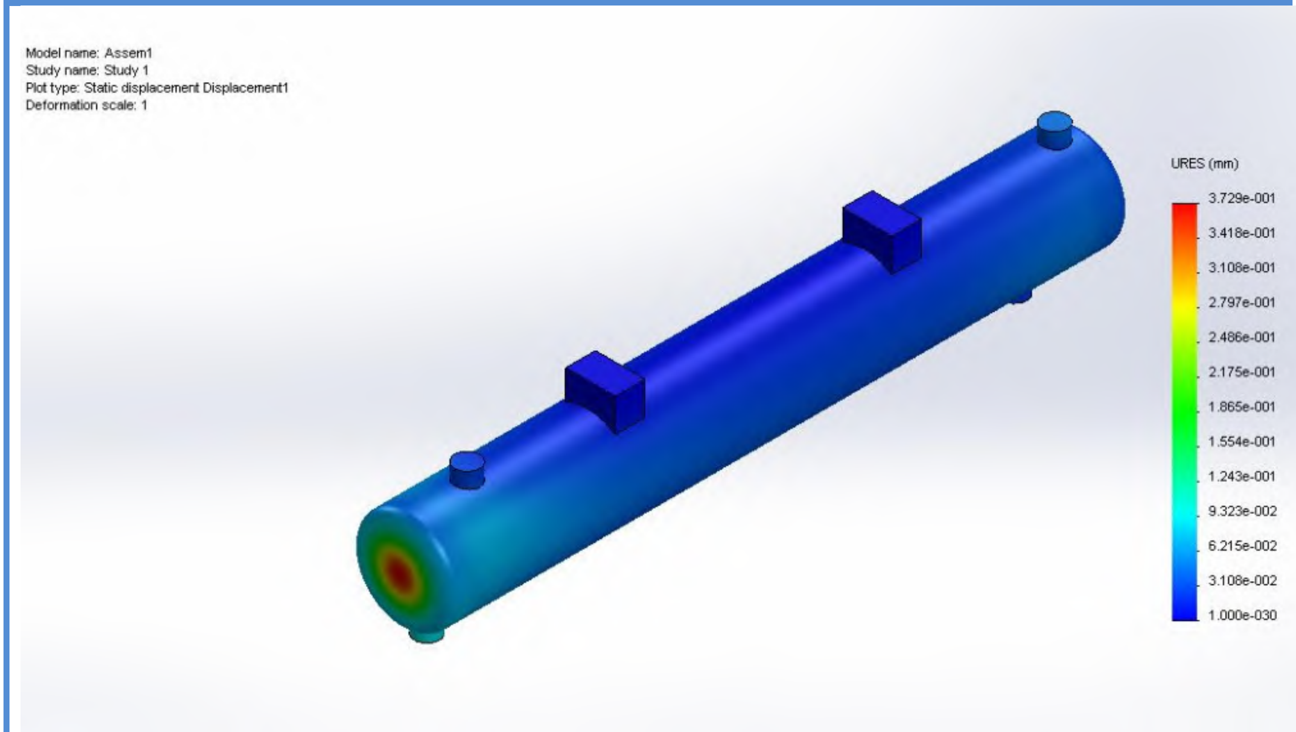
Name	Type	Min	Max
Stress1	VON: von Mises Stress	271854 N/m^2 Node: 50324	4.78356e+008 N/m^2 Node: 26124



Assem1-Study 1-Stress-Stress1

And you can see the max displacement on the shell which is under the max allowable pressure.

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 49728	0.372917 mm Node: 23575



Assem1-Study 1-Displacement-Displacement1

Chapter 6

Heat Transfer Rate

6.Heat Transfer Rate

Heat transfer describes the exchange of thermal energy, between physical systems depending on the temperature and pressure, by dissipating heat. The fundamental modes of heat transfer are *conduction* or *diffusion*, *convection* and *radiation*.

The exchange of kinetic energy of particles through the boundary between two systems which are at different temperatures from each other or from their surroundings. Heat transfer always occurs from a region of high temperature to another region of lower temperature. Heat transfer changes the internal energy of both systems involved according to the First Law of Thermodynamics.[1] The Second Law of Thermodynamics defines the concept of thermodynamic entropy, by measurable heat transfer.

Thermal equilibrium is reached when all involved bodies and the surroundings reach the same temperature. Thermal expansion is the tendency of matter to change in volume in response to a change in temperature.[10]

For heat transfer rate formula is;

$$Q = m * cp * \Delta T$$

m=mass flow rate

Cp = Speific Heat of Water

ΔT = Temperature Difference Between Tube Side

Q=(0.03)x(4.176)x(75-42.29) = 4.099 kj/s   which is 3523.65 kcal/h.

The heat transfer coefficent is calculated from below formulas.

Tube Clearance

$C = P_t - D_{ot} = 0.03-0.02 = 0.01m$

L_B= Baffle spacing = 0.086 m

Cross Flow Area

$A_s=(D_{is}xCxL_B)/P_t = [(0.09)x(0.01)x(0.086)]/(0.03) = 2.58x10^{-3}$

Equivalent Diameter

$D_E=4[(P_t^2-\pi D_{ot}^2/4)/(\pi D_{ot})] = 0.03729$ m

Max Velocity

$V_{max}=Q_s/A_s$ =0.3875 m/s

Reynold Number

$Re= (\rho xV_{vax}xD_E)/\mu = 14392.07 >4000$ so it is turbulant flow

Heat Transfer Coefficent

$h_o= (0.36xKxRe^{0.55}xPr^{0.33})/RxD_E$ R=1 for water

h_o= 2008.042 $W/m^2.K$

Chapter 7

Conclusion

7. Conclusions

The heat transfer and flow distribution is discussed in detail and proposed model is compared with increasing baffle inclination angle. The model predicts the heat transfer and pressure drop with an average error of 20%. Thus the model can be improved. The assumption worked well in this geometry and meshing expect the outlet and inlet region where rapid mixing and change in flow direction takes place. Thus improvement is expected if the helical baffle used in the model should have complete contact with the surface of the shell, it will help in more turbulence across shell side and the heat transfer rate will increase. If different flow rate is taken, it might be help to get better heat transfer and to get better temperature difference between inlet and outlet. Moreover the model has provided the reliable results by considering the standard k-e and standard wall function model, but this model over predicts the turbulence in regions with large normal strain. Thus this model can also be improved by using Nusselt number and Reynolds stress model, but with higher computational theory. Furthermore the enhance wall function are not use in this project, but they can be very useful. The heat transfer rate is poor because most of the fluid passes without the interaction with baffles. Thus the design can be modified for better heat transfer in two ways either the decreasing the shell diameter, so that it will be a proper contact with the helical baffle or by increasing the baffle so that baffles will be proper contact with the shell. It is because the heat transfer area is not utilized efficiently. Thus the design can further be improved by creating cross-flow regions in such a way that flow doesn’t remain parallel to the tubes. It will allow the outer shell fluid to have contact with the inner shell fluid, thus heat transfer rate will increase.

Chapter 8

Reference

8. References

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