International Journal of Science and Research (IJSR)
ISSN (Online): 2319-7064
Impact Factor (2012): 3.358

Shell and Tube Heat Exchanger Performance Analysis

Durgesh Bhatt[1], Priyanka M Javhar[2]

[1]Sri Satya Sai Institute of Science & Technology, Sehore, Madhya Pradesh, India
[2]Sri Satya Sai Institute of Science & Technology, Sehore, Madhya Pradesh, India

Abstract: ***A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, at different temperatures and in thermal contact. In this problem of heat transfer involved the condition where different constructional parameters are changed for getting the performance review under different condition. An excel program has been developed for the ease of calculation and obtaining result after changing different parameters. The tube diameter, tube length, shell types etc. are all standardized and are available only in certain sizes and geometry. And so the design of a shell-and-tube heat exchanger usually involves a trial and error procedure where for a certain combination of the design variables the heat transfer area is calculated and then another combination is tried to check if there is any possibility of increasing the heat transfer coefficient. Since several discrete combinations of the design configurations are possible, the designer needs an efficient strategy to quickly locate the design configuration having the minimum heat exchanger cost. In this particular problem the tube metallurgy and baffle spacing are being changed the results are obtained. In current paper the baffle spacing and tube metallurgy are the parameters considering change and effect of the same of heat transfer coefficient have been considered.***

Keywords: Shell and Tube heat exchanger, Performance analysis, Optimization, M S Excel

1. Introduction

Transfer of heat from one fluid to another is an important operation for most of the chemical industries. The most common application of heat transfer is in designing of heat transfer equipment for exchanging heat from one fluid to another fluid. Such devices for efficient transfer of heat are generally called Heat Exchanger.

1.2 Classification of Heat Exchanger

- According to construction features
- According to heat transfer mechanisms
- According to flow arrangements
- According to transfer processes
- According to surface compactness
- According to number of fluids

Generally, the most basic compact heat ex-changers have a 50% less than volume of that of a comparable shell-and-tube heat exchanger, for a given work. 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.

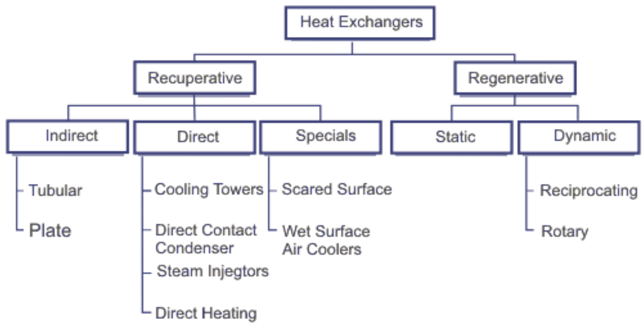

Figure 1: HE according to construction features

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.

1.3 Basic construction of S & T Exchanger

This exchanger, shown in Fig. 1.1, 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, frontend head, rear-end head, baffles, and tube-sheet.

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.

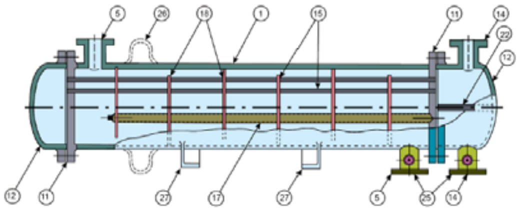

Figure 2: Basic components of STE

Shell-and-tube exchangers are classified and constructed in accordance with the widely used TEMA (Tubular Exchanger Manufacturers Association) standards (TEMA, 1999), DIN and other standards in Europe and elsewhere, 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. 1.2. 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. 1.6. Those exchangers may not be identifiable by the TEMA letter designation.

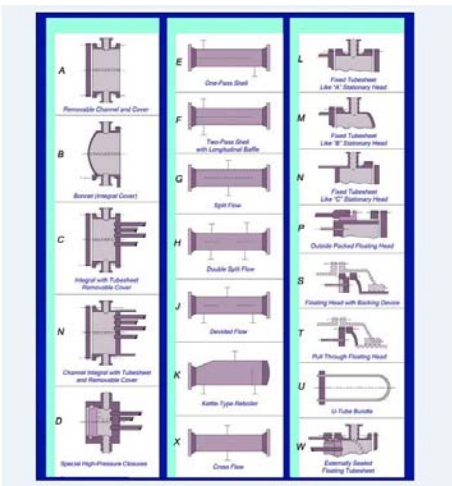

Figure 1: TEMA designation of STE

The three most common types of shell-and-tube exchangers are (1) fixed tube-sheet 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 (see Fig. 1.2), depending on the thermal stresses in the shell, tube, or tube-sheet, 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 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, tube-sheet thickness formulas, and so on, and must be consulted for all these details. Tubular exchangers are widely used in industry for the following reasons.

1.4 Application

Generally the shell and heat type heat exchanger are widely used for various purposes having limitation to be designed for maximum up to 15000 psi, 1000 ^oF & 30000 ft^2/shell. Beyond above given parameter special consideration is required for the design of heat exchanger. The design is ideal for high pressure and temperature services.

- Shell and tube heat exchanger are easy to clean for floating head type configuration so, can be used in dirty services.
- Shell and tube type heat exchanger can be used for higher temperature difference services as it can accommodate thermal expansion.
- They are most suitable for gas services and phase change service.

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 [0.1m^2 (1 ft^2)] to supergiant [over 105m2 (106 ft^2)] surface area. They are used extensively as process heat exchangers in the petroleum-refining and chemical industries; as steam generators, condensers, boiler feed water 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.

2. Literature Review

A Shell and tube heat exchanger is a device in which energy is transferred from one fluid to another across a solid surface. Exchanger analysis and design therefore involve both convection and conduction. Two important problems in heat exchanger analysis are (1) rating existing heat exchangers and (ii) sizing heat exchangers for a particular application. Rating involves determination of the rate of heat transfer, the change in temperature of the two fluids and the pressure drop across the heat exchanger. Sizing involves selection of a specific heat exchanger from those currently available or determining the dimensions for the design of a new heat exchanger, given the required rate of heat transfer and allowable pressure drop. The LMTD method can be readily used when the inlet and outlet temperatures of both the hot and cold fluids are known. When the outlet temperatures are not known, the LMTD can only be used in an iterative scheme. In this case the effectiveness-NTU method can be used to simplify the analysis. The choice of heat exchanger type directly affects the process performance and also influences plant size, plant layout, length of pipe runs and the strength and size of supporting structures. The most commonly used type of heat exchanger is the shell-and tube heat exchanger, the optimal design of which is the main objective of this study. Computer software marketed by companies such as HTRI and HTFS are used extensively in the thermal design and rating of HEs. These packages incorporate various design options for the heat exchangers including the variations in the tube diameter, tube pitch, shell type, number of tube passes, baffle spacing, baffle cut, etc. A primary objective in the Heat Exchanger Design (HED) is the estimation of the minimum heat transfer area required for a given heat duty, as it governs the overall cost of the HE. But there is no concrete objective function that can be expressed explicitly as a function of the design variables and in fact many numbers of discrete combinations of the design variables are possible as is elaborated below. Thus the optimal design of heat exchanger can be posed as a large scale, discrete, combinatorial optimization problem. Most of the traditional optimization techniques based on gradient methods have the possibility of getting trapped at local optimum depending upon the degree of non-linearity and initial guess. Hence, these traditional optimization techniques do not ensure global optimum and also have limited applications. In the recent past, some experts studied on the design, performance analysis and simulation studies on heat exchangers. Modeling and Simulation of Shell and Tube Heat Exchangers Under Milk Fouling was carried out. Dynamic Model for Shell and Tube Heat Exchangers was discussed. Shell and Tube heat exchangers are applied where high temperature and pressure demands are significant and can be employed for a process requiring large quantities of fluid to be heated or cooled. Due to their design, these exchangers offer a large heat transfer area and provide high heat transfer efficiency in comparison with others. Modeling is a representation of physical or chemical process by a set of mathematical relationships that adequately describe the significant process behavior. Improving or understanding chemical process operation is a major objective for developing a process model. These models are often used for Process design, Safety system analysis and Process control. A steady state model for the outlet temperature of both the cold and hot fluid of a shell and tube heat exchanger will be developed and simulated, which will be verified with the experiments conducted. Based on these observations correlations to find film heat transfer coefficients will be developed during any process of refining of chemical manufacturing the heat exchangers are widely used equipment for heat recovery purpose. In most of the power plants, refinery, petrochemical industries heat exchangers are the common equipment to be visualized in units.

The efficiency of the heat exchangers are depends on the how much heat transfer is done during the service. As the service life passes the heat transfer rate decreases and the efficiency as well. The dirt and the deposits in the process fluid tend to adhere to inner and outer surface of the shell and tube type heat exchanger tubes. This phenomenon is known as fouling of the heat exchanger. So keep the required process parameters in maintained condition the tube bundles of the shell and tube type of exchangers are required to clean out at a period of time.

Ebieto, C.E and Eke G.B. [1] in his experimental paper the performance analysis carried out of shell and tube heat exchanger & analytical method was used to develop correlation for the performance analysis.

The thermal analysis of a shell and tube heat exchanger involves the determination of the overall heat-transfer coefficient from the individual film coefficients, and (Kern, 1965). The shell-side coefficient presents the greatest difficulty due to the very complex nature of the flow in the shell. In addition, if the exchanger employs multiple tube passes, then the LMTD correction factor must be used in calculating the mean temperature difference in the exchanger. For the turbulent flow regime ($Re \geq 10^4$), the following correlation is widely used (serth, 2007).

$$Nu = Re^{0.8} Pr^{1/3} (\mu/\mu_w)^{0.14} \quad (1)$$

Where,

Nu = Nusselt Number = hd/k
Re = Reynold's Number = $DV\rho/\mu$
Pr = Prandtl Number = $c_p\mu/k$
D = Inside diameter of the pipe
V = average fluid velocity.

C_p,μ,ρ,k = Fluid properties at avg. bulk temperature.
μ_w = Fluid viscosity evaluated at average wall temperature.

Eq. (1) holds good for $0.5 \leq Pr \geq 17{,}000$ & for pipes $L/D \geq 10$. However for short pipes $10 \leq L/D \geq 60$, the right hand side of the equation is often multiplied by the factor $[1+(D/L)^{2/3}]$ to correct for the entrance and exit effects. (Serth 2007).

For laminar flow in circular pipes (Re < 2100), the seider-Tate correlation takes the form : $Nu = 1.86[RePrD/L]^{1/3} (\mu/\mu_w)^{0.14}$. This equation is valid for 0.5< Pr <17000 and

$$[RePrD/L]^{1/3} (\mu/\mu_w)^{0.14} \quad (2)$$

For low in he transition region ($2100 < Re < 10^4$), the Hausen correlation is:

$$Nu = 0.116(Re^{2/3} - 125) Pr^{1/3}(\mu/\mu_w)^{0.14} (1+(D/L)^{2/3}). \quad (3)$$

In computing the tube-side coefficient hi it is assumed that all tubes in the exchanger are exposed to the same thermal and hydraulic conditions. The value of hi is then the same for all tubes, and the calculation can be made for a single tube. Equations (1), (2), or (3) were used, depending on the flow regime. The tube fluid heat transfer coefficient, hi, can be calculated using;

$$hi = (NuK/D_i) \quad (4)$$

The Delaware method (Serth, 2007) was used to compute the shell-side heat transfer coefficient, ho . In the equation for the overall heat transfer coefficient, the temperature difference, Δtm , is the mean temperature difference between the two fluid streams. Since is independent of position along the exchanger, Δtm is the logarithmic mean temperature difference (Serth, 2007);

$$\Delta Tm = LMTD = \frac{(T1-t2)-(T2-t1)}{IN\left(\frac{T1-t2}{T2-t1}\right)} \quad (5)$$

Equation (5) is valid regardless of whether counter flow or parallel flow is employed. In multi-pass shell and- tube exchangers, the flow pattern is a mixture of co-current and countercurrent flow. For this reason, the mean temperature difference is derived by introducing a correction factor, **F** , which is termed the LMTD correction factor;

$$\Delta Tm = F * \Delta Tm \quad (6)$$

The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. This is corrected using two dimensionless temperature ratios (Serth, 2007); let, **N = No. of shell side passes** then

$$S = \frac{t2-t1}{T1-t1} \quad (7A)$$

$$R = \frac{T1-T2}{T2-t1} \quad (7B)$$

For R ≠1

$$F = \frac{\sqrt{R^2+1}\, In\left(\frac{1-S}{1-RS}\right)}{R-1\left[\frac{2-S\left(R+1-\sqrt{(R^2+1)}\right)}{2-S\left(R+1-\sqrt{(R^2+1)}\right)}\right]} \quad (8)$$

For R= 1

$$F = \frac{2\sqrt{2}}{(1-s)In\left[\frac{2-s(2-\sqrt{2})}{2-s(2-\sqrt{2})}\right]} \quad (9)$$

The required overall heat transfer coefficient is given by;

$$U_{req} = \frac{Q}{AF\Delta Tm} \quad (10)$$

The clean overall heat transfer coefficient is given by;

$$U_c = \left[\frac{Do}{hiDi} + \frac{Do\ln\left(\frac{Do}{Di}\right)}{2k} + \frac{1}{ho}\right]^{-1} \quad (11)$$

And the design overall heat transfer coefficient is given by;

$$U_D = \left(\frac{1}{U_c+R_D}\right)^{-1} \quad (12)$$

The effect of fouling is allowed for in the design by including the inside and outside fouling coefficients. Kern (1965) presented typical values for the fouling factors for common process service fluids used in plane tubes (not finned tubes). The fouling factor for the exchanger is given as (Serth, 2007);

$$R_D = R_{Di}\left(\frac{Do}{Di}\right) + R_{Do} \quad (13)$$

Design problems frequently include specifications of the maximum allowable pressure drops in the two streams. In that case, pressure drops for both streams would have to be calculated in order to determine the hydraulic suitability of the heat exchanger. The pressure drop due to fluid friction in the tubes is given by Equation (15) with the length of the flow path set to the tube length times the number of tube passes (Serth, 2007).

$$\Delta_{pf} = \frac{f\, n_p L G^2}{2\rho D_i s \emptyset} \quad (14)$$

Δ_{pf} = Pressure drop (Pa)
f = Darcy friction factor (dimensionless)
n_p = No. of tube passes (dimensionless)
L = Tube length (m)
G = Mass flux (kg/s m^2)
D_i = Tube inside diameter (m)
ρ = Density of water (kg/m^3)
s = Fluid specific gravity (dimensionless)
$\emptyset$ = Viscosity correction factor (dimensionless)
$\emptyset = (\mu/\mu_w)^{0.14}$ for turbulent and transition flow.
$\emptyset = (\mu/\mu_w)^{0.28}$ For laminar flow

For laminar flow friction factor is given by;
f **= 64/Re** (16)
for turbulent flow (RE > 3000) following equation can be use ;
$f = 0.4137\, R_e^{-0.2385}$ (17)
The minor losses on the tube side are estimated using the following equation:
$\Delta_{pf} = 0.5\text{x}10^{-4} a_r G^2/s$ (18)
where $\mathbf{a_r}$ is the number of velocity heads allocated for minor losses.
Serth (2007) proposed the following expression for computing the shell-side pressure drop:

$$\Delta_{pf} = \frac{f\, d_{op} G^2 (n_b+1)}{2\rho D_e s \emptyset} \quad (19)$$

The shell side friction factor formula is given by;
$f = 144\{ f_1 - 1.25\,(1 - B/De_e)(f_1 - f_2)$ (20)
An approximate equation for $f_1\, and\, f_2$ are as follows:
For $R_e \geq 1000$.
$f_1 = (0.0076 + 0.000166 d_z) R_e^{-0.125}$ $(8 \leq d_z \leq 42)$ (21)
$f_{12} = (0.0016 + 5.8\text{x}10^{-5} 6 d_z) R_e^{-0.157}$ $(8 \leq d_z \leq 23.25)$ (22)
For $R_e < 1000$.
$f_1 = \exp[0.092(InR_e)^2 - 1.48\, InR_e - 0.000526 d_z^2 + 0.047 d_z - 0.0088$
$(8 \leq d_z \leq 42)$ (23)
$f_1 = \exp[0.123(InR_e)^2 - 1.78\, InR_e - 0.00132 d_z^2 + 0.0678 d_z - 1.34$
$(8 \leq d_z \leq 42)$ (24)

B.Jayachandriah & K. Rajasekhar [2] An attempt is made in this paper is for the Design of shell and tube heat exchangers by modeling in CATIA V5 by taking the Inner Diameter of shell is 400 mm, length of the shell is 700 mm and Outer diameter of tube is 12.5mm, length of Tube is 800mm and Shell material as Steel 1008, Tube material as Copper and Brass.

By using modeling procedure Assembly Shell and Tube with water as medium is done. By using ANSYS software, the thermal analysis of Shell and Tube heat exchangers is carried out by varying the Tube materials. Comparison is made between the Experimental results, ANSYS. With the help of the available numerical results, the design of Shell and Tube Heat exchangers can be altered for better efficiency.

Following are the dimensions of the model considered for the study:

No of tubes = 44
Length of the tubes = 800mm
Tube diameter = 25mm
Tube pitch = 32mm
Clearance = Pt – do =32 – 25 =7mm
Tube layout = 90
Shell length = 800mm
Shell diameter = 135
Thickness =9mm.
Thermal properties of steel 1008
Thermal conductivity = 45 W/m oC
Density = 7872 kg/ m3
Specific heat = 481 J/ KG oC
Thermal properties of fresh water
Thermal conductivity = 0.604 W/m oC
Density = 997.4 kg/ m3
Specific heat = 162 J/ KG oC
Thermal properties of Brass
Thermal properties of fresh water
Thermal conductivity = 111 W/m oC
Density = 8600 kg/ m3
Specific heat = 162 J/ KG oC
Thermal properties of fresh water
Thermal conductivity = 400 W/m ^{o}C
Density = 8933 kg/ m^3
Specific heat = 385 J/ KG ^{o}C

2.1 Calculation

The heat release from the shell and tube heat exchangers was obtained by multiplying over all heat transfer co-efficient, Area of tubes and difference of temperatures.

Q= UAΘ_m

Where, A = area of tub, U = Overall heat transfer coefficient
Area of the tubes = A = $\pi d_o L$.
where, do = outside diameter of tubes
L = length of the tubes.

LMTD method:

$$\theta m = LMTD = \frac{(t_{h1} - t_{c1}) - (t_{h2} - t_{c2})}{IN\left(\frac{t_{h1} - t_{c1}}{t_{h2} - t_{c2}}\right)}$$

t_{h1}=hot water inlet
th2=hot water outlet
tc1=cold water inlet
tc2=cold water outlet

For copper:
Heat release: Q= UAΘ_m
Overall heat transfer coefficient:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

1/u = 1/0.604+1/1400
U = 0.603 w/m^2 k
A= 0.05494 m^2
$\Theta_{m\,=}$ 33.7 K
Heat release: Q = 11.7 W.
For brass:
Overall heat transfer coefficient U= 0.938 w/m^2 k
Area = A = 0.05494 m^2
LMTD = 27 k
Heat release Q = 8.34 w.

2.2 Discussions

Thus the ANSYS Results is calculated for copper and brass materials. The heat released from copper material is 15.63w and that from brass is 12.98 w which is less when compared with the copper. Whereas available literature Experimental Results are 14.17w for copper and11.34w for Brass. Therefore, range of variation is within 10% .Hence Results are validated. The temperature distribution profile of whole assembly in the sectional front view .From the figure it is seen that the maximum and minimum temperature for copper is from 316 K to 310 K and for brass is 318 and 308 which is simulated from the ANSYS Workbench result. Hence, it is observed that copper gives better heat transfer rate compared with that of brass.

2.3 Conclusion & Future Work

After performing all the analysis work for shell & tube heat exchangers the following observation had been done. From the study of the result as mentioned in table 1 , after performing the calculation the fluid water for bass output temperature is 310 °k which is nearer to the value mentioned in output temperature of ANSYS. As we change the tube material from the brass to the copper, temperature difference between output temperature of copper & brass had been varied.

Analysis has been done by varying the tube materials and it is found that copper material gives the better heat transfer rates than the brass material.

1. Rate of heat transfer can be improved by varying the tube diameter, length and no of tubes.
2. By changing the pitch lay out rate of heat transfer can be improved.
3. By changing the temperature of tubes and medium rate of heat transfer can be improved.
4. By changing the materials of tubes heat transfer rate can be improved.

3. Problem Statement

The thermal performance of a heat exchanger depends upon so many factors. Some of them are thermal conductivities of involved fluids and materials, velocity of flow, turbulence, quality and quantity of the insulation provided, ambient conditions flow conditions, construction etc. To make any exact prediction about the performance of heat exchanger under a set of loading conditions is always a tough job. However by certain testing and experience predictions up to a certain level can be made. The present paper is also an attempt of analyzing the performance of shell and tube type heat exchanger under certain specified variables and loading conditions.

4. Design Methods

Shell and tube heat exchangers are designed normally by using either Kern's method or Bell-Delaware method. Kern's method is mostly used for the preliminary design and provides conservative results whereas; the Bell- Delaware method is more accurate method and can provide detailed results. It can predict and estimate pressure drop and heat transfer coefficient with better accuracy. In this paper we have described Kern's method of designing in detail. The steps of designing are described as follows by considering a industrial example:

150000 lb per hour of kerosene will be heated from 75 to 120°F by cooling a gasoline stream from 160 to 120°F. Inlet pressure will be 50 psia for each stream and the maximum pressure drop of 7 psi for gasoline and 10 psi for kerosene are permissible. Published fouling factors for oil refinery streams should be used for this application.

4.1 Input Data

Hot fluid inlet temperature (T1)= 160°F
Hot fluid outlet temperature (T2) = 120°F
Cold fluid inlet temperature (t1) = 75°F
Cold fluid outlet temperature (t2) = 120°F
Fouling factor of hot fluid (Rdg) = 0.0005 (for gasoline)
Fouling factor of cold fluid (Rdk) = 0.001 (for kerosene)
Pinlet (for hot fluid) = 50 psia
Pinlet (for cold fluid) = 50 psia
Δpmax (for hot fluid) = 7 psi
Δpmax (for cold fluid) = 10 psia
Mass flow rate of cold fluid (m_k) = 150000 lb.h-1
(Subscripts k for kerosene and g for gasoline).

Fluid properties at caloric temperature
Viscosity:
76°API gasoline, µg=0.2cp (0.484 $lb.ft^{-1}.h^{-1}$)
46°API kerosene, µk =1.6 cp (3.872 $lb.ft^{-1}.h^{-1}$)
Density:
ρg=685 kg.m-3 (42.7 $lb.ft^{-3}$)
ρk=800 kg.m-3 (49.8 $lb.ft^{-3}$)
Thermal conductivity:
kg=0.075 Btu $h^{-1}ft^{-1}$ $°F^{-1}$
kk=0.083 Btu $h^{-1}ft^{-1}$ $°F^{-1}$
Specific heat capacity:
Cg = 0.57 Btu $lb^{-1}ft^{-1}$
Ck = 0.48 Btu $lb^{-1}ft^{-1}$
Specific gravity:
Sg = 0.685
Sk = 0.80

4.2 Energy balance

Assume no heat loss to surround

$$Qg = Qk = m_k C_k(t_1 - t_2) = m_g C_g(T_1 - T_2)$$

mg = 1421055 lb/ hr

4.3 Calculation of heat transfer area and tube numbers Iteration #1:

The first iteration is started assuming 1 shell pass and 2 tube pass shell and tube exchanger with following dimensions and considerations.

Fixed tube plate type
a. 1´´ OD tubes (14 BWG) on 1¼´´ square pitch (PT)
b. Tube length (Lt) = 24´ (the tube length is increased from 16´)
c. 1 shell pass-6 tube pass (tube passes is increased to 6 from 2)
d. Tube ID=0.834´´
e. Flow area per tube=0.546 inch2

The log mean temperature correction factor (FT) for 1-2 shell and tube exchanger:

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

S = (120-75)/(160-75) S = 0.529

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

R = (160-120)/(120-75) S = 0.889

For R ≠1

$$F = \frac{\sqrt{R^2+1}\,ln\left(\frac{1-S}{1-RS}\right)}{R-1\left[\frac{2-S\left(R+1-\sqrt{(R^2+1)}\right)}{2-S\left(R+1-\sqrt{(R^2+1)}\right)}\right]}$$

F = 0.802

$$\Delta Tm = LMTD = \frac{(T1-t2)-(T2-t1)}{lN\left(\frac{T1-t2}{T2-t1}\right)}$$

LMTD = 42.75 ^{0}F.

4.4 Determining the heat transfer area ('A'):

The value of overall heat transfer coefficient (Uo,assm) of 45 Btu h-1ft-2 °F-1 is assumed to initiate the design calculation for the kerosene and gasoline heat exchanger. The approximate range of overall heat transfer coefficient depending on the hot and cold fluid can be found out from text books (**[3] page 845).**

$$A = \frac{Q}{U_{assum} * LMTD * F}$$

$$A = \frac{m_g C_g (T_1 - T_2)}{U_{assum} * LMTD * F}$$

A = 2100 ft^2

Calculating no. of tubes (n_t):

$$Nt = \frac{A}{\pi d_0 L_T} = 335$$

n_t = 368 is taken corresponding to the closest standard shell ID of 31´´ for fixed tube sheet, 1-shell and 6-tube pass exchanger with 1´´ tube OD on 1¼´´ square pitch. The tube-counts are available in heat transfer text book (**[3] Table 9 & 10 page 841-843).**

4.5 Fluid velocity

$R_e = \frac{4m_k\left(\frac{n_p}{n_t}\right)}{\pi d_i \mu}$ = 11571.4 > 10^4 corresponding to n_p=6.

$$u = \frac{R_e \mu_k}{d_i \rho_k}$$

12945.15 ft/h (3.59 ft/s) = 1.04 m/s (so the design velocity is within the acceptable range).

4.6 Determination of heat transfer coefficient

4.6.1 Tube side heat transfer co-efficient (hi)

$$j_H = \frac{h_i d_i}{k}\left(\frac{\mu_k C_k}{k_k}\right)^{-\left(\frac{1}{3}\right)}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

jH=42 for the tube side fluid at Re=11571.4 (**[3] page 834**)
hi= 141.3 Btu h-1ft-1 oF-1

4.6.2 Shell side heat transfer co-efficient (ho)

Assumptions:
a.25% cut segmental baffles
b. Baffles spacing, B= 0.5DS=15.5´´ (half of the shell ID is selected).
c. Equivalent diameter for the shell side:

$$D_e = \frac{4\left(P_T{}^2 - \left(\frac{\pi}{4}\right) d_0{}^2\right)}{\pi d_0}$$

De = 0.082 ft.

d. Shell side cross flow area

$a_s = CBD_s/P_T$ = 0.675 ft2

e. Mass velocity; $G_s = \left(\frac{m_g}{a_s}\right)$ = 210526 lb. h-1.ft-2

$$R_e = \frac{D_e G_s}{\mu_g}$$

f. R_e = 35668

$$j_H = \frac{h_o D_e}{k_g}\left(\frac{\mu_g C_g}{k_g}\right)^{-\left(\frac{1}{3}\right)}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

ho=155.3 Btu h-1 ft-2 °F-1

4.7 Overall heat transfer co-efficient $U_{o,\,Cal}$;

Fouling factor, Rdk=0.001 h ft2 °F Btu-1 for kerosene and Rdg= 0.0005 h ft2 °F Btu-1 for gasoline is taken for this service.

$$U_{o,Cal} = \left[\frac{1}{h_o} + R_{dg} + \left(\frac{A_0}{A_i}\right)\left(\frac{d_0 - d_i}{2K_w}\right) + \left(\frac{A_0}{A_i}\right)\frac{1}{h_i} + \left(\frac{A_0}{A_i}\right) R_{dk}\right]^{-1}$$

$$U_{o,Cal} = 53.5 \text{ Btu h} - 1 \text{ ft} - 2\ ^\circ\text{F} - 1$$

Now, $(U_{o,Cal} - U_{o,assum})/U_{o,assum} * 100$ = 18.9 < 30%.

Therefore, the calculated overall heat transfer co-efficient is well within the design criteria.

4.8 Pressure drop calculation:

4.8.1 Tube side pressure drop calculation;

Friction factor f = 0.00028 x 144 = 0.04032 ft^2/ ft^2 (for Re=11571.4 (**[3] page 836**]).

a_t= (no. of tubes)×(flow area per tube)/(no. of passes) = 0.232 ft^2

Tube side mass velocity, $G_t = (m_k/ a_t)$ = 646552 lb/ hr ft^2.

Frictional pressure drop : $\Delta P_t = \frac{f G_t{}^2 L_t n_p}{7.5*10^{12} * d_i S_k \emptyset_t}$ = 5.81 Psi.

Return loss ΔP_{rt} : (due to change in flow direction of the tube side fluid)

$$\Delta P_{rt} = 1.334 * 10^{-13}(2n_p - 1.5)\left(\frac{G_t{}^2}{S_k}\right)$$

$$\Delta P_{rt} = 0.73\ Psi$$

Total tube side drop neglecting nozzle loss:

$\Delta P_T = \Delta P_t + \Delta P_{rt}$
$= 5.81 + 0.73$
$= 6.54 < 10\ Psi.$

Therefore the tube side pressure drop is within the maximum allowable pressure drop of 10 psi.

4.8.2 Shell side pressure drop calculation

Tube clearance, C=0.25″
Spacing, B=15.5″
a_s = 0.675 ft^2
Mass velocity G_s = 210526 lb/ h ft^2
R_e = 35668
No. of baffles = n_b = Tube length / Baffle spacing = 19

Friction factor f = 0.0017 x 144 = 0.2448 ft^2/ ft^2 with 25% cut segmental baffles (**[3] page 839**).

Shell side frictional pressure drop

$\Delta P_s = \frac{f G_{st}{}^2 D_s (n_b - 1)}{7.5*10^{12} * d_e S_k \emptyset_t}$ = 1.4 < 7.0 Psi.

$\Delta P_{rs} = 0$ (in case of single shell pass flow).

Total shell side drop neglecting nozzle loss:

$\Delta P_s = \Delta P_s + \Delta P_{rs}$
$= 1.4\ Psi$

Therefore the shell side pressure drop is within the maximum allowable pressure drop of 7 psi.

4.9 Over surface and over design:

Over design:

$$Over\ surface = (U_c - U_{o,Cal})/U_c$$

The clean overall heat transfer co-efficient: $U_c = \frac{h_o h_{io}}{h_o + h_{io}}$

$h_{io} = h_i\left(\frac{d_i}{d_o}\right)$ = 117.8 Btu h -1 ft-2 °F-1

=66.98 Btu h -1 ft-2 °F-1 U.

$$\%\ Over\ surface = \frac{66.98 - 53.5}{66.98} x\ 100 = 20\%\ (Acceptable)$$

Over design:

$$\%\ Over\ design = \frac{A - A_{req}}{A_{req}} * 100$$

The design area of heat transfer in the exchanger (n_t = 318)

$A = \pi d_o L_t n_t$ = 2312 ft^2

The required heat transfer area (where, n t =335):
$A_{req} = \pi d_o L_t n_t$ = 2105 ft^2.

=9.8% which is within the acceptable limit. % Overdesign.

5. M.S. Excel Programming for the Given Problem

Based on the above heat exchanger problem an excel program has been developed for further optimization of the design.

Input for calculation			
Description	**Input Value**	**Symbol**	**Unit**
Hot side inlet temp	160	T1	F
Hot side outlet temp	120	T2	F
Cold side inlet temp	75	t1	F
Cold side outlet temp	120	t2	F
Fouling factor of hot fluid	0.0005		
fouling factor of cold fluid'	0.001		
pressure inlet hot fluid	50	psi	
pressure inlet cold fluid	50	psi	
allowable delta P tube	7	psi	
allowable delta P shell	10	psi	
tube side mass flow rate kerosene	150000	mh	lb/hr
shell side mass flowrare gasoline	142105	mc	lb/hr
viscocity tube side - Kerosene	3.872	µ	lb/ft hr
viscocity shell side - Gasoline	0.484	µ	lb/ft hr
density tube side Kerosene	49.8	rho	lb/ft3
densty shell side gasoline	42.7	rho	lb/ft3
thermal conductivity tube side kerosene	0.083	k	btu/hr *ft*f
thermal conductivity shell side gasoline	0.075	k	btu/hr *ft*f
specific heat tube side kerosene	0.48	cpt	btu/lb ft
specific heat shell side gasoline	0.57	cts	btu/lb ft
specific gravity tube side kerosene	0.8	st	
Specific gravity shell side gasoline	0.685	ss	
Tube pitch	1.25	Pt	in
tube id	0.834	di	in
tube od	1	do	in
length	24	L	ft
no of tubes	318	nt	
no of tube side passes	6	np	
Shell ID	31	ds	in
Baffle spacing	15.5	B	
No.of tubes U tubes	335		
Therma conductivity of tube material	70	K	Btu/ hr ft
Flow area per tube	0.546		
Tube inside area Ao	0.785	Ao	ft2
Tube out side area Ai	0.54601146	Ai	ft2
Tube clearance	0.25	C	

Figure 4: Input section of MS Excel program

Output of the calculation			
Description	**Value**	**Symbol**	**Unit**
LMTD	42.45093508	ΔTm	F
Correction Factor R	0.888888889	R	
Correction Factor S	0.529411765	S	
Correction factor FT	0.802	Ft	
Assumed U0	45	Uo, aasumed	Btu / hr ft2 F
Heat transfer area A	2114.80	A	ft2
Calculate no. of tubes Nt	336.7521826	Nt	
	368		
Tube side Reynolds no Re	12717.67998	Re	
Velocity	14227.51636	u	ft/hr
	1.140272043		m/sec
Tube side heat transfer coefficient hi	141.3759087	hi	Btu / hr Ft2 F
Value of Jh	42	jh	
Shell side cross flow area	0.667361111	as	ft2
Mass velocity Gs	212935.692	Gs	lb/ h ft2
Shell side equvalent dia De	0.0825	De	ft
Shell side Reynold no	36312.20292	Re	
Value of Jh	110	jh	
Shell side heat transfer coefficient ho	154.2969844	ho	Btu / hr Ft2 F
Overall heat transfer coefficient calculated			
Calculated overallheat transfer coefficient	54.73203522	Uo, Cal	Btu / hr Ft2 F
Design check for calculated overall heat transfer coefficient	21.62674494	%	Should beless than 30%
Tube side pressure drop calculation			
1. Friction factor	0.04032	f	ft2/ft2
2.Tube flow area	0.232555556	at	ft2
3. Tube side mass velocity	645007.1667	Gt	lb/ h ft2
Frictional pressure drop	5.792633358	ΔPt	Psi
Return loss (Due to change in direction of tube side flow)	0.728423959	ΔPrt	Psi
Total tube side pressure drop (neglecting nozzle loss)	6.521057317	ΔPt	Psi
Shell side pressur drop calculation			
1. No of baffles nb	19		
2. Friction factor f	0.2448	f	
Shell side friction pressure loss	1.352438277	ΔPs	
In case of single shell return loss are o	0		
Total shell side pressure drop neglecting nozzle loss	1.352438277		
Over surface and over design			
First finding hio	117.9075079	hio	
Clean overall heat transfer coefficien Uc	64.28980799	U clean	
% over surface design	17.46284919	%	(20% acceptable limit)
Designed area of heat transfer Ad	2311.04	ft2	
Required area of heat trasnfer Ar	2103.8	ft2	
% overdesign	9.850746269	%	(10 to 20% acceptable limit)

Figure 5: Output section of MS Excel program

By the help of the program it became easy to check the effect of change in various parameters like Baffle spacing, Tube metallurgy on pressure drop, overall heat transfer coefficient. Below are the snaps representing the program input and output section.

At the left hand side input pane with unit and symbol is given for the ease and at the right hand side output pane showing the design calculation. For the identification ease the output pane divided into 04 different categories as below;

1. Calculated overall heat transfer coefficient.
2. Tube side pressure drop.
3. Shell side pressure drop.
4. Over surface and over design.

6. Results and Discussion

6.1 Change in Tube Metallurgy

The change in tube metallurgy will be effected by the change in overall heat transfer coefficient. As we select the metallurgy having higher thermal conductivity the overall heat transfer coefficient increases. The care must be taken that higher the metallurgy higher the cost of construction required, which some time will not be recovered from the operational life or the payback period may be more.

Table 1: Tube material effect on U

Material ----->	70/30 CuNi	90/10CuNi	Al. Brass	Admiralty Brass	Brass
Thermal conductivity of metal	17	26	58	64	74
Overall heat trasnfer coefficier	53.44	54.033	55.2	56.7	57.8

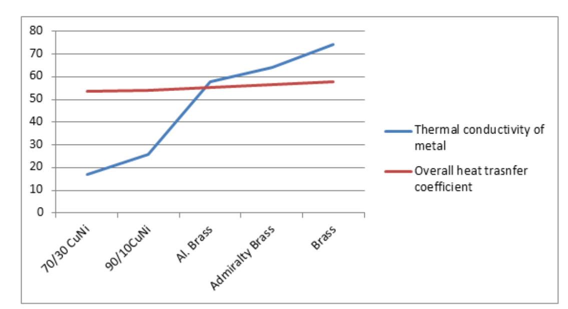

Figure 6: Tube metallurgy Vs. Overall heat transfer coefficient.

6.2 Change in Baffle spacing:

As the baffle spacing is decreased the no of baffles will be increased. Which will lead to increase in shell side Reynolds's number. That will lead to increase in overall heat transfer coefficient. But at the same time pressure drop will be increased. Following are the graph showing the individual and combine effect of baffle spacing Vs. overall heat transfer coefficient & pressure drop.

Table 2: Baffle spacing effect on U

Baffle spacing	0.1D	0.2D	0.3D	0.4D	0.5D
Overall heat tansfer coefficient	77.55	68.28	63.36	58.31	54.74

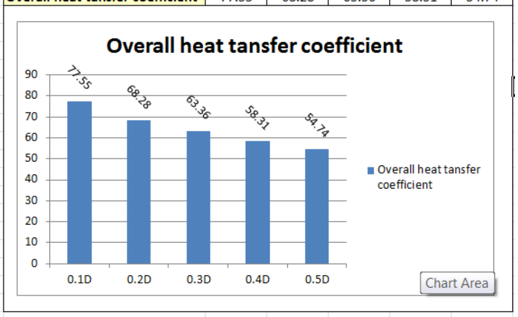

Figure 7: Baffle spacing Vs. Overall heat transfer coefficient

6.3 Effect of baffle spacing on pressure drop:

It can be observed from the following chart that as we decrease the baffle spacing the heat transfer coefficient will be increased but at the same time the pressure drop will increase remarkable. So, for getting better heat transfer from the baffle spacing modification may lead to heavy operational cost. So, the baffle spacing to be selected with available pressure drop margin.

Table 1: Baffle spacing effect on U & Pressure drop

Baffle spacing	0.1D	0.2D	0.3D	0.4D	0.5D
Overall heat tansfer coefficient	77.55	68.28	63.36	58.31	54.74
Pressure drop	158	19.88	6.01	2.53	1.35

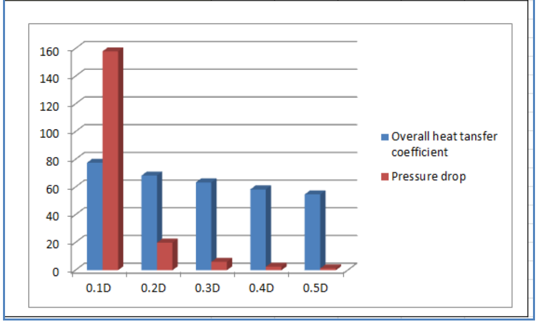

Figure 7: Baffle spacing effect on overall heat transfer coefficient and pressure drop.

7. Conclusion

On the basis of above study it is clear that a lot of factors affect the performance of the heat exchanger and the optimization obtained by the formulas depicts the cumulative effect of all the factors over the performance of the heat exchanger. It is observed that by changing the value of one variable the by keeping the rest variable as constant we can obtain the different results. Based on that result we can optimize the design of the shell and tube type heat exchanger. Higher the thermal conductivity of the tube metallurgy higher the heat transfer rate will be achieved. Less is the baffle spacing , more is the shell side passes, higher the heat transfer but at the cost of the pressure drop. So, while optimization it must be taken care that the advantage in one of the output parameter can affect the other parameters, which can lead to increase in initial or operating cost.

8. Future Scope

Currently developed MS Excel program have many variables which needs to feed manually like liquid properties. The various liquid properties can be saved in database and can be used without feeding manual data. Curve fitting algorithm can be implemented to read the values from the graph when user provides one parameter.

9. Reference

[1] A Gopichand, Prof.A.V.N.L. Sharma, G. Vijay kumar, A. Srividya, "Thermal analysis of shell and tube type heat exchanger using MATLAB and FLOEFD software. ", Volume 1 Issue :3, ,pp 279-281.

[2] A.O. Adelaja, S.J. Ojolo and M.G. Sobamowo, "Computer Aided Analysis of Thermal and Mechanical Design of Shell and Tube Heat Exchangers", Advanced Materials Vol. 367 (2012), pp731-737, Trans Tech Publications Switzerland.

[3] B Jayachandriah1, K. Rajsekhar, "Thermal Computer Aided Analysis of Thermal and Mechanical design of shell and tube exchangers," Advanced materials vol. 367(2012) pp 731-73, Trans Tech publications, switzerland.

[4] Dutta B.K. "Heat Transfer-Principles and Applications", PHI Pvt. Ltd., New Delhi, 1st ed. 2006.

[5] D. Q. Kern, "Process Heat Transfer", McGraw-Hill Book Company, Int. ed. 1965.
[6] Ebieto, C.E. and Eke G.B., "Performance Analysis of Shell and Tube Heat Exchangers using Miscible System: A case study", Journal of Emerging Trends in Engineering and Applied Sciences,2012 3 (5), pp. 899-903.
[7] Indian Standard (IS: 4503-1967): Specification for Shell and Tube Type Heat Exchangers, BIS 2007, New Delhi.
[8] Rajagapal Thundil Karuppa Raj, Srikanth Ganne, "Shell side numerical analysis of a shell and tube heat exchanger considering the effect of baffle inclination angle on a fluid flow", Thundil Karuppa Raj, R., et al: Shell side numerical analysis of shell and tube heat exchangers, Thermal science: Year 2012, Vol.16, No.4, pp 1165-1174.
[9] R. K. Sinnott, Coulson & Richardson"s Chemical Engineering: "Chemical Engineering Design" (volume 6), Butterworth-Heinemann, 3rd ed. 1999.
[10] S. Noie Baghban, M. Moghiman and E. Salehi, "Thermal Analysis of shell side flow of shell and tube type heat exchanger using experimental and theoretical methods ", Institute for Scientific Research, University of Guanajuato Lascurain de Retana No. 5, 36000 Guanajuato, Gto, México.
[11] Usman UR Rehman, "Heat transfer optimization of shell and tube heat exchanger through CFD studies ", Master thesis in innovative and sustainable Chemical engineering from Chalmers University of Technology.
[12] Yusuf Ali Kara, Ozbilen Guraras, "A computer program for designing of shell and tube heat exchanger", Applied Thermal Engineering 24(2004) 1797-1805.

Author Profile

Durgesh Bhatt received BE in 2006 from IEI, India in 2007, Worked as a design engineer in gear manufacturing company, currently perusing master degree from Sri Satya Sai Institute of Science & Technology in Thermal Branch.

Priynka Jhavar is working as a professor in Sri Satya Sai institute of science and technology, Sehore. She is providing her knowledge as a faculty for BE and ME students of Mechanical branch.