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PROCESS DESIGN AND HEAT TRANSFER ANALYSIS OF MINI SHELL AND TUBE HEAT EXCHANGER

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Abstract

A shell and tube heat exchanger improves heat exchange between hotter and colder fluids in chemical processes. Having gained wide acceptance for most applications in energy and chemical industries, shell and tube heat exchangers are now being intensively explored for prototype and small scale functions in laboratories and small industries. Because of the poor heat transfer coefficients and small scale design capabilities associated with shell and tube heat exchangers, most researches have focused on the industrial aspects of the exchangers. For these reasons, the Kern method of process design is proposed to overcome both challenges of poor heat transfer and design. While using the Kern method, Aspen Hysys V6.0 has also been used to design and rate the steady state thermal analysis of the design process. Using cooling water at 39.50^{0}C, results showed that the low pressure steam was cooled from 116^{0}C to 77^{0}C while the final outlet cold stream temperature was obtained at 50.35^{0}C. These imply that it is possible to design a simplified model of the shell and tube heat exchanger for prototype functions.

Keywords: Shell and tube heat exchanger, design, kern method, temperature.

INTRODUCTION

The extent to which shell and tube heat exchangers have been used in several industries, and had increased the efficiency of chemical processes is truly remarkable [1]. More than 35 to 40% of heat exchangers used in the world are shell and tube. Though there are several types of heat exchangers, considerable reasons for preferring the shell and tube are its relatively simple manufacturing, multipurpose applicability, robust geometry construction, easy maintenance, etc. [2]. The use of computer software has been useful in the design and optimization of heat exchangers [3][4]. A computer-based design model can be used for the preliminary design of shell and tube heat exchanger [5]. A simplified model can also be used for the study of thermal analysis of shell and tube heat exchanger of water and oil type [6]. The thermal and mechanical design of the E-type shell and tube heat exchanger was carried out with the aid of computer programming [7]. One of the most preferred reasons for its continuous usage is its reliable operation with a wide range of operating temperature.

Owing to the temperature difference in a shell and tube heat exchanger, heat transfer from fluid at higher temperature to a lower temperature occurs by the mechanism of conduction and convection. The basic principle is that two fluids flow at different temperatures separated by a wall [8]. Over the years, research have shown that prototype application of shell and tube heat exchangers in laboratories and small-scale functions can be achieved through device miniaturization which is a scaling down of the dimensions of its functional units. Heat exchanger efficiency was investigated by a researcher as a convenient approach for heat exchanger analysis that can be used to solve rating and sizing problems [9].The study also presented a new methodology for analyzing network of heat exchangers connected in series. Kern and Bell Delaware methods are investigated methods of resolving heat and design challenges. Though Kern and Bell Delaware methods are most common means of calculating heat transfer coefficients, researchers have shown that the Bell Delaware method is more complicated because of its' parameters which concern various leakages and bypass streams on the shell side [2]. To address this challenge of complexity, it has been convincingly shown that Kern's method provides the simplest model of design [10]. This is because Kern's method develops correlations based on experimental methods in commercial shell and tube heat exchangers. These correlations have to primarily do with the complexity challenge of Bell Delaware method that confronts miniaturization of the shell and tube heat exchanger.

For this reason, miniaturization and small scale application of the shell and tube heat exchanger are being actively explored. The Kern's method is considered to be one that will single handedly counter both challenges of poor heat transfer coefficient and design.

To this end, the Kern's method has gathered much attention as a simple model with the simultaneous capability of addressing both challenges of poor heat transfer and small scale design of shell and tube heat exchangers. This capability of the Kern's method has been investigated in this study. As recent research has brought to light the possibility of increasing the shell and tube heat exchanger

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performance through increase of its heat transfer capability, this article will show how an increase in heat transfer capability and design of a small scale shell and tube heat exchanger can be achieved using the Kern's method, and how its steady state thermal analysis can be justified through the use of Aspen Hysys software.

METHODOLOGY

The most common type of heat exchanger used in industrial application is the shell and tube heat exchangers. The exchangers exhibit more than 65% of the market share with a variety of design experiences of about 100 years. Shell and tube heat exchangers provide typically the heat transfer surface area density ranging from 50 to 500m^2/m^3 and are easily cleaned. The design of STHE includes thermal design and mechanical design respectively. The thermal design of STHE includes:

i. Consideration of process fluids in both shell and tube side
ii. Selection of required temperature specifications
iii. Evaluation of required effective heat transfer surface area including fouling factor
iv. Finding out logarithmic mean temperature difference (LMTD).
v. Limiting the shell and tube side pressure drop
vi. Setting shell and tube side velocity limits.

The mechanical design of Shell & Tube Heat Exchangers includes:

i. Selection of TEMA/ASME layout – based on thermal design
ii. Selection of tube parameters such as size, thickness, layout, pitch and material
iii. Limiting the upper and lower design on the tube length
iv. Selection of shell side parameters such as materials, baffle spacing, gasket and clearances.
v. Design of main shell under internal and external pressure, tube design, baffles design and gasket
vi. Thermal conductivity of tube material
vii. Setting upper and lower design limits on shell diameter, shell length, shell thickness and baffle spacing

The design codes and standards are available and carried out by referring to TEMA/ASME standards. This design is carried out both in analytically calculation and also with dedicated developed software for design and drafting. The dedicated software enables qualified design engineers to accomplish complex design calculations complying strictly with the requisite international codes and standards. The software will assist us to generate fabricated drawings to scale and 3-D images of the STHE thereby giving warning of any foul-up/mis-match in nozzles, RF-pads and in the dimensions of various components engineers to accomplish complex design calculations complying strictly with the requisite international codes and standards. The software will assist us to generate fabricated drawings to scale and 3-D images of the STHE thereby giving warning of any foul-up/mis-match in nozzles, RF-pads and in the dimensions of various components.

3.2: MATERIALS OF CONSTRUCTION AND SOURCES

The materials of construction used for this project are carbon steel for shell side and stainless steel for tube side and these materials are purchased from Iron and Steel rod market in Warn, Delta State and Laptop.

3.3: METHODOLOGY (Design Algorithm for a Shell & Tube Heat Exchanger)

3.3.1: Theoretical Calculation Background (Step – by – Step)

The thermal analysis of a shell and tube heat exchanger involves the determination of the overall heat-transfer coefficient from the individual film coefficients[11]. 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

A heat exchanger can be designed by the LMTD when the inlet and outlet conditions are specified. When the problem is to be determining the inlet and outlet temperatures for a particular heat exchanger, the analysis is performed more easily by using a method based on effectiveness of the heat exchanger and number of heat transfer units (NTU).

The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\in = \frac{actual \quad heat \quad transfer}{\max imum \quad possible \quad heat \quad transfer} = \frac{Q}{Q_{\max}} \quad (3.1)$$

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

Q = $= M_h C_{ph}\left(t_{h_1} - t_{h_2}\right) = M_c C_{pc}\left(t_{c_2} - t_{c_1}\right)$ (3.2)

The fluid capacity rate, $C = MC_p$ therefore

C = $C_h = M_h C_{ph}$ = hot fluid capacity rate, (3.3a)

$C_c = M_c C_{pc}$ = Cold fluid capacity rate, (3.3b)

C_{mm} = the minimum fluid capacity rate $\left(C_h \quad or \quad C_c\right)$,

$C_{\max}$ =the maximum fluid capacity rate $\left(C_h \quad or \quad C_c\right)$.

The number of heat transfer units;

$$NTU = \frac{UA}{C_{min}} \quad (3.4)$$

Where. U = Overall heat transfer coefficient in W/m^{2}K,
A = Heat transfer surface area in m$^{2.}$
Therefore, the effectiveness;

$$\in = \frac{C_h\left(t_{h_i} - t_{h_o}\right)}{C_{min}\left(t_{h_i} - t_{c_i}\right)} = \frac{C_c\left(t_{c_o - t_{c_i}}\right)}{C_{min}\left(t_{h_i - t_{c_i}}\right)} \quad (3.5)$$

The governing equations for design problem are usually given as follows,
Heat rate,

$$Q = C_h\left[T_{h_i} - T_{h_o}\right] = C_c\left[T_{c_o} - T_{c_i}\right] \quad (3.6)$$

Where,
Q = Heat duty of the exchanger, W
C_h = Specific heat of the hot fluid, J/KgK
C_c Specific heat of the cold fluid, J/KgK
T_{hi} = Temperature of the hot fluid inside, K
T_{ho} Temperature of the hot fluid outside, K
T_{ci} = Temperature of the cold fluid inside, K
T_{co} = Temperature of the cold fluid outside, K
Where heat capacity rate for both hot and cold fluid is $C = MC_p$

Where $\dot{m}$ = mass flowrate, Kg/sec
C = heat capacity rate.
The temperature difference AT_m, is termed logarithmic mean temperature difference between two fluid streams respectively. Since it is independent of position along the exchanger.

$$LMTD = \Delta T_m = \frac{\left(T_1 - t_2\right) - \left(T_2 - t_1\right)}{In\left[\frac{T_1 - t_2}{T_2 - t_1}\right]} \quad (3.7)$$

Or

$$\Delta_{T_m} = \frac{\left(T_{H_1} - T_{C_1}\right) - \left(T_{H_2} - T_{C_2}\right)}{\left(\frac{\left(T_{H_1} - T_{C_1}\right)}{\left(T_{H_2} - T_{C_2}\right)}\right)} \quad (3.8)$$

T_{H1} = Inlet hot fluid temperature, Tm=Outlet hot fluid temperature
T_{C1} = Inlet cold fluid temperature, T_{C2} =Outlet cold fluid temperature
Equations (3.6) & (3.7) are 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 logarithmic mean temperature difference is derived by introducing a correction factor, F, which is termed the LMTD correction factor;

$$\Delta T_m = F * \Delta T_m \quad (3.9)$$

Where, F = Correction factor.
The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. Also correction factors used in rate equation have been worked out by analysis subject to a set of simplifying assumptions, for a variety of situations. In the olden days, the formulae for them were considered too cumbersome to use.
Figure C14.a-d in Appendix C of textbook by Mills display such graphs, therefore graphs were prepared by plotting F(P,R) and these are parameters on which F depends.
Nowadays, one can compute these dimensionless factors quickly with a pocket calculator and this is corrected using two dimensionless temperature ratios (Serth, 2007);
Let, N No. of shell side passes then;

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (3.10a)$$

$$R = \frac{T_1 - T_2}{T_2 - t_1} \quad (3.10b)$$

Given next are the two common factors,

$$F_{1,2} = \frac{\left[\frac{R^2+1}{R-1}\right] In\left[\frac{1-p}{1-PR}\right]}{In\left[\frac{\Delta + \sqrt{R^2+1}}{\Delta - \sqrt{R^2+1}}\right]} \quad (3.11)$$

$$F_{2-4} = \frac{\left[\frac{\sqrt{R^2+1}}{2(R-1)}\right] In\left[\frac{1-P}{1-PR}\right]}{In\left[\frac{A+B+\sqrt{R^2+1}}{A+B-\sqrt{R^2+1}}\right]} \quad (3.12)$$

Where $A = \frac{2}{p} - 1$ *and B =* $B = \frac{2}{p}\sqrt{(1-P)(1-PR)}$ (3.13)

The formula given above for $F_{1,2}$ also applies for one shell pass and 2, 4, (or any multiple of 2) tube passes. Likewise, the formula for F_{24} also applies for two shell passes and 4, 8, (or any multiple of 4).

For Shell Side Calculation

Some of the following dimensions will be used in shell and tube heat exchangers which are; Lt = Tube length, Nt = Number of tubes, Np Number of pass, Ds = Shell inside diameter, Nb Number of baffle and B Baffle spacing.

The baffle spacing is obtained by

$$R = \frac{Lt}{N_b + 1} \quad (3.14)$$

Shell side Tube layout

For a cross section of both square and triangular pitch layouts, the tube pitch Pt, and the clearance Ct, between adjacent tubes are both defined and the equivalent diameter is obtained.

The equivalent diameter is obtained by;

$$R = \frac{4A_c}{P_{heated}} \quad (3.15)$$

The equivalent diameter for the square pitch layout is obtained by;

$$D_e = \frac{4\left[P_t^2 - \frac{\pi d_0^2}{4}\right]}{\pi d_0} \quad (3.16)$$

The equivalent diameter for the triangular pitch layout is obtained by;

$$D_e = \frac{4\left[\frac{\sqrt{3}P_0^2}{4} \frac{\pi d_0^2}{8}\right]}{\frac{\pi d_0}{2}} \quad (3.17)$$

The cross-flow area of the shell A5 is defined as;

$$A_S = \frac{D_S C_t B}{P_c} \quad (3.18)$$

Where, A_S = Shell side cross flow area, m2

B= Baffle spacing, m

C=Tube clearance P_t- D_o

D_o = Tube outer diameter, m

P_t = Tube pitch, m

Shell side mass velocity is obtained by;

$$G_S = \frac{m_S}{A_S} \quad (3.19)$$

Where $\dot{m}_S$ =mass flowrate of shell side, Kg/s, As= cross flow area of the shell, rn²

$$\mathrm{Re}_S = \frac{D_e G_s}{\mu} \quad (3.20)$$

Where D_e = equivalent diameter, m μ = dynamic or absolute viscosity, Ns/m2

Re_S = Reynolds number for the shell side.

Shell side Nusselt number is given by (Kern, 1965).

$$N_u = 0.36\left[\frac{D_e G_s}{\mu_B}\right]^{0.55}\left[\frac{C_p \mu_g}{K}\right]^{0.33}\left[\frac{\mu_g}{\mu_w}\right]^{0.14} \qquad (3.21)$$

Where;
K = Thermal conductivity, W/mK
μ_w = Dynamic viscosity of water fluid, Ns/m^2
μ_B = Shell fluid dynamic viscosity at average temperature, Ns/m^2
G_s = Mass velocity of shell side, Kg/m^{2}s
De = Equivalent diameter of shell side, m.
The Bell-Delaware method was used to compute the shell side heat transfer coefficient, "h_o" which is calculated and obtained as;

$$h_0 = \frac{N_u K}{D_e} \qquad (3.22)$$

Where;
h_o = heat transfer coefficient, W/m2K
K = Thermal conductivity, W/mK
The shell side pressure drop can be calculated and obtained by using

$$\Delta P_{shell} = \frac{f G_s^2 D_s (N_B + 1)}{2 p D_e \left(\frac{u}{u_s}\right)^{0.14}} \qquad (3.23)$$

$$N_B = \frac{L_s}{B} \qquad (3.24)$$

Where;
Δ P_{shell} Pressure drop for shell side, Pa
N_B = Number of baffles
f =Fanning factor for flow on the shell side
L_S = Shell side length, m
B = Baffle spacing,
Res = Reynolds number for the shell side
Ds Shell side diameter, rn
De Equivalent diameter, m
p Density of the shell side fluid, Kg/rn3 and
i= Viscosity of the shell side fluid.
The **Tube Side Calculation**
The diameter ratio d_t, is defined by;

$$d_r = \frac{d_0}{d_i} \qquad (3.25)$$

Where
d_r = diameter ratio,
d_o = outer diameter, m and
d_i inner diameter, m.
The tube pitch ratio Pr is defined by;

$$P_r = \frac{P_t}{d_r} \qquad (3.26)$$

The tube clearance Ct, is obtained by;

$$C_t = P_t - d_o \qquad (3.27)$$

The number of tube Nt, can be predicted in fair approximation with the shell inside diameter Ds,

$$N_t = \frac{\left(CTP \ \frac{\pi D_S^2}{4}\right)}{Shade \ \ Area} \qquad (3.28)$$

And

Shade Area = $CL.P_t^2$ (3.29)

Where, CTP is the tube count constant that account for the incomplete coverage of the shell diameter by the tubes, due to necessary clearance between the shell and the outer tube circle and tube omissions due to tube pass lanes for multiple pass design.

CTP = 0.93 for one-pass exchanger,
CTP = 0.90 for two-passes exchanger and
CTP *0,85* for three-passes exchanger
Also CL is the tube layout constant
CL 1.00 for square-pitch layout and
CL Sin (60°) =0.8660 for triangular-pitch layout
Therefore, the number of tubes Nt can be obtained by;

$$N_t = \frac{\pi}{4}\left[\frac{CTP}{CL}\right]\frac{D_S^2}{P_t^2} \qquad (3.30)$$

$$N_t = \frac{\pi}{4}\left[\frac{CTP}{CL}\right]\frac{D_S^2}{P_r^2 d_0^2} \qquad (3.31)$$

The tube side heat transfer coefficient "h_1", can be obtained by the following calculations below;

$$A_t = \frac{\pi d_t^2}{4} \qquad (3.32)$$

Where, A_p= Heat transfer area based on the tube surface, m^2

$$A_{tp} = \frac{N_t A_t}{number \quad of \quad tube \quad passes} \qquad (3.33)$$

Where, Nt = Number of tubes.

$$G_t = \frac{\dot{m}_t}{A_{tp}} \qquad (3.34)$$

Where, Gt = Mass velocity of tube, Kg/m^2s

$$\mu_t = \frac{G_t}{p} \qquad (3.35)$$

Where, p = Density of fluid at average temperature, Kg/rn^3

$$\mathrm{Re}_t = \frac{u_t p d_1}{\mu} \qquad (3.36)$$

Where Ret = Reynolds number for the tube side flow.
Using the Petukhov and Kirillov correlation to determined and obtained the Nusselt number for the tube side;

$$N_u = \frac{\left(\frac{f}{2}\right)\mathrm{Re}_t P_r}{1.07 + 12.7\left(\frac{f}{2}\right)^{1/2}\left(P_r^{2/3} - 1\right)} \qquad (3.37)$$

$$f = (1.58\, In\, \mathrm{Re}_t - 3.28)^{-2}$$

Where;
f= Friction factor of flow,
Ret Reynolds number of the tube side flow,
Pr = Prandtl number,
Therefore, the tube side heat transfer coefficient is then found as;

$$h_i = \frac{N_u K}{d_t} \qquad (3.38)$$

The tube side pressure drop can be calculated by first estimating the Darcy friction factor for flow through the tubes from the value of Reynolds number and the relative roughness, and applying the viscosity correction. Then, this friction factor is used to evaluate the pressure drop for flow through the tubes.

$$\Delta P_{value} = f_{corrected}\,\frac{L}{Di}\left[\frac{1}{2}pV_B\right] x \; Number \quad of \quad tube \quad passes \qquad (3.39)$$

Or

$$\Delta P_{tube} = \left[4f\,\frac{LN_p}{d_i} + 4N_B\right] x \left[\frac{p_t V_m^2}{2}\right] \qquad (3.40)$$

Where $V_m = \frac{u_t}{10}$
N = Number of passes,
f= friction factor of tube side

L = Length of the tube, m
Di = Inner diameter of the tube, m
V = Average flow velocity through a single tube, m/s
p = Density of the tube side fluid, Kg/rn^3.
3.3.2: Software Model Calculation (Step—by-Step with procedure using Aspen IIYSYS Heat Exchanger V6.O)
An attempt is made in this research thesis for the design and modeling of a shell and tube heat exchangers by modeling using Aspen HYSYS Heat Exchanger V6.0 by taking the Inner Diameter of shell to be 152.4mm, Length of the Shell to be *590mm* and Outer Diameter of tube to 12.5mm, Length of Tube to be 4 10mm and Shell material as Carbon Steel with Tube material as Stainless Steel.
By using Aspen HYSYS Heat Exchanger V6.0 software, the simulation of thermal analysis of a Shell and Tube Heat Exchangers is carried out. Comparison is made between the Numerical results, and Aspen HYSYS Software results. 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 to be considered for this study;
No of tubes = 16
Length of the tubes = 4 10mm
Tube outer and inner diameters = 12.5mm & 11.5mm
Tube pitch = 20.5mm
Clearance = P - d_0 32-25=7mm
Tube layout = 90
Shell length 590mm
Shell outer diameter = 168.30mm
Shell inner diameter = 152.4mm
Shell thickness = 15.9mm
Thermal properties of Carbon Steel Include;
Thermal conductivity = 54W/m°C
Density = 7850kg/rn^3
Specific heat 481J/Kg°C
Thermal properties of cooling water include;
Thermal conductivity = 0.604W/m°C
Density = 997,5Kg/rn^3
Specific heat = 162J/Kg°C
Thermal properties of Stainless Steel include;
Thermal conductivity = 45W/m°C
Density = 8030Kg/m^3
Specific heat = 462J/KG°C

3.4: GETTING STARTED USING ASPEN HYSYS DESIGN SOFTWARE

3.4.1: Business Background

Aspen Shell & Tube Exchanger enables optimum design, rating or simulation of shell and tube, double pipe, or multi-tube hairpin exchangers for both the expert and casual user. Integration with process simulators and other AspenTech engineering tools allows for improved overall process optimization through better collaboration across engineering disciplines. Aspen Shell & Tube Exchanger addresses a wide range of application needs, serving both the engineering contractor and the equipment fabricator, with the ability to share models from conceptual design to operational troubleshooting. It facilitates the full range of practical process applications, including reflux condensers, kettle reboilers, thermosyphon reboilers, falling film evaporators, and multi-shell, multi-pass feed-effluent trains. This flexibility allows the process streams to be single phase, boiling or condensing vapors, single component or any mixture with or without non-condensable gases in any condition (including superheated vapor, saturated vapor, or sub-cooled liquid).

3.4.2 Introduction to Aspen Shell and Tube Exchanger

Aspen Exchanger Design & Rating (Aspen EDR) software provides the most comprehensive package available for designing and rating heat exchangers. EDR software is used daily for thermal, mechanical and process engineering by engineering contractors, process operators and exchanger fabricators throughout the world. Aspen Shell & Tube Exchanger can be used to design all major industrial shell & tube exchanger equipment types in any combination of processes including single phase heating or cooling and boiling or condensation. Typically, users save between 10-30% on equipment cost by effectively designing their exchangers using Aspen Exchanger Design & Rating (EDR).
Given a process requirement and physical property data, the program conducts a comprehensive design search to find the optimum cost arrangement capable of satisfying the process constraints.
The program provides detailed exchanger geometry and performance details as well as a specification sheet, setting plan, and tube layout drawings.
Completed designs can be transferred to Aspen Shell & Tube Mechanical for complete mechanical design to the requirements of ASME, TEMA, or other leading international design codes and standards. Explanations for each step will be provided.

3.4.3: Design Problem Statement

We have been challenged with designing a shell side condenser containing a mixture of steam and methanol and using water as a coolant. Aspen Shell & Tube Exchanger provides the unique capability of automating the design process for optimizing the sizing of the heat exchanger to lowest cost. The design calculation will determine the shell length and diameter, the nozzle sizes, the number of tubes and passes, the number of baffles and baffle cut. Other details such as shell and header type baffle type, tube type and layout will use program defaults. Thedesign logic will optimize the heat transfer against the allowable pressure drop on both the shell and tube sides. The program has built in heuristic rules, which will stop searching once it realizes it has achieved the optimal calculation.

The process data details used for this example are shown in Table 1

Table 1: Process **Data (Source:** Aspentech EDR 2006 Licensed to: LEGENDS).

Fluids	Hot Side - Steam 40% & Methanol 60%	Cooling Water	Units
Total Flowrate	12.6	76	Kg/s
Temperature (In/Out)	*95/55*	*35/-*	°C
Inlet Pressure	1	6.5	Bar(abs)
Allowable Pressure Drop	0.05	0.7	bar
Fouling Resistance	0.00018	0.00018	m2K/W

The process data details used for this example are shown in Table 1

3.4.5 ADDITIONAL DESIGN INFORMATION

a. Geometry
The Geometry tab contains all user-specified information pertaining to exchanger configuration, shell and tube dimensions, and the overall results of the simulation after the case is run.

b. Process
The Process tab contains data regarding the physical conditions of the process streams, pressure drop, and fouling resistance of the fluids.

c. Errors & Warnings
Check the Errors & Warnings tab after running the optimization to identify operational and performance risks that were identified with the design

4.1: RESULT ANALYSIS

4.1.1: STEPWISE PROCEDURE FOR CALCULATION

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

a. The outlet temperatures of shell and tube heat exchanger are computed by equations (3.7) and (3.8) respectively.
b. The logarithmic mean temperature difference to the shell and tube are computed using equations (3.6) or (3,7).
c. Reynolds number on shell side flow is calculated using equation (3.20).
d. Nusselt number on shell side using the equation (3.21) by using Macadam’s correlation is computed.
e. Heat transfer coefficient on shell side using the equation (3.22).
f. Pressure drop ‘on shell side using the following equation (3.23) is calculated.
g. Reynolds number on tube side flow using equation (3.36) is calculated.
h. Nusselt number on tube side using the equation (3.37) by using Petukhov and Kirillov correlation is computed.
i. Heat transfer coefficient on tube side using the equation (3.38) is calculated.
j. Tube side pressure drop by using equations (3.40) and (3.30or 3.31) is calculated.

4.2: NUMERICAL SOLUTION PROCEDURE AND RESULTS

1) Area of heat exchanger is taken as 15m^2.
2) Assumed overall heat transfer coefficient is taken as 3,000 W/m^{2}k.
3) Number of tubes is taken as 16
4) Number of passes is 2
5) Shell side outer diameter and inner diameter are taken as $D_{s_{outer}} = 6.63"$ (168.3mm) and $D_{s_{outer}} = 6"$ respectively.

4.2.1: Assumptions

6) Tube side outer diameter and inner diameter are taken as d_0=12.5mrn and d_i=1 1.5mm respectively.
7) Mass flowrates of both cold fluid and hot fluid are $rh = 76kg/s$ and m_h = 12.6kg/s respectively.
8) Bundle arrangement of the tubes is Square Pitch.
9) Hot fluid and Cold fluid are Low Pressure Steam which is a mixture of 40% Steam + 60% Methanol and Cooling Water respectively.

Table 2: Inputs Process Data for Low Pressure Steam — Cooling Water

INPUTS PARAMETER	SHELL SIDE	TUBE SIDE	SYMBOLS	UNITS
Hot Fluid		**60% Methanol &** 40% Steam		
Cold Fluid	Cooling Water			
Inlet Temperature	35	95	T_1 & t_1	ºC
Mass Flowrates	12.6	76	$\dot{m}_h$ & $\dot{m}_c$	kg/s
Shell Side Outer Diameter	0.1683		Ds(outer)	M
Shell Side Inner Diameter	0.1524		Ds(inner)	M
Tube Side Inner Diameter		0.0115	di	M
Tube Side Outer Diameter		0.0125	do	M
Materials of Construction	Carbon Steel	Stainless Steel		
Thermal conductivity for carbon steel	54		k	W/m.k
Thermal conductivity for stainless steel		45	k	W/m.k
Specific heat of the shell side fluid	4200		$(C_p)s$	J/kg.k
Absolute viscosity of the shell side fluid	0.0006537		$(\mu_b)s$	Kg/m.s
Thermal conductivity of the shell side fluid	0.6280		Ks	W/m.k
Density of the shell side fluid	997..5		*Ps*	Kg/m^3
Prandtl number of the shell side fluid	4.340		$(P_r)S$	
Specific heat capacity of the tube side fluid		10,133.33	$(C_p)t$	J/kg.k
Absolute viscosity of the tube side fluid		0.0000572	$(\mu_b)t$	Kg/m.s
Thermal conductivity of the tube side fluid		0.045	Kt	W/m.k
Density of the tube side		792.645	Ps	Kg/m^3
Prandtl number of the tube side fluid		1.08	$(P_r)t$	

Table 3: Other Process Inputs Parameters

PARAMETER	VALUES	SYMBOLS	UNITS
Pitch size	0.0205	P_t	*m*
Clearance	0.008	C	*m*
Baffle spacing from the tube side	0.137	B	*m*
Tube length	0.410	L_t	*m*
Shell thickness	0.0159	T_s	*m*
Total shell length	0.590	L_s	*m*
Number of baffles	2		*m*
Diameter of baffles	0.152	D_B	*m*
Diameter of drilled hole	0.0125	$D_{d_{hole}}$	*m*
Dimensions for the floating head tube			
External diameter of the tube sheet	0.139		*m*
Internal diameter of the tube sheet	0.216		*m*
Length of floating head	0.125		*m*
Number of tube passes	2		
Number of shell pass	1		
Dimensions for the channel head tube sheet			
External diameter	0.165		*m*
Internal diameter	0.153		*m*
Length of channel head	0.125		*m*

Table 4: Theoretical Output Results Obtained

PARAMETER	VALUES	SYMBOLS	UNITS
Outlet temperature	51	55	ºC
Reynolds number, Re	391.634	5,376,184	
Nusselt number Nu	698	5,910	
Heat transfer coefficient, h	14,465	**23,127**	
Friction factor, f	0.1511	0.002224	
Pressure drop. ΔP	99,893(14.49)	22,889.52(3.32)	Pa(Psi)
Mass velocity, Gs	8,446.42	15,162.45	Kg/m^{2}s
Area, A	0.008998	0.000813	M^2

Table 5: Other Theoretical Calculated Output Results

PARAMETER		
Shell length Ls	0.590	M
Shell equivalent diameter, De	0.03031	M
Tube diameter ratio, dr	1.087	-
Tube pitch ratio, Pr	0.019	**M**
Tube clearance, Ct	0.008	M
Number of Tubes, Nt	16	-
Tube side fluid flow velocity, u_t	34	m/s
Hot fluid capacity rate	127,680	$Jk^{-}S^{-}$
Cold fluid capacity rate	319,200	$JK^{-}S^{-}$
Fluid capacity ratio	0.40	-
Number of heat transfer units NTU	0.352	-
Effectiveness, $\in$	0.667	-
Heat transfer rate, Q	5,107,200	J/S
Cold fluid outlet stream temperature, t_{∞}	51	ºC
Logarithmic mean temperature difference, LMTD	30.44	ºC
LMTD correction factor	2.96	ºC

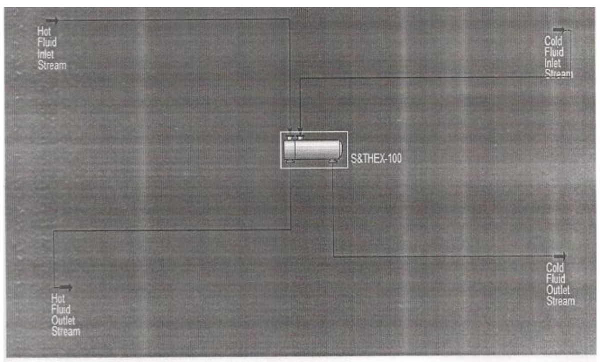


Figure 1: HYSYS Generated Shell and Tube Heat Exchanger Design.

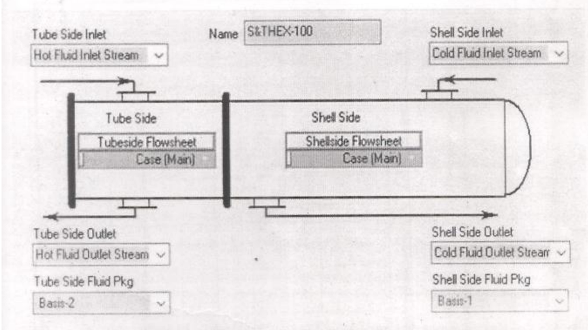
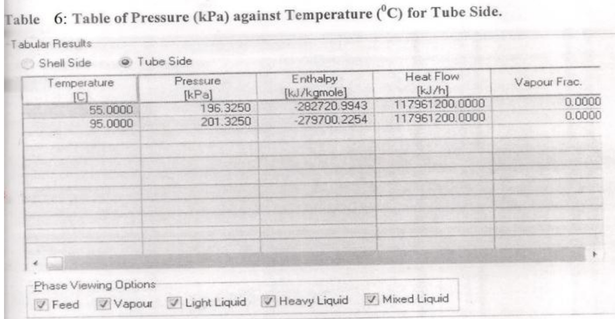


Figure 2: Design Layout of Shell & Tube Heat Exchanger.

Table 6: Table of Pressure (kPa) against Temperature (^{0}C) for Tube Side.



Tabular Results

Shell Side Tube Side

Temperature [C]	Pressure [kPa]	Enthalpy [kJ/kgmole]	Heat Flow [kJ/h]	Vapour Frac.
55.0000	196.3250	-282720.9943	117961200.0000	0.0000
95.0000	201.3250	-279700.2254	117961200.0000	0.0000

Phase Viewing Options: Feed, Vapour, Light Liquid, Heavy Liquid, Mixed Liquid

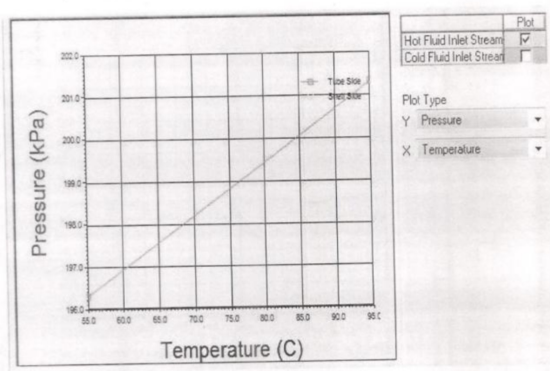


Figure 3: Graph of Pressure against Temperature for Hot Fluid Inlet Stream.

Table 7: Table of Pressure (Psia) against Temperature (^{0}F) for Shell Side.

Tabular Results

Shell Side Tube Side

Temperature [F]	Pressure [psia]	Enthalpy [Btu/lbmole]	Heat Flow [Btu/hr]	Vapour Frac.
95.0000	108.9704	-122192.0349	111805618.9043	0.0000
123.8000	98.8178	-121674.4544	111805618.9043	0.0000

Phase Viewing Options: Feed, Vapour, Light Liquid, Heavy Liquid, Mixed Liquid

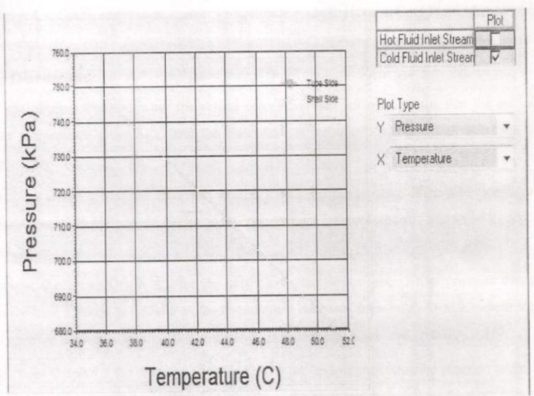


Figure 4: Graph of Pressure against Temperature for Cold Fluid Inlet Stream Only.

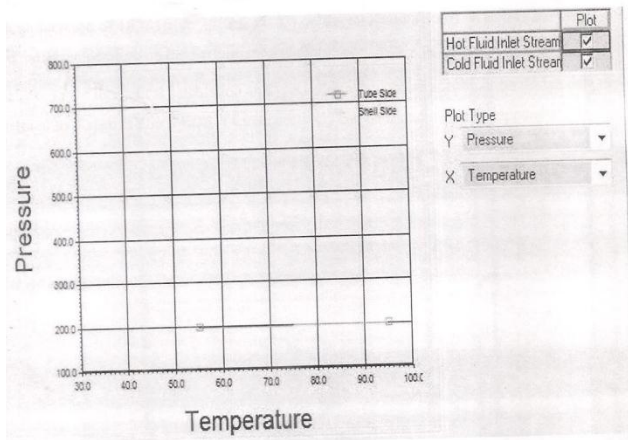


Figure 5: Graph of Pressure against Temperature for both Hot and Cold Fluid Inlet Streams

4.3: Discussion

We can observe from both our theoretical calculation and software results that the cold fluid outlet temperature gives 51°C and the fluid flow is turbulent for both shell and tube sides respectively.

The above Figure 3 for hot fluid inlet streams shows that the pressure of the tube side increases linearly from 196.4kPa to 201.4kPa as the temperature increases from 55°C to 95°C and vice versa.

Likewise, the above Figure 4 for the cold fluid inlet stream shows that pressure decreases linearly from 75OkPa to 68OkPa as the temperature increases from 34°C to 51°C which signify that pressure is inversely proportion at the shell side on the performance section.

Also, from Figure 5 for both hot and cold fluid inlet streams show that the pressure of the tube side remains constant and unaffected at 200kPa as the temperature increases from 55°C to 95°Cwhile the pressure of the shell side shows a decrease of about 6OkPa (from 75OkPa to 69OkPa) temperature with an increase of 16°C (from 35°C to 51°C) on the temperature axis, which is our target cold fluid outlet temperature on the performance section.

Conclusion

On the basis of the above study, it is clear that a lot of factors affect the performance of a shell and tube heat exchanger and the effectiveness obtained by the heat transfer analysis depicts the cumulative effect of all the factors over the performance of the shell and tube heat exchanger. It was observed that the effectiveness of the shell and tube heat exchanger is better when the hot fluid flows into the tubes as compared to the condition in which it had to be flown through the shell, thereby eliminating heat loss by conduction through the shell. Moreover, it means that flowing cold water through the shell side may be beneficial, however, sure comments cannot be made about this universally as it is a very special case of equal heat capacity rates of both the fluid. Increase in mass velocity and Reynolds number at both shell and tube sides are related with the increase in the turbulence and consequently an increase in the heat transfer rates. Whereas the square pitch layout was considered in this study, different tube bundle layout patterns and configurations such as triangular pitch, rotated triangular pitch and rotated square pitch for heat transfer analysis on shell and tube heat exchanger can be further explored.

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