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Comparison of Shell and Tube Heat Exchanger using Theoretical Methods, HTRI, ASPEN and SOLIDWORKS simulation softwares

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ABSTRACT
The aim of this article is to compare the design of Shell and Tube Heat Exchanger with baffles. Baffles used in shell and tube heat exchanger improve heat transfer and also result in increased pressure drop. Shell and tube heat exchanger with single segmental baffles was designed with same input parameters using 1) Kern's theoretical method; 2) ASPEN simulation software and 3) HTRI simulation software 4) SOLIDWORKS simulation software. Shell side pressure drop and heat transfer coefficient are predicted. The results of all the three methods indicated the results in a close range. The proven theoretical methods are in good agreement with the simulation results.
***Keywords* –** ASPEN, HTRI, Kern's theoretical method, Segmental baffles, Shell and Tube Heat Exchanger

I. INTRODUCTION

For the past few decades, shell and tube exchangers are widely used in many engineering applications, such as chemical engineering processes, power generation, petroleum refining, refrigeration, air-conditioning, food industry, etc. Shell and tube heat exchangers are relatively simple to manufacture, and have multi-purpose application possibility when compared with other types of Heat exchangers. It was reported that more than 30% of the heat exchangers in use are of the shell-and-tube type.

Baffles play a significant role in Shell and tube heat exchanger assembly. They provide support for tubes, enable a desirable velocity to be maintained for the shell-side fluid flow, and prevent the tubes from vibrating. Baffles also guide the shell-side flow to move forward across the tube bundle, increasing fluid velocity and heat transfer coefficient. If one takes the most commonly used single segmental baffles as an example, heat transfer is improved as the baffles guide the shell side fluid to flow in a zigzag pattern between the tube bundle, which enhances the turbulence intensity and the local mixing.

Gaddis D [1] reported that the 9th edition of standards and design recommendations of Tubular Exchanger Manufacturers Association (TEMA) was released in 2007.

Kern method [2] and Bell–Delaware method [3] are the most commonly used correlations based approaches for designing the shell side. While Kern method gives conservative results, suitable for the preliminary sizing, Bell–Delaware method is a detailed accurate in estimating heat transfer coefficient and the pressure drop on the shell side for common geometric arrangements. Bell–Delaware method can indicate the existence of possible weaknesses in the shell side design, but cannot point out where these weaknesses are.

Gaddis and Gnielinski [4] studied the pressure drop on the shell side of STHX with segmental baffles.

Karno and Ajib [5] reported from their studies on baffle spacing that baffle cut and baffle spacing are the most important geometric parameters that effect pressure drop as well as heat transfer coefficient on the shell side of a STHX.

Bin Gao et al [6] carried out experimental studies on discontinuous helical baffles at different helical angles of 8^o, 12^o, 20^o , 30^o and 40^o and reported that the performance of baffle at 40^o helix angle was the best among those tested.

Sirous et al [7] replaced a segmental tube bundles by a bundle of tubes with helical baffles in a shell and tube heat exchanger to reduce pressure drop and fouling and hence reduce maintenance and operating cost in Tabriz Petroleum Company.

Farhad et al [8] reported from simulation studies that for same helix angle of 40^{o}and same mass flow rate, heat transfer per unit area decreases with increase in baffle space. However, for same pressure drop, the most extended baffle space obtains higher heat transfer. Pressure gradient decreases with increase in baffle space.

Yonghua et al [9] developed a numerical model of STHX based on porosity and permeability considering turbulence kinetic energy and its dissipation rate. The numerical model was solved over a range of Re from 6813 to 22,326 for the shell side of a STHX with flower baffles. Simulations results agreed with that of experiments with error less than 15%.

Yingshuang et al [10] carried out experimental investigations on flower baffled STHX and the original segmental baffle STHX models and reported that the overall performance of the flower baffled heat exchanger model is 20–30% more efficient than that of the segmental baffle heat exchanger under same operating conditions.

Edward et al [11] presented the procedure for evaluating the shell side pressure drop in shell-and-tube heat exchangers with segmental baffles. The procedure is based on correlations for calculating the pressure drop in an ideal tube bank coupled with correction factors, which take into account the influence of leakage and bypass streams, and on equations for calculating the pressure drop in a window section from the Delaware method.

Young et al [12] reported from simulation studies on STHX with helical baffles using commercially available CFX4.2 codes and concluded that the performance of STHX with helical baffles is superior to that of a conventional STHX. Fluid is in contact with the tubes flowing rotationally in the shell and hence reduced the stagnation zones in the shell side, thereby improving heat transfer.

Sparrow & Reifschneider [13], Eryener [14], Karno & Ajib [15] carried out studies on the effects of baffle spacing in a STHX on pressure drop and heat transfer.

Li and Kottke [16,17] and Karno and Ajib [18] carried out investigations on the effect of tube arrangement in STHX from heat transfer view point.

From literature review, it is observed that different studies on heat transfer coefficient and pressure drop in STHX with different baffle shape, spacing, and tube spacing have been carried out. It is observed that comparison of theoretical design methods of STHX with that of simulations using software have not been done.

II. DESIGN OF SHELL AND TUBE HEAT EXCHANGER

A shell and tube heat exchanger with single segmented baffles is designed. Single segmented baffle are chosen as they are the most widely used, large data is available and hence can be theoretically designed.

A water-water 1-2 pass shell and tube heat exchanger is designed considering the data in the following Table 1.

Table 1 Data for design of heat exchanger

Shell Side Fluid-Hot Water		
Property	Unit	Value
T_{HI}	°C	90
T_{HO}	°C	70
Density	kg/m^3	971.8
Specific Heat Capacity	kJ/kgK	4.1963
Viscosity	mPas	0.354
Conductivity	W/mK	0.67
Fouling Factor	-	0.0002
Flow Rate	kg/s	0.3
Tube Side Fluid-Cold Water		
T_{CI}	°C	30
T_{CO}	°C	38
Density	kg/m^3	984
Specific Heat Capacity	kJ/kgK	4.178
Viscosity	mPas	0.725
Conductivity	W/mK	0.623
Fouling Factor	-	0.0002
Flow Rate	kg/s	0.7533

Hot fluid is considered to flow in the shell as a thumb rule says that fluid with low flow rate should always be in shell side. A vice versa heat exchanger was also designed which was inferior with respect to hot fluid shell side design. Thus, confirming the thumb rule. With the above basic data a shell and tube heat exchanger was designed by
1) Theoretical Method (Kern's Method).
2) ASPEN Simulation Software.
3) HTRI Simulation Software
4) Solidworks Simulation Software.

2.1 Design of STHX by Kern's Theoretical Method:

This method is employed as it is simple to use and the design is reliable. All the empirical equations in this section are as proposed by Donald Q. Kern.

Design of heat exchanger with this method is illustrated as follows:

Logarithmic Mean Temperature Difference LMTD is calculated as:

$$(\Delta T_{lm}) = \frac{(T_{Hi} - T_{Co}) - (T_{Ho} - T_{Ci})}{\ln\left(\frac{(T_{Hi} - T_{Co})}{(T_{Ho} - T_{Ci})}\right)} \quad (1)$$

$$= 45.74°C$$

For One shell pass and two tube passes,

$$R = \frac{(T_{Hi} - T_{Ho})}{(T_{Co} - T_{Ci})} = 2.5 \quad (2)$$

$$S = \frac{(T_{Co} - T_{Ci})}{(T_{Hi} - T_{Ci})} = 0.133 \quad (3)$$

LMTD correction factor is read from graph given by Kern D.Q. [2] for one shell pass and two or more tube passes using R and S values as

$F_t = 0.99$

$$\text{Corrected } \Delta T_{lm} = F_t \Delta T_{lm} \quad (4)$$

$= 0.99 \times 45.74 = 45.15$°C

It is assumed that U = 785W/m^2K

Heat Load is given by:

$$(Q) = mC\Delta T \quad (5)$$

$= 0.3 \times 4.1963 \times (90-70) = 25.18$kW

Provisional Area is given by:

$$A = \frac{Q}{U \Delta T_{lm}} \quad (6)$$

$$= \frac{25180}{785 \times 45.15} = 0.71m^2$$

Choose 21.34mm OD, 18.04mm ID, 1.068m long Copper tubes.

Allowing for tube-sheet thickness, take L = 1.038m

$$\text{Area of one tube} = \pi d_o L \quad (7)$$

$= \pi \times 0.02134 \times 1.038 = 0.0696m^2$

Number of tubes N is given by

$$(N) = \frac{0.71}{0.0696} = 10 \quad (8)$$

1.35 triangular pitch is used to maintain good ligament

Bundle Diameter D_b is given by

$$(D_b) = d_o \left(\frac{N}{0.249} \right)^{\frac{1}{2.207}} \quad (9)$$

$$21.34 \left(\frac{10}{0.249} \right)^{\frac{1}{2.207}} = 113.73mm$$

Fixed U-tube Head is used. From FigureA3, Bundle diametrical Clearance = 10mm

Shell diameter (D_s) = D_b + 10 = 113.73 + 10 = 123.73mm

Nearest Standard Pipe size of 168.28mm is considered as Shell Diameter.

1.1.1 Prediction of Tube Side Heat Transfer Coefficient

Tube cross-sectional area is given by

$$\frac{\pi}{4} \times d_i^2 = \frac{\pi}{4} \times 18.04^2 = 255.6mm^2 \quad (10)$$

$$\text{Tubes per pass} = \frac{N}{2} = \frac{10}{2} = 5$$

Total Flow Area = $5 \times 255.6 = 1.278 \times 10^{-3}m^2$

$$\text{Cold Water mass velocity} = \frac{0.753}{1.278 \times 10^{-3}}$$

$= 597.3$kg/sm^2

$$\text{Linear velocity (u)} = \frac{597.3}{984} = 0.6m/s$$

$$Re = \frac{\rho u d_i}{\mu} = \frac{984 \times 0.6 \times 18.04 \times 10^{-3}}{0.725 \times 10^{-3}}$$

$= 14666.3$

$$Pr = \frac{C\mu}{k} = \frac{4.178 \times 0.725 \times 10^{-3}}{0.623} = 4.86$$

$$\frac{L}{d_i} = \frac{1038}{18.04} = 57.54$$

$J_h = 4 \times 10^{-3}$ is taken from graph given by Kern D.Q. [2]

$$h_i = \frac{J_h \, Re \, Pr^{0.33} \, k}{d_i} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (11)$$

$$= \frac{4 \times 10^{-3} \times 14666.3 \times 4.86^{0.33} \times 0.623}{18.04 \times 10^{-3}} (0.9)^{0.14}$$

$= 3072.3$W/m^{2o}C

1.1.2 Prediction of Shell Side Heat Transfer Coefficient:

Baffle Spacing (B) = 50.8mm

Tube Pitch (P_t) =1.35 × d_i = 1.35 × 21.34 = 28.8mm

Cross Flow Area (A_s) is given by:

$$\left(\frac{P_t - d_o}{P_t} \right) \times D_s \times B \quad (12)$$

$$= \left(\frac{28.8 - 21.34}{28.8} \right) \times 168.3 \times 50.8 \times 10^{-6}$$

$= 2.2146 \times 10^{-3}m^2$

$$\text{Hot water mass velocity} = \frac{0.3}{2.2146 \times 10^{-3}}$$

$= 135.47$kg/sm^2

Equivalent Diameter is given by

$$de = \frac{1.1}{d_i} \left(P_t^2 - 0.917 \, (21.34)^2 \right) = 21.23mm$$

$$Re = \frac{\rho u d_e}{\mu} = \frac{135.47 \times 21.23 \times 10^{-3}}{0.354 \times 10^{-3}} = 8124$$

$$Pr = \frac{C\mu}{k} = \frac{4.1963 \times 0.0.354 \times 10^{-3}}{0.67} = 2.22$$

Choose 29% baffle cut, from figureA4, $J_h = 7 \times 10^{-3}$

$$h_s = \frac{J_h \, Re \, Pr^{0.33} \, k}{d_e} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

$$= \frac{7 \times 10^{-3} \times 8124 \times 2.22^{0.33} \times 0.67}{21.23 \times 10^{-3}} (0.9)^{0.14}$$

$= 2101.5 W/m^{2o}C$

1.1.3 Prediction of Overall Heat Transfer Coefficient:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_s} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2 \times k_s} + \frac{d_o}{d_i}(2 \times F) \tag{13}$$

$$\frac{1}{U} = \frac{1}{3072.3} + \frac{1}{2101.5} + \frac{21.34 \times 10^{-3} \ln\left(\frac{21.34}{18.04}\right)}{2 \times 385} + \frac{21.34}{18.04}(2 \times 0.0002)$$

$U = 782 W/m^{2o}C$

Well near the assumed value of $785 W/m^{2o}C$

1.1.4 Prediction of Pressure Drop on Tube side

From graph given by Kern DQ [2], for Re = 14666.3

$J_f = 5 \times 10^{-3}$

$$\Delta P = N_p \left(8 J_f \left(\frac{L}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14} + 2.5 \right) \frac{\rho u^2}{2} \tag{14}$$

$= 1.8 kPa$

1.1.5 Prediction of Pressure drop on Shell-Side

From graph given by Kern DQ [2], at Re = 8124

$J_f = 4.5 \times 10^{-2}$

$$\Delta P = \left(8 J_f \left(\frac{D_s}{d_e}\right)\left(\frac{L}{B}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14} \right) \frac{\rho u^2}{2} \tag{15}$$

$= 64.77 Pa$

The results of this method are the

1. Overall Heat Transfer Coefficient $U= 782 W/m^2C$
2. Tube-side Pressure Drop $\Delta P = 1.8 kPa$
3. Shell-side Pressure Drop $\Delta P = 64.77 Pa$.

2.2 Design of STHX using ASPEN simulation software:

This software can be used to design, rate, simulate and do cost prediction of a heat exchanger. Here ASPEN is used to simulate the heat exchanger designed by Kern's theoretical method. In simulation mode of this software all the data related to geometry of heat exchanger and the properties of fluids are to be stated as input to the software. Flow rates and input temperatures of the fluid streams are also to be stated. The software then gives output in terms of the output temperature attained by the streams. It generates a specification sheet called TEMA sheet which indicates the overall Heat transfer coefficient, Pressure Drop in both shell-side and tube-side and many other parameters involved in heat exchanger design.

The input for ASPEN simulation software in this case is as shown in the following Table 2,

Table2 Input to ASPEN simulation Software

I. Problem Definition		
A. Application Options		
1. General		
Calculation Mode	Simulation	
Location of Hot fluid	Shell-Side	
Select Geometry Based on	SI standards	
Calculation Method	Advanced method	
2. Hot side		
Application	Liquid, no phase change	
Simulation Calculation	Output temperature	
3. Cold side		
Application	Liquid, no phase change	
Simulation Calculation	Output temperature	
B. Process Data		
Fluid Name	Shell-Side hot water	Tube-Side cold water
Mass flow rate (kg/s)	0.3	0.753
Inlet Temperature (°C)	90	30
Operating Pressure abs (bar)	1	1
Fouling Resistance (m^2K/W)	0.0002	0.0002
I. Property Data		
Properties of fluids were imported form ASPEN database		
I. Exchanger Geometry		
A. Shell/Heads		
Front Head Type	B-bonnet bolted or integral tube-sheet	
Shell Type	E-one pass shell	
Rear Head Type	U – U-tube bundle	
Exchanger Position	Horizontal	
Shell Inner diameter (mm)	154.05	
B. Tube		
Number of Tubes	10	
Number of Tubes Plugged	0	
Tube length (mm)	1038	
Tube Type	Plain	
Tube Outside Diameter (mm)	21.34	
Tube wall Thickness (mm)	1.65	
Tube Pitch (mm)	28.8	
Tube Pattern	45°	
Tube Material	Copper	
C. Baffles		
Baffle Type	Single Segmental	
Baffle Cut (%)	29	
Baffle Orientation	Horizontal	
Baffle Thickness (mm)	3.2	
Baffle Spacing (mm)	50.8	
Number of Baffles	16	
D. Nozzles		
Outside diameter of shell side Inlet nozzle (mm)	26.645	
Inside diameter of shell side Inlet nozzle (mm)	26.645	
Outside diameter of tube side Inlet nozzle (mm)	26.645	
Inside diameter of tube side Inlet nozzle (mm)	26.645	
V. Construction Specifications		
A. Materials of Construction		
Shell	Carbon Steel	
Tube-Sheet	Carbon Steel	
Baffles	Carbon Steel	
Heads	Carbon Steel	
Nozzle	Carbon Steel	
Tube	Copper	
B. Design Specifications		

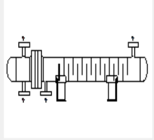
1. Codes and Standards	
Design Code	ASME Code Sec VIII Div 1
Service Class	Refinery Service
TEMA Class	C-General Class
Material Standard	ASME
Dimensional Standard	ANSI - American

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 152.4 - 1038	mm	Type BEU Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 0.7	m²	Shells/unit 1		Surf/shell (eff.)	0.7 m²
8	PERFORMANCE OF ONE UNIT					
9	Fluid allocation		Shell Side		Tube Side	
10	Fluid name		hot water		cold water	
11	Fluid quantity, Total	kg/s	0.3		0.7533	
12	Vapor (In/Out)	kg/s	0	0	0	0
13	Liquid	kg/s	0.3	0.3	0.7533	0.7533
14	Noncondensable	kg/s	0	0	0	0
15						
16	Temperature (In/Out)	°C	90	70.08	30	37.97
17	Dew / Bubble point	°C				
18	Density Vapor/Liquid	kg/m³	/ 971.8	/ 971.8	/ 984	/ 984
19	Viscosity	mPa s	/ 0.354	/ 0.354	/ 0.725	/ 0.725
20	Molecular wt, Vap					
21	Molecular wt, NC					
22	Specific heat	kJ/(kg K)	/ 4.196	/ 4.196	/ 4.178	/ 4.178
23	Thermal conductivity	W/(m K)	/ 0.67	/ 0.67	/ 0.623	/ 0.623
24	Latent heat	kJ/kg				
25	Pressure (abs)	bar	1	0.98743	1	0.97673
26	Velocity	m/s	0.17		0.75	
27	Pressure drop, allow./calc.	bar	0.11	0.01257	0.20684	0.02327
28	Fouling resistance (min)	m² K/W	0.0002		0.0002	0.00024 Ao based
29	Heat exchanged 25.1	kW			MTD corrected	45.21 °C
30	Transfer rate, Service 790.2		Dirty 790.2		Clean 1206.4	W/(m² K)

Figure 1 Heat Exchanger Specification sheet by ASPEN Simulation.

31	CONSTRUCTION OF ONE SHELL				Sketch
32			Shell Side	Tube Side	
33	Design/vac/test pressure:g	bar	3.44738/ /	3.44738/ /	
34	Design temperature	°C	126.67	126.67	
35	Number passes per shell		1	2	
36	Corrosion allowance	mm	3.18	0	
37	Connections	In mm	1 19.05 / -	1 25.4 / -	
38	Size/rating	Out	1 19.05 / -	1 25.4 / -	
39	Nominal	Intermediate	/ -	/ -	



40	Tube No. 5Us OD 21.34	TksAvg 1.65 mm	Length 1038 mm Pitch 28.8 mm
41	Tube type Plain	#/m Material Copper	Tube pattern 30
42	Shell Carbon Steel ID 154.05 OD 168.12 mm		Shell cover Carbon Steel
43	Channel or bonnet Carbon Steel		Channel cover -
44	Tubesheet-stationary Carbon Steel -		Tubesheet-floating -
45	Floating head cover -		Impingement protection None
46	Baffle-cross Carbon Steel	Type Single segmental Cut(%d) 29.22 H	Spacing: c/c 50.8 mm
47	Baffle-long -	Seal type	Inlet 0 mm
48	Supports-tube U-bend 0	Type	
49	Bypass seal	Tube-tubesheet joint	Exp. 2 grv
50	Expansion joint -	Type None	
51	RhoV2-Inlet nozzle 1190	Bundle entrance 15	Bundle exit 1 kg/(m s²)
52	Gaskets - Shell side Flat Metal Jacket Fibe	Tube Side	Flat Metal Jacket Fibe
53	Floating head -		
54	Code requirements ASME Code Sec VIII Div 1		TEMA class R - refinery service
55	Weight/Shell 122.9	Filled with water 141.2	Bundle 20.2 kg

Figure 2 TEMA Construction details of Shell and Tube Heat Exchanger given by ASPEN Simulation

The output of APSEN Simulation software gives the specification sheet shown in Fig. 1 and TEMA specification sheet shown in Fig. 2.

2.3 Design of STHX HTRI Simulation Software:

This software can be used to design, rate and simulate a heat exchanger. Here HTRI is used to simulate the heat exchanger designed by Kern's theoretical method. In simulation mode of this software all the data related to geometry of heat exchanger and the properties of fluids are to be stated as input to the software. Flow rates and input temperatures of the fluid streams are also to be stated. The software then gives output in terms of the output temperature attained by the streams. It generates a specification sheet called TEMA sheet which indicates the overall Heat transfer coefficient, Pressure Drop in both shell-side and tube-side and many other parameters involved in heat exchanger design. This Software also provides necessary drawings of the heat exchanger.

The input for HTRI simulation software in this case is as shown in the following Table 3.

Table 3 Input data to HTRI Simulation Software

I. Case Mode	Simulation	
II. Exchanger Service	Generic Shell and Tube	
III. Process Conditions		
Fluid Name	Shell-Side hot water	Tube-Side cold water
Mass flow rate (kg/s)	0.3	0.753
Inlet Temperature (°C)	90	30
Operating Pressure abs (bar)	1	1
Fouling Resistance (m^2K/W)	0.00 02	0.000 2
V. Shell Geometry		
TEMA Type	B-E-U	
Shell ID (mm)	154.05	
Orientation	Horizontal	
Hot Fluid	Shell Side	
V. Baffle Geometry		
Type	Single Segmental	
Orientation	Perpendicular	
Baffle Cut (%)	29	
Baffle Spacing (mm)	50.8	
Baffle Thickness (mm)	3.2	
Crosspasses	17	
VI. Tube Geometry		
Type	Plain	
Length (m)	1.038	
Tube OD (mm)	21.34	
Wall Thickness (mm)	1.65	
Pitch (mm)	28.8	
Layout Angle	45°	
Tube Pass	2	
Tube Count	10	
Tube Material	Copper	
VII. Nozzles		
Standards	ANSI	
Outside diameter of shell side Inlet nozzle (mm)	26.645	
Inside diameter of shell side Inlet nozzle (mm)	26.645	
Outside diameter of tube side Inlet nozzle (mm)	26.645	
Inside diameter of tube side Inlet nozzle (mm)	26.645	
Inlet Type	Radial	
Outlet Type	Radial	
Radial Position of inlet nozzle on shell	Top	
Longitudinal Position of inlet nozzle on shell	At Rear Head	
Radial Position of inlet nozzle on shell	Opposite Side	
Location of nozzle at U-bend	Before U-bend	
Number at each location	1	
VIII. Property Data		

Properties of fluids were imported form HTRI database

The output of HTRI Simulation software gives the specification sheet shown in Fig. 3 and TEMA specification sheet shown in Fig. 4.

HTRI — HEAT EXCHANGER SPECIFICATION SHEET — Page 1 — SI Units

		Job No.		
Customer		Reference No.		
Address		Proposal No.		
Plant Location		Date 11/10/2015	Rev	
Service of Unit		Item No.		
Size 154.050 x 1037.97 mm	Type BEU Horz.	Connected In	1 Parallel	1 Series
Surf/Unit (Gross/Eff) 0.70 / 0.68 m2	Shell/Unit 1	Surf/Shell (Gross/Eff) 0.70 / 0.68 m2		
PERFORMANCE OF ONE UNIT				
Fluid Allocation	Shell Side		Tube Side	
Fluid Name	hot water		cold water	
Fluid Quantity, Total kg/hr	1080.01		2711.89	
Vapor (In/Out)				
Liquid	1080.01	1080.01	2711.89	2711.89
Steam				
Water	1080.01	1080.01	2711.89	2711.89
Noncondensables				
Temperature (In/Out) C	90.00	70.84	30.00	37.66
Specific Gravity	0.9722	0.9722	0.9844	0.9844
Viscosity mN-s/m2	0.3540	0.3540	0.7250	0.7250
Molecular Weight, Vapor				
Molecular Weight, Noncondensables				
Specific Heat kJ/kg-C	4.1964	4.1964	4.1781	4.1781
Thermal Conductivity W/m-C	0.6702	0.6702	0.6232	0.6232
Latent Heat kJ/kg				
Inlet Pressure kPa	100.001		100.001	
Velocity m/s	9.572e-2		0.60	
Pressure Drop, Allow/Calc kPa	100.002	0.563	100.002	3.021
Fouling Resistance (min) m2-K/W	0.000200		0.000200	
Heat Exchanged W 24116.9		MTD (Corrected)	45.0 C	
Transfer Rate, Service 781.91 W/m2-K	Clean	1199.87 W/m2-K	Actual	787.39 W/m2-K

Figure 3 Heat Exchanger Specification sheet by HTRI Simulation.

CONSTRUCTION OF ONE SHELL				Sketch (Bundle/Nozzle Orientation)
		Shell Side	Tube Side	
Design/Test Pressure	kPaG	1034.21 /	1034.21 /	
Design Temperature	C			
No Passes per Shell		1	2	
Corrosion Allowance	mm			
Connections Size & Rating	In mm	1 @ 26.645	1 @ 26.645	
	Out mm	1 @ 26.645	1 @ 26.645	
	Intermediate	@	@	

Tube No. 5U	OD 21.340 mm	Thk(Avg) 1.650 mm	Length 1.038 m	Pitch 28.800 mm	Layout 45
Tube Type	Plain		Material COPPER		
Shell	ID 154.050 mm	OD mm	Shell Cover		
Channel or Bonnet			Channel Cover		
Tubesheet-Stationary			Tubesheet-Floating		
Floating Head Cover			Impingement Plate	None	
Baffles-Cross	Type SINGLE-SEG.	%Cut (Diam) 29.0	Spacing(c/c) 50.800	Inlet 50.380 mm	
Baffles-Long		Seal Type			
Supports-Tube		U-Bend	Type		
Bypass Seal Arrangement		Tube-Tubesheet Joint			
Expansion Joint		Type			
Rho-V2-Inlet Nozzle	297.87 kg/m-s2	Bundle Entrance 27.17	Bundle Exit 1.57	kg/m-s2	
Gaskets-Shell Side		Tube Side			
-Floating Head					
Code Requirements			TEMA Class		
Weight/Shell 107.25		Filled with Water 132.71	Bundle 18.74	kg	

Figure 4 TEMA Construction details of Shell and Tube Heat Exchanger given by HTRI Simulation

2.3 Design of STHX using Solidworks Flow Simulation Software:

A commercially available CFD code (SOLIDWORKS FLOW SIMULATION) has been used to carry out the numerical calculations for the studied geometries. A three dimensional geometrical model of the problem is developed with SOLIDWORKS software. Mesh generation is done. The physical model is presented in Fig. 5. The tube material is Copper while the other components are carbon steel. The physical properties of carbon steel and copper are taken from the SOLIDWORKS database. Thermal properties of water are also taken from the SOLIDWORKS database.

The water inlet boundary conditions are set as Flow opening inlets and outlet boundary conditions are set as Pressure opening outlets. The exterior wall is modeled as adiabatic. The simulation is solved to predict the heat transfer and fluid flow characteristics by using *k-ε* turbulence model.

Following are the boundary conditions assumed:

1) Shell Side Inlet was set as Flow opening the mass flow rate varied from 0.1kg/s to 0.5kg/s for different simulations and temperature was set to 363.15K.
2) Tube Side Inlet was set to Flow opening the mass flow rate was set to 0.7533kg/s and the temperature was set to 303.15K.
3) Both shell side and tube side were set as Pressure openings with pressure set to Atmospheric Pressure.

Figures 6, 7 and 8 show the variations in pressure, temperature, and velocity within the STHX with single segmental baffles simulated using Solidworks Simulation software.

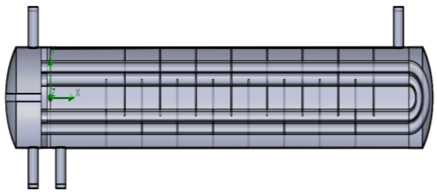


Figure 5 2D view of the Shell and Tube Heat Exchanger designed

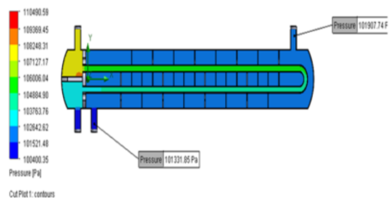


Figure 6 Pressure variation in STHX

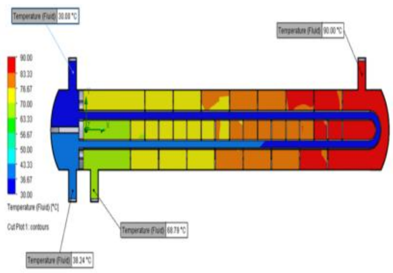


Figure 7 Temperature variatiion in STHX

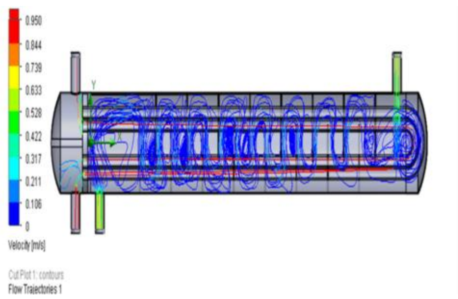


Figure 8 Velocity variation in STHX

III. RESULTS AND DISCUSSION

Table 4 shows the variations in the Overall Heat transfer coefficient, Shell side outlet temperature, and shell side temperature difference.

Table 4 Comparison of Overall Heat Transfer Coefficient, Shell side outlet temperature and Shell side temperature difference predictions

Heat Exchanger Design Method	Outlet Temperature °C	Overall HTC W/m^{2}K	Temperature Difference °C
Kern's method	70	782	20
ASPEN Simulation	70.08	790.2	19.92
HTRI Simulation	70.84	781.91	19.16
CFD Simulation	68.79	852.46	21.21

It is observed from Fig. 9 that Kern's method, and HTRI simulations have similar values of Overall Heat transfer coefficient, while that obtained from ASPEN simulation is little higher, and that obtained from CFD simulations using Solidworks software is the highest with variation of over 9% when compared to Kern's theoretical method. This is variation in Solidworks software results may be due to better grid convergence of the solution while the theoretical values are based o empirical correlations only.

Similarly, It is observed from Fig. 10 that shell side temperature difference is almost similar with Kern's method and ASPEN method, while that with HTRI simulation showed a lesser value, while that with CFD simulation using Solidworks software is higher by 6%. This variation in Solidworks software results may be owing to improvement in computation capability due to finer meshes in flow field.

Fig. 11 shows that the Shell side outlet temperature is very similar with Kern's method, and APSEN simulation. On the other hand, HTRI simulation is greater by 1.2% while that by Solidworks Simulation is lesser by 1.7%.

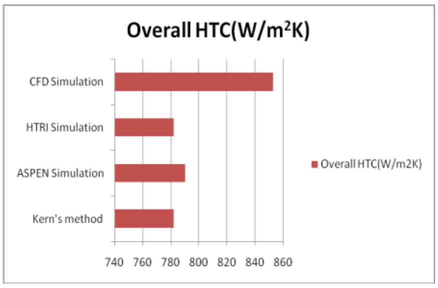


Figure 9 Variation in Overall Heat Transfer coefficient with different design softwares

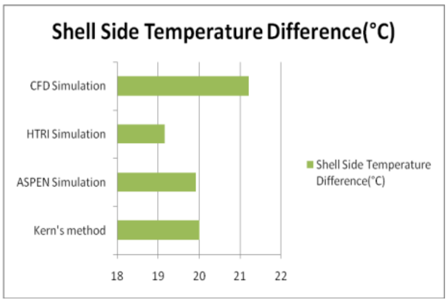


Figure 10 Variation in Shell Side Temperature Difference with different design softwares

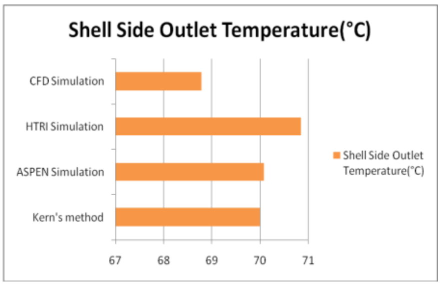


Figure 11 Variation in Tube Side Outlet Temperature with different design softwares

IV. CONCLUSIONS

A Shell and Tube Heat Exchanger was designed with same input parameters using Kern's method, ASPEN simulation software, HTRI simulation software and by SolidWorks Flow Simulation software and the Overall heat transfer coefficient values are 782, 790.2, 781.9 and 852.6 W/m^{2}K respectively. Simulation results of Overall heat transfer coefficient with Kern's method ASPEN and HTRI software are similar while, that with SolidWorks software is greater by 9%. Shell side temperature drop is greater by 6% with Solid works software. All the three Methods obtained almost same results for the same geometry of heat exchanger. Thus, it can be concluded that the results generated with single segmental baffle configuration are real time.

REFERENCES

[1] Gaddis D, editor. Standards of the Tubular Exchanger Manufacturers Association. (Tarrytown (NY): TEMA Inc. 2007).

[2] Kern DQ, Process heat transfer. (New York (NY): McGraw-Hill, 1950).

[3] Bell KJ. Delaware method for shell side design. In: Kakac S, Bergles AE, Mayinger F, editors. Heat exchangers: thermal–hydraulic fundamentals and design. New York: Hemisphere, 1981, 581–618.

[4] Gaddis ES, and Gnielinski V., Pressure drop on the shell side of shell-and-tube heatexchangers with segmental baffles. Chem Eng Process 36, 1997, 149–59.

[5] Karno A, Ajib S., Effects of baffle cut and number of baffles on pressure drop and heat transfer in shell-and-tube heat exchangers – numerical simulation. Int JHeat Exchangers 7, 2006, 299–322.

[6] Bin Gao, Qincheng Bi, Zesen Nie and Jiangbo Wu, Experimental study of effects of baffle helix angle on shell-side performance of shell-and-tube heat exchangers with discontinuous helical baffles, Experimental Thermal and Fluid Science, 68, 2015, 48–57.

[7] Sirous Zeyninejad Movassag, Farhad Nemati Taher, Kazem Razmi and Reza Tasouji Azar, Tube bundle replacement for segmental and helical shell and tube heat exchangers: Performance comparison and fouling investigation on the shell side, Applied Thermal Engineering, 51, 2013, 1162-1169.

[8] Farhad Nemati Taher, Sirous Zeyninejad Movassag, Kazem Razmi and Reza Tasouji Azar, Baffle space impact on the performance of helical baffle shell and tube heat exchangers, Applied Thermal Engineering, 44, 2012, 143-149.

[9] Yonghua You, Aiwu Fan, Suyi Huang and Wei Liu, Numerical modelling and experimental validation of heat transfer and flow resistance on the shell side of a shell-and-tube heat exchanger with flower baffles, International Journal of Heat and Mass Transfer, 55, 2012. 7561–7569.

[10] Yingshuang Wang, Zhichun Liu, Suyi Huang, Wei Liu and Weiwei Li, Experimental investigation of shell-and-tube heat exchanger with a new type of baffles, Heat and Mass Transfer, 47, 2011, 833–839.

[11] Edward S. Gaddis and Volker Gnielinski Pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles, Chemical Engineering and Processing, 36, 1997, 149-159.

[12] Young-Seok Son and Jee-Young Shin, Performance of a shell-and-tube heat exchanger with spiral baffle plates, KSME International Journal, Volume 15(11), 2001, 1555-1562.

[13] Sparrow EM and Reifschneider LG., Effect of interbaffle spacing on heat transfer and pressure drop in a shell-and-tube heat exchanger. Int J Heat Mass Transfer 29, 1986, 1617–1628.

[14] Eryener D., Thermo economic optimization of baffle spacing for shell and tube heat exchangers, Energy Convers Manage 47, 2006, 1478–1489.

[15] Karno A, and Ajib S., Effects of baffle cut and number of baffles on pressure drop and heat transfer in shell-and-tube heat exchangers – numerical simulation, Int J Heat Exchangers, 7, 2006, 299–322.

[16] Li HD and Kottke V., Visualization and determination of local heat transfer coefficients in shell-and-tube heat exchangers for staggered tube arrangement by mass transfer measurements, Exp Therm Fluid Sci, 17, 1998,210–6.

[17] Li HD, and Kottke V., Visualization and determination of local heat transfer coefficients in shell-and-tube heat exchangers for in-line tube arrangement by mass transfer measurements, Heat Mass Transfer, 33, 1998371–6.

[18] Karno A, and Ajib S., Effect of tube pitch on heat transfer in shell-and-tube heat exchangers – new simulation software, Heat Mass Transfer, 42, 2006, 263–70.

NOMENCLATURE

A	Area (m^2)
A_s	Cross Flow Area (m^2)
B	Baffle Spacing (m)
C	Specific Heat Capacity ($J\ kg^{-1}\ K^{-1}$)
D_b	Bundle Diameter (m)
D_i	Inside diameter of shell (m)
D_s	Outside diameter of shell (m)
d_e	Equivalent Diameter (m)
d_i	Inside diameter of tube (m)
d_o	Outside diameter of tube (m)
F	Fouling Factor.
F_t	Log Mean Temperature Difference Correction Factor
h	Enthalpy ($J\ kg^{-1}K^{-1}$)
h_i	Tube side Film Heat Transfer Coefficient ($W\ m^{-2}\ K^{-1}$)
h_s	Shell side Film Heat Transfer Coefficient ($W\ m^{-2}\ K^{-1}$)
J_f	Friction Factor
J_h	Heat Transfer Factor
k	Thermal Conductivity, Turbulent kinetic energy.
L	Length (m)
m	Mass Flow Rate ($kg\ s^{-1}$)
N	Number of tubes.
N_p	Number of tube side passes
P_{in}	Pressure at inlet of the shell
P_{out}	Pressure at outlet of the shell (
ΔP	Pressure Drop.
P_t	Pitch.
Q	Heat Load.
T_{Ci}	Tube side fluid inlet temperature.
T_{Co}	Tube side fluid outlet temperature.
T_{Hi}	Shell side fluid inlet temperature.
T_{Ho}	Shell side fluid outlet temperature.

ΔT_{lm} Log Mean Temperature Difference.
t Time.
U Overall Heat Transfer Factor.
u Velocity.
Le Lewis Number.
Re Reynolds number.
Pr Prandtl Number.
x Co-ordinate.
y Co-ordinate.
z Co-ordinate.

Greek Letters
ρ Density.
μ Dynamic Viscosity.
ε Turbulent dissipation energy.