International Engineering Research Journal

Design of Shell and Tube Heat Exchanger for Heat Recovery from Hydraulic Oil

Sunil R Taware[a], Dipak SPatil[b] , Dr.R.R.Arakerimath[c]

[a]ME Student, Department of mechanical engineering., SPU Pune, GHRCEM, Wagholi, Pune, India.
[b]PG Coordinator, Department of mechanical engineering., SPU Pune, GHRCEM, Wagholi, Pune, India.
[c]Head of Department, mechanical engineering., SPU Pune, GHRCEM, Wagholi, Pune, India.

Abstract

As we know that a shell and tube heat exchanger is designed where high pressures and high pressure differences between the fluids relative to the environment are applied. These exchangers are generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. There is too much flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length and arrangement. One fluid flows inside the tubes, and other is across and along the tubes. In this project, the hot fluid will be cooled using tap water with the help of shell and tube heat exchanger. A characteristic of heat exchanger design is the procedure of specifying a design heat transfer area and pressure drops and checking whether the assumed design satisfies all requirement or not. The purpose of the project is how to design the HE which is the majority type of liquid-to-liquid heat exchanger. A Simplified approach to design a Shell & Tube Heat Exchanger [STHE] for hydraulic oil and process industry application is presented. The design of STHE includes thermal design of STHE involves evaluation of required effective surface area (i.e. number of tubes) and finding out log mean temperature difference [LMTD].The design was carried out by referring ASME/TEMA standards

Keywords:*Shell and Tube Heat Exchanger [STHE], ASME, TEMA, LMTD,HE.*

I. INTRODUCTION

A heat exchanger is a devicethat is used for heat transfer of internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat transfer surface and ideally they do not mix. HE are used in process, petroleum, power, transportation, refrigeration,air conditioning, cryogenic, heatrecovery, alternate fuels and other industries. Therelation was formulated by Newton and is called Newton's law of cooling, which is given by

$Q = h*A*dT$

Where, h is heat transfer coefficient [W/m^2K],A is the heat transfer area [m^2], and T is the temperature difference [K].

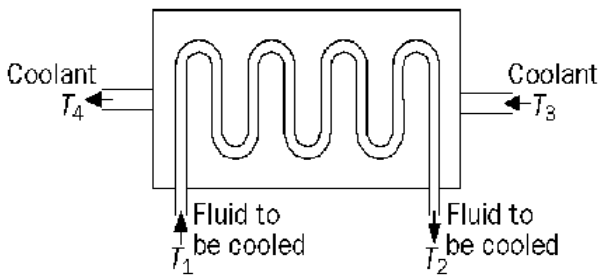


Fig.1: The basic heat transfer mechanism[18]

Shell and tube heat exchangers are used primarily for liquid-to-liquid and liquid-to-phase change heat transfer applications. STHE generally for gas-to-liquid and gas-to-gas heat transfer applications, primarily when the operating temperature and pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchanger's works.

There are so many type of internal constructions are used in STHE 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.

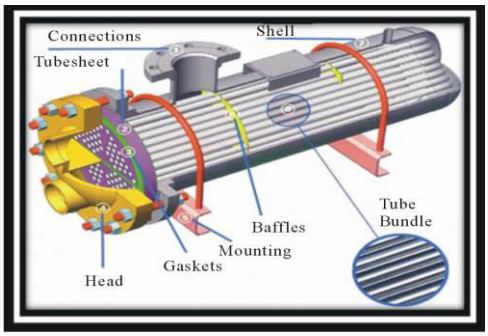


Fig.2: The basicComponents Shell and Tube Heat Exchanger[19]

Tema Standards

STHE are classified and constructed incorporate with the widely used TEMA (Tubular ExchangerManufacturers Association) standards. Notation system used in TEMA to designate major types of combination, the firstletter indicating the front-end head type, the second the shell type, and the third the rear-end head type. There are common STHE are AKT,

AES, BEM, AEP, CFU, and AJW. It should be emphasize that there are several special types of shell and tube heat exchangers economically available that are different from those of above.

Classification Based on TEMA Construction:

Followings are three basic classification based on TEMA, based on their end connection and shell type.

a. BEM b. CFU c. AES

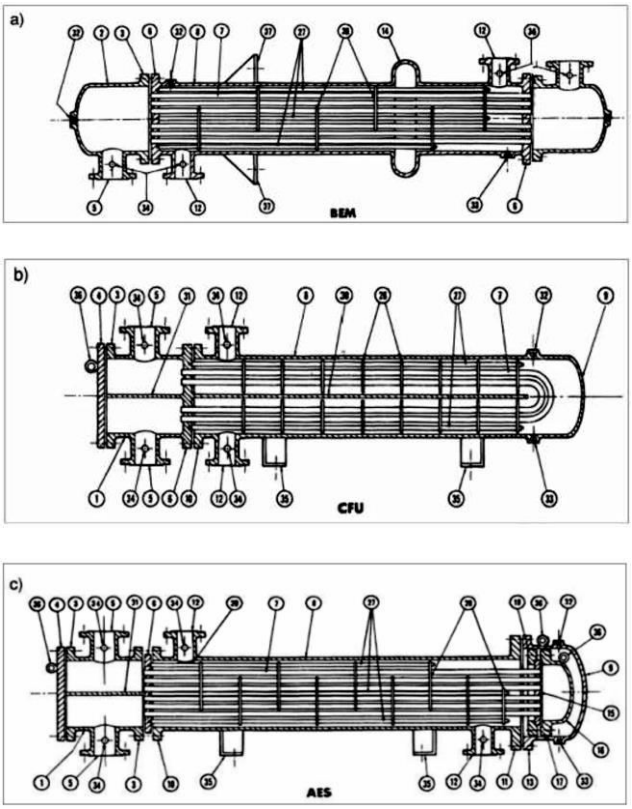


Fig 3: Construction Parts and Connections[10]

1	Stationary (Front) Head—Channel	20	Slip-on Backing Flange
2	Stationary (Front) Head—Bonnet	21	Floating Tubesheet Skirt
3	Stationary (Front) Head Flange	22	Floating Tubesheet Skirt
4	Channel Cover	23	Packing Box Flange
5	Stationary Head Nozzle	24	Packing
6	Stationary Tubesheet	25	Packing Follower Ring
7	Tubes	26	Lantern Ring
8	Shell	27	Tie Rods and Spacers
9	Shell Cover	28	Transverse Baffles or Support Plates
10	Shell Flange—Stationary Head End	29	Impingement Baffle or Plate
11	Shell Flange—Rear Head End	30	Longitudinal Baffle
12	Shell Nozzle	31	Pass Partition
13	Shell Cover Flange	32	Vent Connection
14	Expansion Joint	33	Drain Connection
15	Floating Tubesheet	34	Instrument Connection
16	Floating Head Cover	35	Support Saddle
17	Floating Head Flange	36	Lifting Lug
18	Floating Head Backing Device	37	Support Bracket
19	Split Shear Ring		

II.LITERATURE REVIEW

The subject of shell and tube heat exchanger (STHE) has a wide variety of process and phenomena. A vast amount of the material is published regarding STHE which depicts various factors affecting the thermal efficiency of the STHE. On the basis of that a brief summary is reviewed as follows:

Su Thet Mon Than, Khin Aung Lin, Mi Sandar Mon etal.[6]In this paper data is evaluated for heattransfer area and pressure drop and checking whetherthe assumed design satisfies all requirement or not.The primary aim of this design is to obtain a highheat transfer rate without exceeding the allowablepressure drop.

The decreasing pattern of curves of ReynoldsNumber and heat transfer coefficient states that the Re and h are gradually decreases corresponding as high as tube effective length. Gradual decrease in Reynolds Number means there is significant decrease in pressure drop respectively.

Rajiv Mukherjee etal.[2]explains the basics of exchanger thermal design, covering such topics as: STHE components; classification of STHEs according to construction and according to service; data needed for thermal design; tube side design; shell side design, including tube layout, baffling, and shell side pressure drop; and mean temperature difference. The basic equations for tube side and shellside heat transfer and pressure drop. Correlations for optimal condition are also focused and explained with some tabulated data. This paper gives overall idea to design optimal shell and tube heat exchanger. The optimized thermal design can be done by sophisticated computer software however a good understanding of the underlying principles of exchanger designs needed to use this software effectively.

Yusuf Ali Kara, Ozbilen Guraras et al.[3]]Prepared a computer based design model for preliminary design of shell and tube heat exchangers with single phase fluid flow both on shell and tube side. The program determines the overall dimensions of the shell, the tube bundle, and optimum heat transfer surface area required to meet the specified heat transfer duty by calculating minimum or allowable shell side pressure drop. He concluded that circulating cold fluid in shell-side has some advantages on hot fluid as shell stream since the former causes lower shell-side pressure drop and requires smaller heat transfer area than the latter and thus it is better to put the stream with lower mass flow rate on the shell side because of the baffled space.

M.Serna and A. Jimenez et al.[4]They have presented a compact formulation to relate the shell-side pressure drop with the exchanger area and the film coefficient based on the full Bell–Delaware method. In addition to the derivation of the shell side compact expression, they have developed a compact pressure drop equation for the tube-side stream, which accounts for both straight pressure drops and return losses. They have shown how the compact formulations can be used within an efficient design algorithm. They have found a satisfactory performance of the proposed algorithms over the entire geometry range of single phase, shell and tube heat exchangers.

Andre L.H. Costa, Eduardo M. Queirozetal.[5]Studied that techniques were employed according to distinct problem formulations in relation to: (i) heat transfer area or total annualized costs, (ii) constraints: heat transfer and fluid flow equations, pressure drop and velocity bound; and (iii) decision variable: selection of different search variables and its characterization as integer or continuous. This paper approaches the optimization of the design of shell and tube heat exchangers. The formulation of the problem seeks the minimization of the thermal surfaces of the equipment, for certain minimum excess area and maximum pressure drops, considering discrete decision variables.

Important additional constraints, usually ignored in previous optimization schemes, are included in order to approximate the solution to the design practice. describes to consider suitable baffle spacing in the design process, a computer program has been developed which enables designers to determine the optimum baffle spacing for segmentally baffled shell and tube condensers. Throughout the current research, a wide range of design input data specification for E and J types shell and tube condensers have been considered and their corresponding optimum designs for different values of W1 have been evaluated. This evaluation has been led to some correlation for determining the optimum baffle spacing.

M. M. El-Fawal, A. A. Fahmy and B. M. Taher:[7] In this paper a computer program for economical design of shell and tube heat exchanger using specified pressure drop is established to minimize the cost of the equipment. The design procedure depends on using the acceptable pressure drops in order to minimize the thermal surface area for a certain service, involving discrete decision variables. Also the proposed method takes into account several geometric and operational constraints typically recommended by design codes, and provides global optimum solutions as opposed to local optimum solutions that are typically obtained with many other optimization methods.

III. DESIGN PROCEDURE - SHELL AND TUBE HEAT EXCHANGERS,[1][16]

In order to develop relationships betweenthe heat transfer rate Q, surface area *A*, fluidterminal temperatures, and flow rates in a heat exchanger, the basic equations used for analysis are the energy conservation and heat transfer rate equations. The energy conservation equation for an exchanger having an arbitrary flow arrangement is,

Heat Transfer Rate: $Q=U*A*\Delta T_m$

$= mh*Cp_h*(Th_1 - Th_2)$

$= m_c*Cp_c*(Tc_2 - Tc_1)$

Here, $\dot{m}_h$ and $\dot{m}_c$ are mass flow rate of hot fluid and mass flow rate of cold fluid respectively. From the above equation we can find the outlet temperature of cold water.

To use this equation, it is necessary to determine the heat transfer coefficient and the temperature difference. For a shell and tube heat exchanger the required average temperature difference is the log mean temperature difference (LMTD) calculated by,

$$\Delta T_m = (T_{h1}-T_{c2})-(T_{h2}-T_{c1}) / \ln((T_{h1}-T_{c2})/(T_{h2}-T_{c1})$$

Heat Transfer Area is calculated by,

$$Q_{act} = U * A * \Delta T_m$$

Overall Heat Transfer Co-efficient [U] for the water to oil fluid is between 110 to 340 w/m2 °C

Correcting the LMTD:[10]

The maximum driving force for heat transfer is always the log mean temperature difference(LMTD) when two fluid streams are in countercurrent flow.The true mean temperaturedifference of such flow arrangements will differ from the logarithmic mean temperature difference by a certain factor dependent on the flow pattern and the terminal temperatures. This factor is usually designated as the log mean temperature difference correction factor, F. the factor F may be defined as the ratio of the true mean temperature difference (MTD) to the logarithmic mean temperature difference. The heat transfer rate equation incorporating F is given by,

$$\mathbf{Q_{act}=U*A*F_dT_{LM}}$$

The correction factor charts are availablefrom many sourcesthese parameters are cross-referenced on the appropriate chart to find the F factor. F factor curves drop off rapidly below 0.8. Consequently, if the design is indicating an F less than 0.8, we probably need to redesign (add tube passes, increase temperature differences, etc.) to get a better approximation of counter-current flow and thus higher F=0.9 values.

Tube and shell Diameters:

The most common sizes used are Ø3/4" and Ø1". Use the smallest diameter for greater heat transfer area with a minimum of Ø3/4" tube due to cleaning considerations and vibration. For shorter tube lengths say < 4ft can be used Ø1/2" tubes.Heat transfer area for the tube of the diameter d_o and length L can be given by following equation from which numbers of tube can be estimated,

$$A = \pi*d_o*Nt*L$$

Shell diameter can be obtained as,

$$Ds = 0.637 *\sqrt{CL/CTP} *[(A*(PR)2*do)/L]^{(1/2)}$$

CL=0.87 for 30° and 60°

CTP=0.9 for two tube pass

PR is usually between 1.25 to 1.5

Tube Arrangement

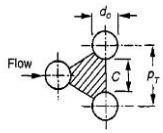
Triangular patternprovides a more robust tubesheet construction.Square pattern simplifies cleaning and has a lowershell side pressure drop.

Tube pitch is defined as:

$P_T=d_o+C$

P_T= tube pitch

d_o= tube outside diameter



C= clearance

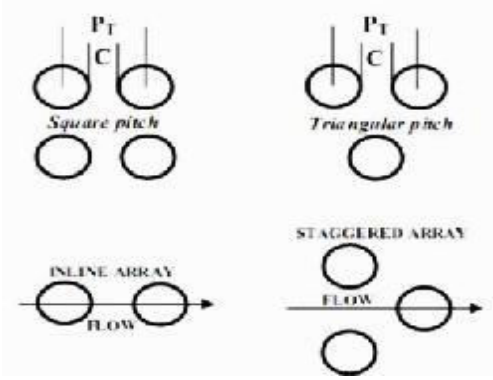


Fig 4: pitch layout[10]

Pitch:
Pitch can be determined as ,

$$P_T=P_R*d_o$$

Baffle Spacing:
The practical range of single segmental baffle spacing is 40% - 50% of shell diameter,

$$B_s =(0.4 \text{ to } 0.5) * D_s$$

Number of Baffle:
Numbers of baffles for the tube length L and baffle spacing Bs can be given by ,

$$N_b = (L / B_s) - 1$$

Baffle Cut Geometry: Baffle Cut:

The baffle cut vary from 20-35% of shell diameter with the most common being 20-25% as it affords the highest heat transfer for a given pressure drop.

$$B_c = (0.20 \text{ to } 0.35)* D_s$$

Bundle To Shell Clearance:
Bundle to Shell Clearance is calculated based upon the following equation,

$$L_{bb}= [12+ (0.005*Ds)]/1000 \text{ (m)}$$

Bundle diameter D_b can be estimated using constants shown:

$$D_b=d_o(N_t/K_1)^{1/n}$$

Where:
d_o= Tube Outside Diameter.
N_t= Number of tubes
K_1&n taken from table

Outer Tube Limit Diameter:
It is difference between the shell diameter and Bundle to Shell Clearance,

$$D_{otl}=D_s- L_{bb}$$

Centerline Tube Limit Diameter: [17]
It is difference between the Outer Tube Limit Diameter and tube diameter,

$$D_{ctl}= D_{otl}-d_o$$

Diametral clearance
The diametral clearance between shell diameter and baffle diameter is,

$$L_{sb}= 3.1+(0.004* Ds))/1000 \text{ (m)}$$

Baffle Diameter:
It is difference between the shell diameter and diametric clearance,

$$D_b= D_s- L_{sb}$$

TUBE SIDE HEAT TRANSFER COEFFICIENT AND PRESSURE DROP[1]

The fluid flows through tube is water so properties are
ρ = 985 kg/m3
μ_t=0.000485 at average tube side temperature,
Tube side flow area
Tube side flow area is determined as

$$A_{tp}= [\pi/4*d_{i2}]*[N_t/2]$$

Here di is tube inner diameter and Nt is numbers of passes.
Average Velocity In Tube:
Average fluid velocity in the tube is given by

$$V_m= m_t/ (\rho* A_{tp})$$

Here mt is mass flow rate of tube side fluid; ρ is density of fluid at tube side fluid temperature.
Tube Side Reynolds Number:
Tube Side Reynolds Number given by,

$$R_{et}= \rho*V_m* d_i/ \mu_t$$

Friction factor for the tube is,

$$f_t=(1.58\ln Ret -3.28)^{-2}$$

It is require to find the properties like Prandtl number and thermal conductivity of fluid at tube side fluid temperature to find tube side heat transfer coefficient, Nusselt number for the tube side fluid is;

$$Nub=[(ft/2)*(Ret1000)*Pr]/[(1+12.7(ft/2)0.5*(pr2/3-1)]$$

Tube side heat transfer coefficient is estimated as;

$$h_i= N_{ub}*k/di$$

Tube Side Pressure Drop:[10]
The tube side pressure drop can be calculated byknowing the number of tube passes (Np) and length(L) of heat exchanger; the pressure drop for the tubeside fluid is given by equation,

$$\Delta P_t = 4f\frac{LN_p}{d_i}\frac{G_i^2}{2\rho}$$

The change of direction in the passes introduction inthe passes introduction an additional pressure dropdueto sudden expansions and contractions that thetube fluid experiences during a return that isaccounted for allowing four velocity head per passThe total pressure drop of the side becomes:

$$\Delta P_t = \left(4f\frac{LN_p}{d_i}+4N_p\right)\frac{\rho\mu_m^2}{2}$$

SHELL SIDE HEAT TRANSFER COEFFICIENTS AND PRESSURE DROP[1]

Equivalent diameter(De)
Triangular pitch layout, for which the equivalent diameter is given by following equation.

$$De= 4*\{[((P_T^2\sqrt{3})/4)-(\pi*d_o^2/8)] / (\pi*d_o/2)\}$$

Here do is tube outer diameter.

Clearance between Tubes
Clearance between two tubes is estimated as,

$$C= P_T- d_o$$

Bundle Cross Flow Area
Bundle Cross Flow Area can determined by following equation,

$$A_s= D_s*C*B_s/ PT$$

Shell Side Mass Velocity
The variable that affects the mass velocity are shell diameter Ds, clearances between the adjacent tubes and the cross flow area As. shell mass velocity is found with,

$$U_s=m_s/ A$$

Here m_s is mass flow rate of fluid in shell side.

Shell Side Reynolds Number
Reynolds number for the shell side flow can be estimated as,

$$R_{es}= U_s* D_e /\mu s$$

Here Us is kinematic viscosity of fluid at given fluid temperature.
Approximate Wall Temperature:
Approximate Wall Temperature is determined as,
$T_w= (T_{h1} + T_{h2} +T_{c2} +T_{c1})/4$

Shell side heat transfer coefficient
To find the shell side heat transfer coefficient it requires to find the properties like viscosity, thermal conductivity of working fluid at approximate wall temperature and viscosity at shell side fluid temperature.

$$ho=[k/De*0.36]*[De*Us/\mu s]^{0.55}*[Cp*\mu s/k]^{0.33}*[\mu s/ \mu w]^{0.14}$$

Above equation gives the shell side heat transfer coefficient.

Shell Side Pressure Drop:[10]
The calculation of shell side pressure drop issignificantly more complicatedas the shell sideflow path is considerably more complex. For our purposes, we will use a correlation presented bywhich can be taken to an appropriate chart and usedto get a friction factor. Note that the chart provides a dimensionalfriction factor (unlike the dimensionless values used for pipe flow). The friction factor has to be transformed to a pressure drop, a count of how many times the fluid crosses the tube bundle is needed. It crossesbetween the baffles, so the cross will be one more than the number of baffles, NB. The number ofbaffles can be determined using the baffle spacing:

$$N_b+1=L/P_B$$

The pressure drop is then determined using theEquivalent diameter, cross flow velocity, frictionfactor, number of crosses, and fluid properties:

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

Where,
$\emptyset s = (\mu b+ \mu s)\ 0.14$
N_b = Number of baffles
$(N_b + 1)$ = Number of times fluid passes to the tube bundle
Friction factor (f) calculated from:

$$f = e^{(0.576-0.19\ln(Res)}$$

Overall Heat Transfer Coefficient
Overall heat transfer coefficient is determined by following equation,

$$U = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o ln(\frac{d_o}{d_i})}{2k} + \frac{1}{h_o}}$$

IV. RESULTS AND DISCUSSION

Example 1: Following are the operating parameters while designing the shell & tube heat exchanger:
1. Inlet temperature of hot water, T_{h1}= 65 °C
2. Outlet temperature of hot water, T_{h2}= 58 °C
3. Inlet temperature of cold water, T_{c1}= 30 °C
4. Mass flow rate of cold water, m_c= 0.042 kg/s
5. Mass flow rate of hot water, m_h= 0.024 kg/s

Based on the above methodology the calculated results for fabrication of STHE are summarized below;

Table 1: Calculated Results

Design Parameter	Value
Cold Outlet Temperature(T_{c2})	41 °C
Heat Duty(Q)	1912 W
LMTD (ΔT_m)	25.94 °C
Heat Transfer Area By LMTD (A)	0.40935 m^2
Numbers Of Tubes (N_t)	26 nos
Shell Diameter (D_s)	0.1622 m
Pitch (P_T)	0.027 m
Baffle Spacing (B_s)	0.0811 mm
Numbers Of Baffles (N_b)	4 nos
Baffle Cut (B_c)	0.048 m

Experimental Setup

Above results which are helps to fabricate shell and tube heat exchanger for maximum cooling of oil and According to above calculated values we fabricated the shell & Tube Heat Exchanger as below:

Fig 5: STHE ExperimentalSetup

Example 2: This example presented here was the one reported by Polley, Panjeh Shahi and Nunez to demonstrate the inverse design methodology. The original example involved water on the tube side of the exchanger and the fluid on shell side is viscous oil. The tube side and shell side pressure drops for this situation are 11.66 kPa and 13.7 kPa. These are the allowable ΔP subsequently used in M M El Fawal[7] and design done with his model. From both authors get the results as shown in table 4. The sample operating conditions and the design data are shown in Tables 2 and 3. The flow rates, temperatures, allowable pressure drops, and physical properties of streams are fixed. It is required to determine the optimum area and optimum cost of shell-and-tube heat exchanger.

Table 3: Shell and tube exchanger design data– physical properties

Physical Properties	Shell side	Tube side
Fluid	oil	Water
Flow rate (kg/s)	22.4	77.96
Fluid density (kg/m3)	740	1000
Heat capacity (J/kg. K)	2407	4187
Viscosity (Pa. s)	0.494	1
Thermal conductivity (W/m. K)	0.105	0.61
Inlet temp. (deg K)	373	280
Outlet temp (deg K)	317	290
Allowable Δ P (kPa)	14	12

Table 4: Shell and tube exchanger design problem – geometry

Geometry Values	Values	unit
Tube OD	16	mm
Tube ID	14	mm
Tube layout	30	deg
Tube pitch	21	mm
Baffle-to-shell clearance	6	mm
Tube-to-baffle clearance	0.6	mm
Bundle-to-shell clearance	10	mm

Table 5: Comparison of shell and tube heat exchanger designs

Geometry	Polley et al.	M M El Fawal et al.	Present work
Shell diameter (mm)	563	520	531.371
Tube length (mm)	1815	1728	1550
Baffle cut (%)	29.3	26.8	25
Baffle spacing (mm)	253	228	265.685
No. of baffles	6	5	4.834
No. of tubes	574	548	520.000
No. of tube passes	2	2	2
Required area (m2)	52.3	49.39	40.481
Installed area (m2)	52.3	49.39	40.481
Shell side Re	21398	27926	19572.895
Shell side Δ P (kPa)	13.7	13.493	13.590
Tube side Δ P (kPa)	11.69	11.61	11.339
Shell side heat coefficient (W/m2K)	1406	1471	1371.642
Tube side heat coefficient (W/m2K)	6641	6750	4809.549
Pumping cost ($/year)	950	2424	1206.314
Area cost ($/year)	3150	2826	1979.630

From above tables the graphical representation of Comparison between Polley et al, M M El Fawal et al. and our present design of shell and tube heat exchanger as below:

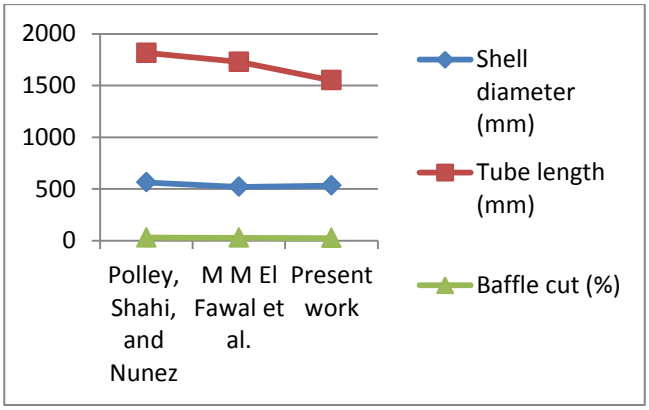


Fig 6: Comparison of Shell diameter, Tube length, baffle cut with Polley et al, M M El Fawal et al. and present work.

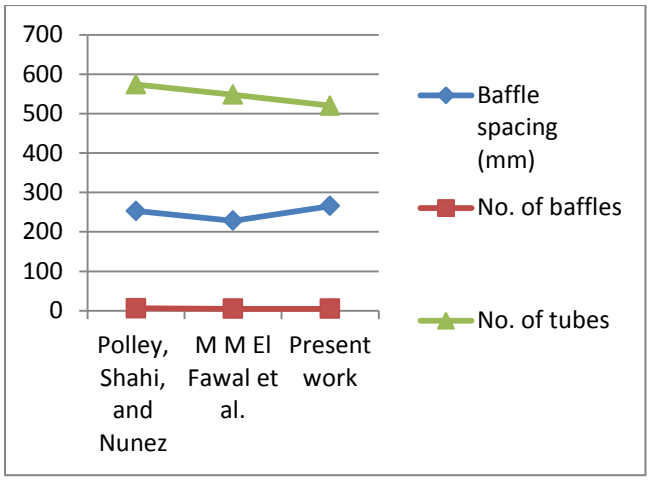


Fig 7: Comparison of Baffle spacing, No. of baffles, No. of tubes with Polley et al, M M El Fawal et al. and present work.

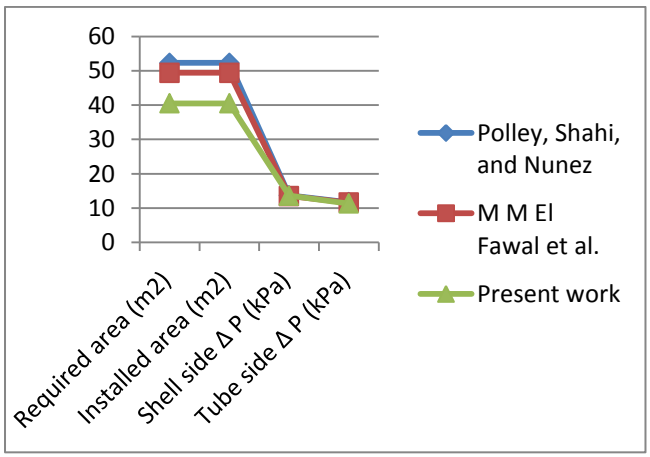


Fig 8: Comparison of Heat transfer areas, Shell side pressure drop, Tube side pressure drop with Polley et al, M M El Fawal et al. and present work.

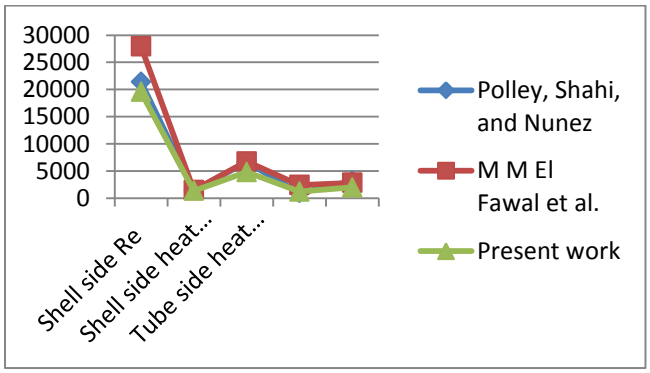


Fig 9: Comparison of Shell side Reynolds no, Shell side heat transfer coefficient, Tube side heat transfer coefficient, Pumping cost, Area cost with Polley et al, M M El Fawal et al. and present work.

V. CONCLUSIONS

1. The STHE design was carried out using TEMA/ASME standards mathematically. It is found that design of STHE obtained by mathematical approach is very easy, simple, advance & less time consuming as comparing to existing models and method used in different Indian industries. It is a good solution to bridge the gates between institution and industries. Through this project a "shell and Tubeheat exchanger" is developed under standard conditions and which is helped to know how to achieve low cost semi automation application.

2. In this work, an optimization model for the design of a shell and tube heat exchanger has been proposed. The optimization strategy based upon the presented analytical optimization analysis is developed on a MS Excel. Important additional constraints, usually ignored in previous optimization schemes, are included in order to approximate the solution to the design practice. In case example 2, the obtained results in the present work are consistent and with the corresponding values reported by Polley et al. and M M El Fawal et al[7]. and it is more efficient in terms of providing excellent optimum solutions than standard optimization method reported by Polley et al. and M M El Fawal et al[7].

3. Also the result of the study ends up with the final conclusion that the use of the model provides the best solutions with higher quality gives more flexibility in geometry ,good performance at low cost together with short duration of real time.

FUTURE SCOPE

Present we are only design the fabricated of shell and tube heat exchanger, in future we can perform the experiment on this rig by varying the temperatures and mass flow rates of hydraulic oil as well as water and check the performance of this shell and tube heat exchanger for various industrial applications.

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