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Thermal designing of shell & tube heat exchanger: a review

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ABSTRACT
Thermal designing of a shell and tube heat exchanger typically includes the determination of heat transfer area ,number of tubes, tube length and tube diameter ,tube layout, number of shell and tube passes ,tube pitch, number of baffles, its type and size, shell and tube side pressure drop. This paper reveals the effects various parameters in the thermal designing of shell and tube heat exchanger.

Introduction

A shell and tube heat exchanger is a type of heat exchanger consists of a shell(a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes and another fluid flows over the tubes (through the shell)to transfer heat between the two fluids. The sets of tubes is called a tube bundle. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for high pressure applications.

Ahmad Fakheri[11] in his paper shows that how to calculate the efficiency of the heat exchangers based on the second law of thermodynamics. He says that corresponding to every heat exchanger there is an ideal balanced counter flow heat exchanger which has the properties of same UA, same AMTD and minimum entropy generation corresponding to minimum losses and irreversibility. The efficiency of the heat exchanger may be calculated by comparing the heat transfer capability of actual heat exchanger with that of the ideal heat exchanger.

Rajeev Mukherjee[12] 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 shell side 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.

Jiangfeng Guo et.al[13] took some geometrical parameters of the shell-and-tube heat exchanger as the design variables and the genetic algorithm is applied to solve the associated optimization problem. It is shown that for the case that the heat duty is given, not only can the optimization design increase the heat exchanger effectiveness significantly, but also decrease the pumping power dramatically.

A. Pignotti[14] in his paper established relationship between the effectiveness of two heat exchanger configurations which differ from each other in the inversion of either one of two fluids.

M. S. Bohn[15] in his article presents a method of calculating the electric power generated by a thermoelectric heat exchanger. The method presented in this paper is an extension of the NTU method used to calculate heat-exchanger's heat-transfer effectiveness. The effectiveness of thermoelectric power generation is expressed as the ratio of the actual power generated to the power that would be generated if the entire heat-exchanger area were operating at the inlet fluid temperatures.

V.K. Patel and R.V. Rao[16] explore the use of a non-traditional optimization technique; called particle swarm optimization (PSO), for design optimization of shell-and-tube heat exchangers from economic view point. Minimization of total annual cost is considered as an objective function.

Three design variables such as shell internal diameter, outer tube diameter and baffle spacing are considered for optimization. Two tube layouts viz. triangle and square are also considered for optimization.

Hetal Kotwal and D.S Patel[17] focus on the various researches on Computational Fluid Dynamics (CFD) analysis in the field of heat exchanger. Different turbulence models available in CFD tools i.e. Standard k-ε model, k-ε RNG model, Realizable k-ε, k-ω and RSM model in conjunction with velocity pressure coupling scheme and have been adopted to carry out the simulation. The steady increase in computing power has enable model to react for multi- phase flows in realistic geometry with good resolution. The quality of the solution has proved that CFD is effective to predict the behaviour and performance of heat exchanger.

Shiv Kumar Rathore and Ajeet Bergaley[18] worked with the aim to identify the advantages of low-finned tube Heat Exchangers over Plain tube (Bare Tube) units. To use finned

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tubes to advantage in this application, several technical issues were to be addressed; (1) Shell side and tube side Pressure, (2) Cost, (3) Weight and (4) Size of Heat Exchanger, Enhanced tubular heat exchangers results in a much more compact design than conventional plain tube units, obtaining not only thermal, mechanical and economical advantages for the heat exchanger, but also for the associated support structure, piping and skid package unit, and also notably reduce cost for shipping and installation of all these components. A more realistic comparison is made on the basis of respective cost per meter of tubing divided by the overall heat transfer coefficient for the optimized units, which gives a cost to performance ratio. This approach includes the entire thermal effect of internal and external heat transfer augmentation and fouling factors in the evaluation. This is typically quite close to reality and easy for the thermal designer to evaluate himself. The results of this analysis shows that the finned tube heat exchanger is more economical than Conventional Bare tube Exchanger, The tube side pressure drop and fluid velocity is higher than the conventional bare tube exchanger, which prevent fouling inside the tubes, The shell side pressure drop is some lesser but fluid velocity is higher than the conventional heat exchanger which saves the outer surface of tubes from fouling creation and fluid transfer time. The shell diameter of finned tube Exchanger is lesser than Conventional bare tube heat exchanger, which saves sheet material and reduces the size of the shell, which helps to easily installation in the plant. Hari Haran et.al[19] proposed a simplified model for the study of thermal analysis of shell and tube type heat exchangers of water and oil type is proposed. The robustness and medium weighted shape of Shell and Tube heat exchangers make them well suited for high pressure operations. This paper shows how to do the thermal analysis by using theoretical formulae and for this they have chosen a practical problem of counter flow shell and tube heat exchanger of water and oil type, by using the data that come from theoretical formulae they designed a model of shell and tube heat exchanger using Pro-E and done the thermal analysis by using ANSYS software and comparing the result that obtained from ANSYS software and theoretical formulae. For simplification of theoretical calculations they have also done a C code which is useful for calculating the thermal analysis of a counter flow of water-oil type shell and tube heat exchanger. The result after comparing both was that they were getting an error of 0.0274 in effectiveness.

Major parts of a Shell and tube exchanger

The shell and tube exchanger consists of four major parts:

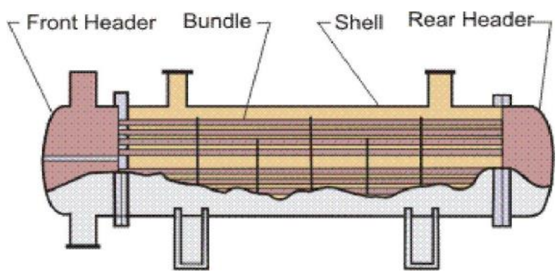


Fig.1. Shell and tube exchanger

Front Header—this is where the fluid enters the tube side of the exchanger. It is sometimes referred to as the Stationary Header.2. Rear Header—this is where the tube side fluid leaves the exchanger or where it is returned to the front header in exchangers with multiple tube side passes.

Tube bundle—this comprises of the tubes, tube sheets, baffles and tie rods etc. to hold the bundle together.

Shell—this contains the tube bundle

Selection of a Shell for a heat exchanger

Shell is a container for the shell fluid and the tube bundle is placed inside the shell. Sell diameter should be selected in such a way to give a close fit to the tube bundle. The clearance between the tube bundle and inner shell wall depends on the type of heat exchanger. Shells are usually fabricated from standard steel pipe with satisfactory corrosion allowance.

Selection of Tube for a Heat exchanger

Tube OD of ¾ and 1" are very common to design a compact heat exchanger. The most efficient condition for heat transfer is to have the maximum number of tubes in the shell to increase turbulence.. The tube thickness is expressed in terms of BWG (Birmingham Wire Gauge) and true outside diameter (OD). The tube length of 6, 8, 12, 16, 20 and 24 ft are preferably used. Longer tube reduces shell diameter at the expense of higher shell pressure drop. Finned tubes are also used when fluid with low heat transfer coefficient flows in the shell side. Stainless steel, admiralty brass, copper, bronze and alloys of copper-nickel are the commonly used tube materials.

Tube-layout and Tube-pitch

Tube layout is characterized by the included angle between tubes standard types of tube layouts are the square and the equilateral triangle. Triangular pitch (30^o layout) is better for heat transfer and surface area per unit length (greatest tube density.) Square pitch (45^0 & 90^o layouts) is needed for mechanical cleaning30°,45° and 60° are staggered, and 90° is in line. For the identical tube pitch and flow rates, the tube layouts in decreasing order of shell-side heat transfer coefficient and pressure drop are:30°,45°,60°, 90°.

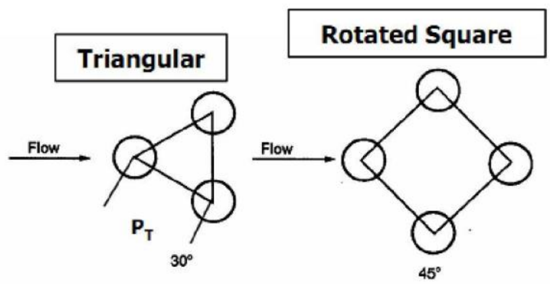


Fig 2.Tube layout: Triangular and Rotated square

The 90° layout will have the lowest heat transfer coefficient and the lowest pressure drop.. The square pitch (90° or 45°) is used when jet or mechanical cleaning is necessary on the shell side. In that case, a minimum cleaning lane of ¼ in. (6.35 mm) is provided

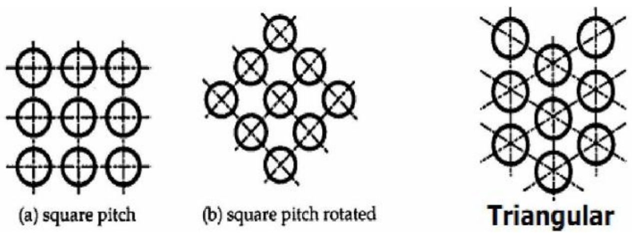


Fig.3.Tube Pitch

The square pitch is generally not used in the fixed header sheet design because cleaning is not feasible. The triangular pitch provides a more compact arrangement, usually resulting in smaller shell, and the strongest header sheet for a specified shell-side flow area. It is preferred when the operating pressure difference between the two fluids is large.

Tube pitch

Tube pitch is the shortest centre to centre distance between the adjacent tubes.

Selection of Tube-pitch

The selection of tube pitch is a compromise between a close pitch (small values of Pt/do) for increased shell-side heat transfer and surface compactness, and an open pitch (large values of Pt/ do) for decreased shell-side plugging and ease in shell-side cleaning. Tube pitch Pt is chosen so that the pitch ratio is 1.25 < Pt/do < 1.5.When the tubes are to close to each other (Pt/do less than 1.25) , the header plate (tube sheet) becomes to weak for proper rolling of the tubes and cause leaky joints. Tube layout and tube locations are standardized for industrial heat exchangers .However, these are general rules of thumb and can be "violated" for custom heat exchanger designs.

Tube-count

The number of tubes that can be accommodated in a given shell ID is called tube count. The number of tubes in an exchanger depends on the fluid flow rates, available pressure drop. The number of tubes is selected such that the tube side velocity for water and similar liquids ranges from 0.9 to 2.4 m/s (3 to 8 ft/sec) shell-side velocity from 0.6 to 1.5 m/s (2 to 5 ft/sec). The lower velocity limit corresponds to limiting the fouling, and the upper velocity limit corresponds to limiting the rate of erosion. When sand and silt are present, the velocity is kept high enough to prevent settling.

Selection of Tube material

Requirement for low cost, light weight, high conductivity, and good joining characteristics often leads to the selection of aluminum for the heat transfer surface. Stainless steel is used for food processing or fluids that require corrosion resistance.

In general, one of the selection criteria for exchanger material depends on the corrosiveness of the working fluid.

Selection of Tube diameter

From the heat transfer viewpoint, smaller-diameter tubes yield higher heat transfer coefficients and result in a more compact exchanger. However, larger-diameter tubes are easier to clean and more rugged. The most common plain tube sizes have 15.88,19.05, and 25.40 mm (5/8, ¾, 1 inches) tube outside diameters. The foregoing common sizes represent a compromise. For mechanical cleaning, the smallest practical size is 19.05 mm. For chemical cleaning, smaller sizes can be used provided that the tubes never plug completely.

Selection of Tube length

Tube length affects the cost and operation of heat exchangers. Longer the tube length (for any given surface area) fewer tubes are needed, requiring less complicated header plate with fewer holes drilled • Shell diameter decreases resulting in lower cost. Typically tubes are employed in 8, 12, 15, and 20 foot lengths. . Mechanical cleaning is limited to tubes 20 ft and shorter, although standard exchangers can be built with tubes up to 40 ft. Shell-diameter-to-tube-length ratio should be within limits of 1/5 to 1/15 5. Maximum tube length is dictated by architectural layouts transportation (to about 30m.)

Tube sheet, its attachment and selection

The tubes are fixed with tube sheet that form the barrier between the tube and shell fluids. The tubes can be fixed with the tube sheet using ferrule and a soft metal packing ring. The tubes are attached to tube sheet with two or more grooves in the tube sheet wall by "tube rolling". The tube metal is forced to move into the grooves forming an excellent tight seal. This is the most common type of fixing arrangement in large industrial exchangers. The tube sheet thickness should be greater than the tube outside diameter to make a good seal. The recommended standards (IS:4503 or TEMA) should be followed to select the minimum tube sheet thickness.

Considerations for Fluid Allocation to tube side

Tube side is preferred under these circumstances: For Fluids which are prone to foul higher velocities will reduce buildup. Mechanical cleaning is also much more practical for tubes than for shells. Corrosive fluids are usually best in tubes as tubes are cheaper to fabricate from exotic materials very high temperature fluids require alloy construction. Toxic fluids to increase containment Streams with low flow rates to obtain increased velocities and turbulence. High pressure streams since tubes are less expensive to build strong. Streams with a low allowable pressure drop

Considerations for Fluid Allocation to shell side

Viscous fluids go on the shell side, since this will usually improve the rate of heat transfer. On the other hand, placing them on the tube side will usually lead to lower pressure drops. Judgment is needed

Baffle

Baffles are used to increase the fluid velocity by diverting the flow across the tube bundle to obtain higher transfer co-efficient.

Need of Baffles

Baffles serve two functions:. Support the tubes for structural rigidity, preventing tube vibration and sagging. Divert the flow across the bundle to obtain a higher heat transfer coefficient.

Baffle spacing

The distance between adjacent baffles is called baffle-spacing. The baffle spacing of 0.2 to 1 times of the inside shell diameter is commonly used. Baffles are held in positioned by means of baffle spacers. Closer baffle spacing gives greater transfer co-efficient by inducing higher turbulence. The pressure drop is more with closer baffle spacing.

Various types of baffles used

Various types of baffles used are Cut-segmental baffle, Disc and doughnut baffle, Orifice baffle

Cut-segmental baffle

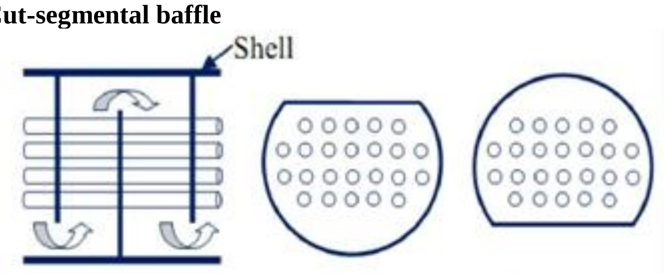


Fig 4. Cut Segmental Baffle

In case of cut-segmental baffle, a segment (called baffle cut) is removed to form the baffle expressed as a percentage of the baffle diameter. Baffle cuts from 15 to 45% are normally used. A baffle cut of 20 to 25% provide a good heat-transfer with the reasonable pressure drop. The % cut for segmental baffle refers to the cut away height from its diameter.

Disc and ring baffle

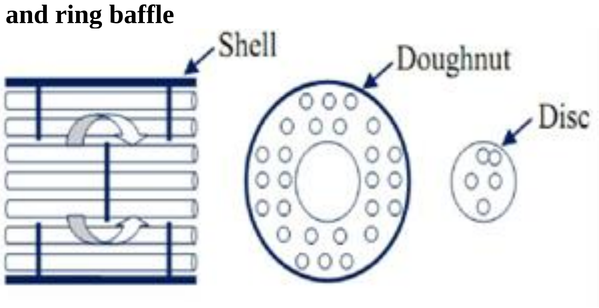


Fig.5.Disc and doughnut baffle

Disc and ring baffles are composed of alternating outer rings and inner discs, which direct the flow radially across the tube field. The potential bundle-to-shell bypass stream is eliminated . This baffle type is very effective in pressure drop to heat transfer conversion

Orifice baffle

In an orifice baffle shell-side-fluid flows through the clearance between tube outside diameter and baffle-hole diameter

Shell pass and tube pass

A pass is when liquid flows all the way across from one end to the other of the exchanger. We will count shell passes and tube passes. An exchanger with one shell pass and two tube passes is a 1-2 exchanger. Almost always, the tube passes will be in multiples of two (1-2, 1-4, 2-4, etc.). Odd numbers of tube passes have more complicated mechanical stresses, etc. An exception: 1-1 exchangers are sometimes used for vaporizers and condensers. A large number of tube passes are used to increase the tube side fluid velocity and heat transfer coefficient and minimize fouling. This can only be done when there is enough pumping power since the increased velocity and additional turns increases the pressure drop significantly. Higher velocities in the tube result in higher heat transfer coefficients, at the expense of increased pressure drop. Therefore, if a higher pressure drop is acceptable, it is desirable to have fewer but longer tubes (reduced flow area and increased flow length). Long tubes are accommodated in a short shell exchanger by multiple tube passes. The number of tube passes in a shell generally range from 1 to 10. The standard design has one, two, or four tube passes.

Common types of shell and tube exchangers

The common types of shell and tube exchangers are: Fixed tube-sheet exchanger (non-removable tube bundle HX). Removable tube bundle HX: a. floating-head and b. U-tube exchanger.

Fixed tube-sheet exchanger (non-removable tube bundle)

The simplest and cheapest type of shell and tube exchanger is with fixed tube sheet design. In this type of exchangers the tube sheet is welded to the shell and no relative movement between the shell and tube bundle is possible.

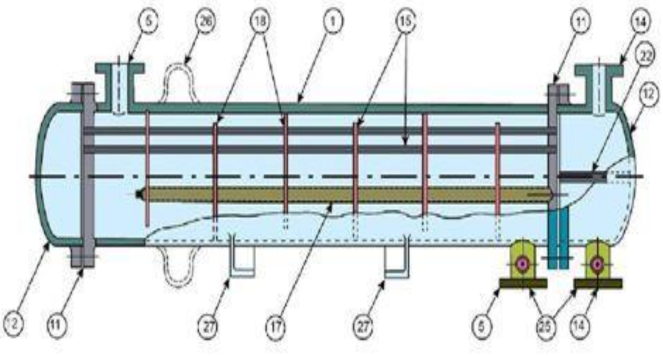


Fig.6 Fixed tube-sheet exchanger (non-removable tube bundle)

1. Shell
2. Shell cover
3. Shell flange (channel end)
4. Shell flange (cover end)
5. Shell nozzle or branch
6. Floating tube sheet
7. Floating head cover
8. Floating head flange
9. Floating head gland
10. Floating head backing ring
11. Stationary tube sheet
12. Channel or stationary head
13. Channel cover
14. Channel nozzle or branch
15. Tube (straight)
16. Tubes (U-type)
17. Tie rods and spacers
18. Transverse (or cross) baffles or support plates
19. Longitudinal baffles
20. Impingement baffles
21. Floating head support
22. Pass partition
23. Vent connection
24. Drain connection
25. Instrument connection
26. Expansion bellows
27. Support saddles
28. Lifting lugs
29. Weir
30. Liquid level connection

Removable tube bundle Heat Exchanger

Tube bundle may be removed for ease of cleaning and replacement. Removable tube bundle exchangers further can be categorized into 2 . a. floating-head and b. U-tube exchanger.

Floating-head exchanger

It consists of a stationery tube sheet which is clamped with the shell flange. At the opposite end of the bundle, the tubes may expand into a freely riding floating-head or floating tube sheet. A floating head cover is bolted to the tube sheet and the entire bundle can be removed for cleaning and inspection of the interior.

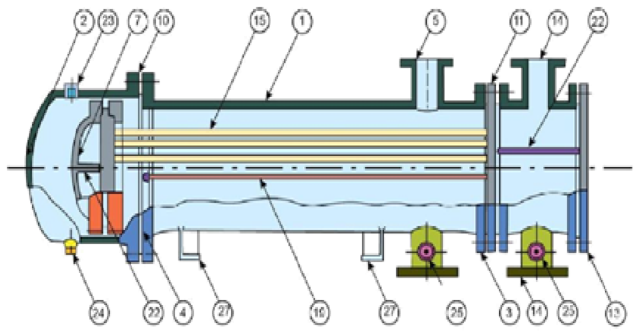


Fig.7 Floating-head exchanger

U-tube exchanger

This type of exchangers consists of tubes which are bent in the form of a "U" and rolled back into the tube sheet shown in the Figure below

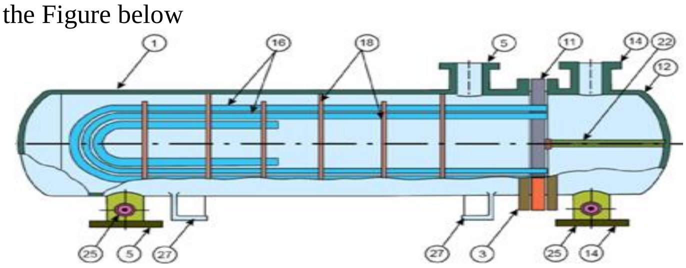


Fig 8. U-tube exchanger

This means that it will omit some tubes at the centre of the tube bundle depending on the tube arrangement. The tubes can expand freely towards the "U" bend end.

P.S.Gowthaman and S.Sathish [1] in their work a comparison is made by analyzing the segmental and helical baffle in a heat exchanger .They found that higher heat transfer and lower pressure drop is achieved in a helical baffle compared to segmental baffle. They had created a virtual model of helical and segmental baffle by using PRO-E and analyzing in a Fluent

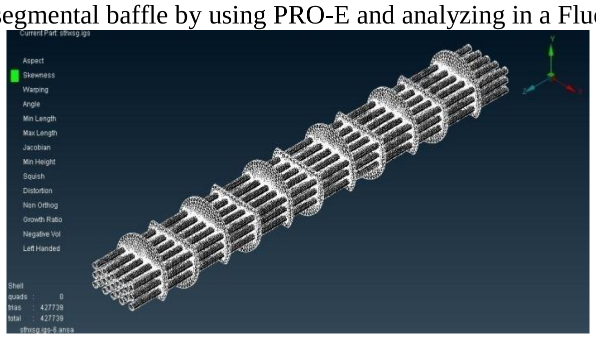


Fig.9 Surface mesh with Tube and Shell in Segmental

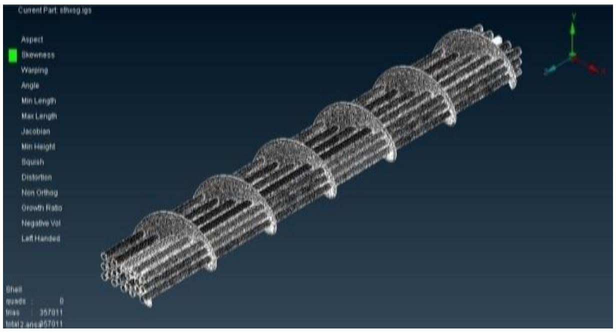


Fig.10 Surface mesh with Helical Baffle

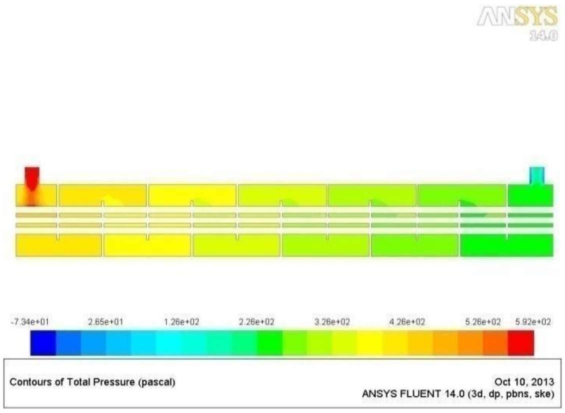


Fig.11 Total pressure in segmental midplane

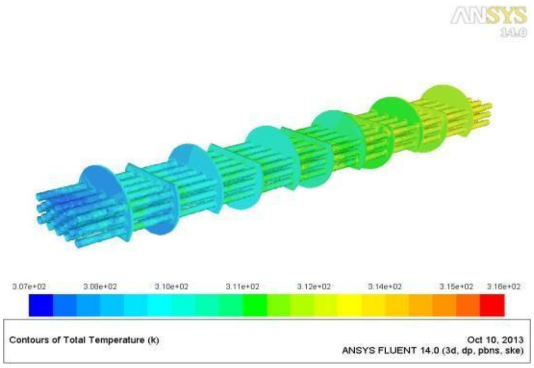
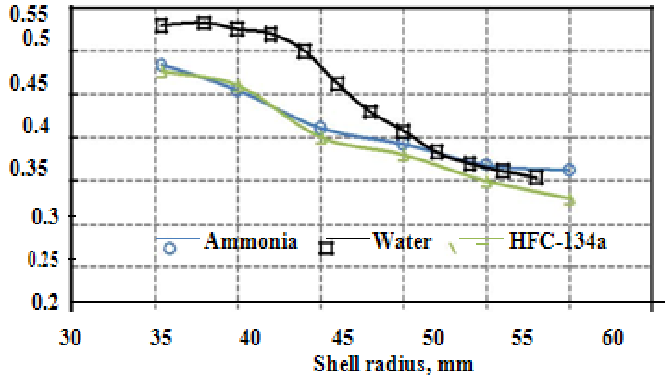
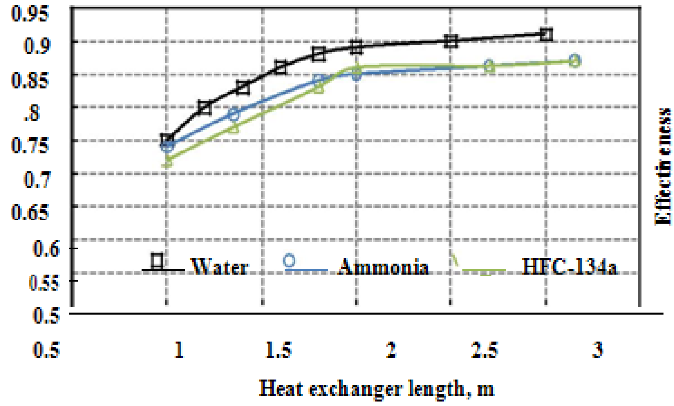


Fig.12 Total Temperature in Segmental Baffle

Amarjit Singh and Satbir S Seghval[2] had studied the different effects in Shell and Tube heat exchanger by increasing Reynolds no. with segmental baffles at 0^0,30^0,and 60^0.The model is studied with four segmental baffles. They found that heat transfer coefficient increases with increase in Reynolds no., Nusselt No. increases with increase in Reynolds no. The value of LMTD increases with increase in Reynolds Number.

S.N.Hossain and S.Bari [3] had conducted an experimentally connecting a shell and tube heat exchanger at an exit of diesel engine having specification engine 13B,Tayota made ,4cylinder Water cooled diesel engine,102 mm bore and 105 mm stroke, compression ratio17.6, Torque 217 Nm.@200rpm. The experiments had been conducted by usingHFC,134a,and ammonia as the working fluid. It is found that it can increase the overall efficiency of diesel engine. This technology will reduce the fuel consumption and there by will also reduce green house gases, and this emissions per Kw of power van be achieved with this heat exchanger by using HFC,134 a and ammonia respectively.





Andre L.H,Costa and Eduardo.m.Queiroz[4] described a method for the optimization of SHTE. The formation of the problem seeks the minimization of thermal surface of the equipment, for certain minimum excess area and maximum pressure drops ,considering discrete decision variables. Importnt additional constraints usually ignored in previous optimization schemes are included in order to approximate the solution to the design practice.

K.Anand,V.K.Pravin and P.H.Veena [5] had designed the SHTE based on Bell Delaware method

STEP 1: Calculate the shell side area at or near the centre line for one cross flow section S_m,

$S_m = L_b*[(D_s – D_{otl}) + \{(D_{otl} – D_o)*(P_t - D_o)\}/P_t]$

STEP 2: Calculate shell side mass velocity G_s and linear velocity U_s.

$G_s = m_s / S_m$

STEP 3: Calculate shell side Reynolds number R_{es}.

$R_{es} = (G_s* D_o) / \mu_s$

STEP 4: Calculate shell side Prandtl number P_{rs}.

$P_{rs} = (C_{ps}*\mu_s) / K_s$

STEP 5: Calculate the colburn j factor j_i.

$j_i = a_1*[\{1.33 / (P_t / D_o)\} \wedge a]* (R_{es}\wedge a_2)$

STEP 6: Calculate the value of the coefficient a.

$a = a_3 / [1+ \{0.14* (Res\wedge a4)\}]$

STEP 7: Calculate the ideal heat transfer coefficient h_i.

$h_i = j_i *C_{ps}*(m_s / S_m)*\{(1/P_{rs})\wedge(2/3)\}*\{ (\mu_s / \mu_w)\wedge 0.14\}$

STEP 8: Calculate shell to baffle leakage area for one baffle S_{sb}.

$S_{sb} = D_s*(\Delta sb / 2)*[\pi - \cos^{-1}(\Theta)]$

STEP 9

$S_{tb} = (\pi*D_o)*(\Delta tb / 2)*N_t*[(1 +F_c)/ 2]$

STEP 10: Calculate

$(S_{sb} + S_{tb}) / S_m$

STEP 11: Calculate the fraction of the crossflow area available for bypass flow F_{bp}.

$F_{bp} = (L_b / S_m)*(Ds - D_{otl})$

STEP 12: Calculate the shell side heat transfe r coefficient for the exchanger h_o.

$h_o = h_i *J_c*J_L*J_b*J_s *J_r$

Chandrkant B Kothare [5] had developed a sophisticated and user-friendly computer software using visual Basic 6.0(As a primary programming language)for the hydraulic design of SHTE based on the D.Q.Kern method. The software consists of five forms designed in visual basic at front end .User should fill the relevant information in to the forms which include fluid property, shell side property, tube side property, form containing different graphs and solution. Design results can be printed and saved. Data base required for calculation is as stored at back end in access

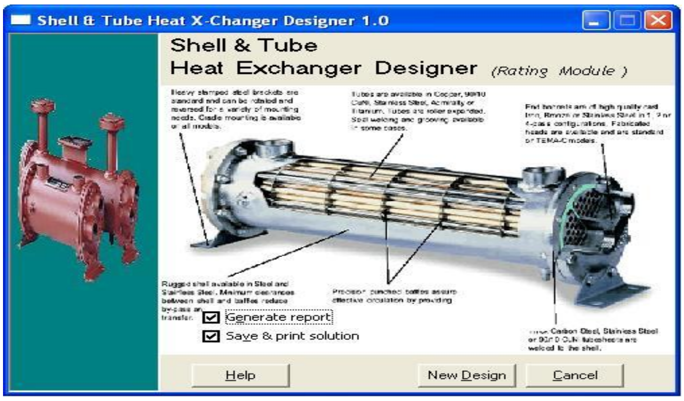


Fig.13 SHTE Designer

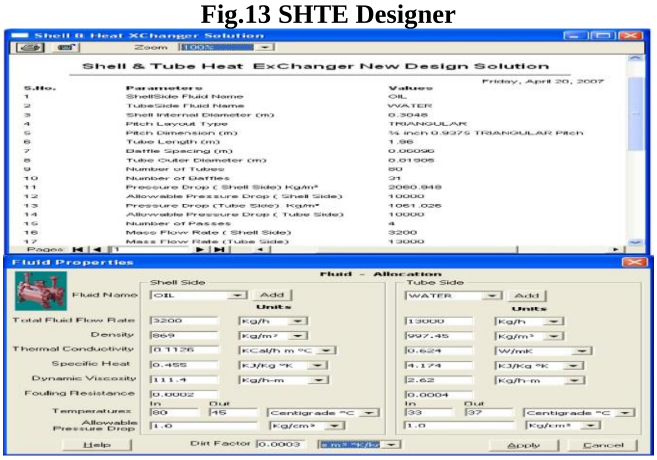


Fig.14 SHTE Designer solution

Vindhya Vasiny Prasad Dubey,Raj Rajat Varma and Piyush Shankar Varma[6]had designed a simplified model of shell and tube heat exchanger using kerns method to cool the water from 550 c to 450 c by using water at room temperature, and carried out steady state analysis on ANSYS -14,to justify the design. The heat exchanger had been tested under the various flow conditions using insulation of aluminum foil, cotton wool,tape,form,paper etc.It is found that effectiveness of the heat exchanger is better when hot fluid in to the tubes. After using insulation the heat transfer rate increases

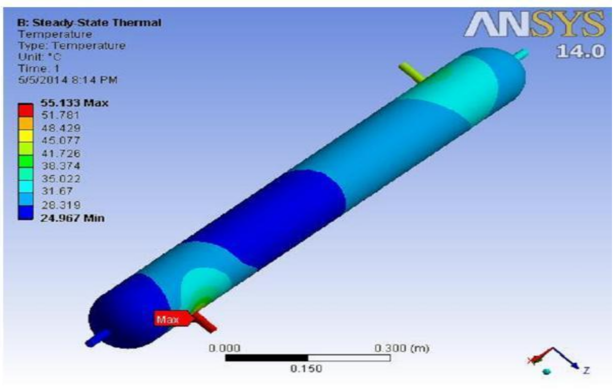


Fig.15 CFD Analysis

Lutcha and Nemcansky[7] upon investigation of the flow field patterns generated by various helix angles used in helical baffle geometry found that the flow patterns obtained in their study are similar to plug flow condition which is expected to decline pressure at shell side and increase heat transfer process significantly.

Stehlik[8] studied the effect of optimized segmental baffles and helical baffles in heat exchanger based on Bell-Delaware method and demonstrated the heat transfer and pressure decline correction factors for a heat exchanger.

Oil- Water Shell and Tube Heat Exchangers with various baffle geometries of 5 continuous helical baffles and one segmental baffle and test results were compared for performance with respect to their heat transfer coefficient and pressure decline values at shell side by Kral [9] When they have made comprehensive comparison on the most important geometric factor of helix angle, 40^0 helix angle outperformed the other angles with respect to the heat transfer per unit shell side fluid pumping power or unit shell side fluid pressure decline. The flow patterns in the shell side of the shell and tube heat exchanger with continuous helical baffles were found always rotational and helical due to the peculiar geometry of the continuous helical baffles which resulted in a significant increase in heat transfer coefficient per unit pressure decline in the shell and tube heat exchanger.

The continuous helical baffles when designed well can prevent the flow induced vibration and fouling in the shell side. Similar results on fouling were reported by Murugesan and Balasubramanian[10]. The experimental study of Shell and Tube Heat Exchangers with the use of continuous helical baffles can result approximately ten per cent hike in heat transfer coefficient in comparison with that of traditional segmental baffles for the same shell side pressure decline. Development of non-dimensional correlations for heat transfer coefficient and pressure decline based on the experimental data on proposed

Conclusion

This paper explains the important considerations to be given for the thermal design of shell and tube heat exchanger. It describes the significance of heat transfer area ,number of tubes, tube length and diameter, tube layout, number of shell and tube passes, tube pitch, number of baffles, its type and size, shell and tube side pressure drop. Also found that ANSYS can also be used as tool for the thermal designing.

From the Numerical Experimentation Results it is Confirmed that the Performance of a Tubular Heat Exchanger can be improved by Helical Baffles instead of Segmental Baffles. Use of Helical Baffles In Heat Exchanger Reduces Shell side Pressure drop, ,pumping cost, weight, fouling etc as compare to Segmental Baffle for a new installation. The Ratio of Heat to increased cross flow area resulting in lesser mass flux through out the shell Transfer Coefficient to Pressure Drop as higher than that of Segmental Baffle. The Pressure Drop in Helical Baffle heat exchanger is appreciably lesser as Compared to Segmental Baffle heat exchanger.

Helical Baffle is the much higher than the Segmental baffle because of Reduced By Pass Effect &Reduced shell side Fouling. The Helical Baffle is three times Higher than the Segmental Baffle. The heat transfer coefficient increases with increase in Reynolds number in shell-and-tube heat exchanger for both hot fluid inlet and cold fluid inlet. The Nusselt number increases with increase in Reynolds number in shell-and-tube heat exchanger for both hot fluid inlet and cold fluid inlet. The value of temperature constants and decreased with increase in Reynolds number. The value of pressure drop gradually increases with increase in Reynolds number.

Nomenclature

S_m = Area of the shell side cross flow section (m).
P_t = Tube pitch (m).
D_o = Tube outside diameter (m).
D_i = Tube inside diameter (m
$D_{s=}$ Shell inside diameter (m).
$L_{b=}$ Baffle spacing (m)
$L_{s=}$ Length of shell (m).
L_t = Length of tube (m).
$t_{b=}$ Tube thickness (m).
$G_{s=}$ Shell side mass velocity (kg/ m^2-s).
$G_{t=}$ Tube side mass velocity (kg/ m^2-s).
$U_{s=}$ Shell side linear velocity (m/s).
$\mu_{s=}$ Shell side fluid Viscosity (N-s/ m^2).
$\mu_{t=}$ Tube side fluid viscosity (N-s/ m).
$\mu_{w=}$ Viscosity a wall temperature (N-s/ m^2).
C_{ps} = = Shell side fluid heat capacity (kJ/kg'K).
$C_{pt=}$ = Tube side fluid heat capacity (kJ/kg'K).
$K_{s=}$ Shell side fluid thermal conductivity (kJ/s-m'K).
$K_{t=}$ Tube side fluid thermal conductivity (kJ/s-m'K)
h_o= Shell side heat transfer coefficient (W/ m^2'K).
h_i = Shell side ideal heat transfer coefficient (W/ m^2'K).
$N_{b=}$ Number of baffles
$N_{t=}$ Number of tubes
f = Friction factor
ΔP_s= Shell side pressure drop (Pa).
$N=_p$ Number of tube passes
$U_{t=}$ Tube side linear velocity (m/s).
$m_{s=}$ Mass flow rate of the fluid on shell side (kg/s)
$m_{t=}$ Mass flow rate of the fluid on tube side (kg/s).
$\rho_{s=}$ Shell side fluid density (kg/m^3).
$\rho_{t=}$ Tube side fluid density (kg/m^3).
R_{es} Shell side Reynolds number

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