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Thermal Design of Shell and Tube Heat Exchanger

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Abstract- The purpose of this paper is to design the shell and tube heat exchanger for Diesel Locomotive. A characteristic of heat exchanger design is the procedure of specifying a design, then calculating the heat transfer coefficients, heat transfer surface area and pressure drops and checking whether the assumed design satisfies all requirements. The inlet temperature of shell and tube heat exchanger is about 86°C. This temperature must be cooled down to a constant temperature 78°C. The design has been done using Bell Delaware method in order to obtain various dimensions such as shell, tubes, baffles etc. Then the thermal simulation in COMSOL Multiphysis has been performed by applying several thermal loads on variable baffle number. Viscous fluids go on the shell side, since this will usually improve the rate of heat transfer. The pressure values from the simulations results compared with theoretical.

Indexed Terms- COMSOL Multiphysis, Diesel Locomotive, and Shell and tube heat exchanger

I. INTRODUCTION

Heat exchangers are devices used to enhance or facilitate the flow of heat. Every living thing is equipped in some way or another with heat exchangers. They are commonly used as oil coolers, power, condensers, preheaters and steam generators in both fossil fuel and nuclear-based energy production applications.

Shell and tube heat exchangers still take a noted place in many industrial processes. They are widely used because of their robut and flexible design. However, conventional heat exchangers with segmental baffles in shell side have some shortcomings resulting in the relatively low conversion of pressure drop into a useful heat transfer. There are many types of shell and tube heat exchanger but the most common types in use are-

1. U tube heat exchanger
2. Straight tube (1- Pass or 2- Passes).

The optimum thermal design of a STHE involves the consideration of many interacting design parameters which can be summarized as follows;

Part A —Thermal Design
The thermal design of STHE includes:

1. Process fluid assignments to shell side or tube side.
2. Selection of stream temperature specifications.
3. Setting shell side and tube side pressure drop design limits.
4. Setting shell and tube side velocity limits.
5. Selection of heat transfer models and fouling coefficients for shell side and tube side.

Part B — Mechanical Design
The mechanical design of STHE includes:

1. Selection of heat exchanger TEMA layout and number of passes.
2. Specification of tube parameters –size, layout, pitch and material.
3. Setting upper and lower design limits on tube length.
4. Specification of shell side parameters – materials, baffles cut, baffles spacing and clearances.
5. Setting upper and lower design limits on shell diameter, baffles cut and baffle spacing.

To develop calculations there are several design andrating packages available.

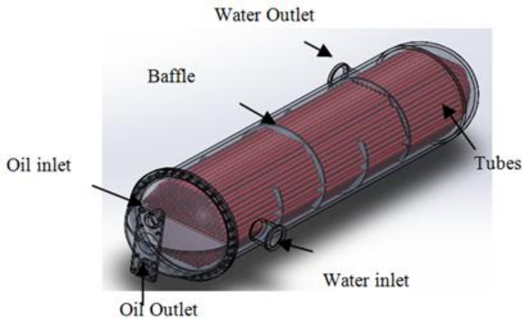

Figure 1. The Basic Flow of Shell and Tube Heat Exchanger

II. MATHEMATICAL MODELLING

A: Thermal Analysis

There are many methods to calculate the shell inside diameter. Therefore following one is the formula for finding the shell inside diameter.

Tube matrix diameter;

$$D_m = 0.637\times\sqrt{\frac{0.87}{0.9}}\times\left(\pi\times d_o^2\times N_t\times P_t^2\right)^{0.5} \quad (1)$$

Shell inside diameter;

$$Ds = 1.075 \quad D_m$$
(2)

Where, Nt = number of tubes,

P_t = tube pitch,

d_o =tube outer diameter.

1) Tube Side Heat Transfer Coefficient:

The tube side heat transfer coefficient hi can be calculated from the Nusselt number correlation.

$$N_u = 0.023\times Re^{0.88}\times Pr^{0.33} \quad (3)$$

$$N_u = \frac{h_t\ d_i}{k_t} \quad (4)$$

$$h_t = 0.023\times\frac{k}{d_i}\times Re^{0.88}\times Pr^{0.33} \quad (5)$$

Where k = thermal conductivity

d_i = tube inner diameter

2) Sell Side Heat Transfer Coefficient:

For the segmental baffled shell and tube heat exchanger, the Bell-Delaware method is usually used in the shell and tube heat exchanger design. In this method, the shell side heat transfer coefficient is determined by correcting the ideal heat transfer coefficient through considering the various leakage and bypass flow streams and ideal heat transfer coefficient is calculated by using the following correlation of Nusselt number.

$$N_u = 1.04(Re_s)^{0.4}(Pr_s)^{0.36} \quad (6)$$

$$N_u = \frac{h_i d_o}{k_s} \quad (7)$$

$$h_i = 1.04\ (k_s/d_o)(Re_s)^{0.4}(Pr_s)^{0.36}$$
(8)

From the above equation the shell side heat transfer coefficient is given by considering the various leakages and by pass flow streams.

$$h_s = \left(J_c J_l J_b J_r J_s\right) h_i \quad (9)$$

Where, J_c =baffle cut correction factor

J_l =baffle leakage correction factor

J_b=bundle bypass correction factor

J_r= laminar flow correction factor

J_s= unequal baffle spacing correction factor

h_i= the ideal tube blank heat transfer.

3) Overall Heat Transfer Coefficient:

The overall heat transfer coefficient U depend on the tube side and shell side heat transfer coefficient and fouling resistance which can be calculated as follows.

$$\frac{1}{U} = \frac{1}{h_t}\left(\frac{d_o}{d_i}\right) + R_t\left(\frac{d_o}{d_i}\right) + \left(\frac{L_w}{k_w}\right)\left(\frac{d_o}{d_i}\right) + R_s + \left(\frac{1}{h_o}\right) \quad (10)$$

Where, L_w= tube wall thickness

k_w = thermal conductivity of tube wall

4) Log Mean Temperature Difference

Heat flows between the hot and cold streams due to the temperature difference across the tube acting as a driving force. The difference will vary with axial location. Average temperature or effective temperature difference for either parallel or counter flow may be written as:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (11)$$

The correction factors, F_T, can be found the theatrically and presented in analytical form. The equation given below has been shown to be accurate for any arrangement having 2-tube passes per shell pass.

$$F_{1\text{-}2}=\frac{\left[\frac{\sqrt{R^2+1}}{R\text{-}1}\right]\left(\ln\left(\frac{1\text{-}P}{1\text{-}PR}\right)\right)}{\left[\frac{2/P\text{-}1\text{-}R+\sqrt{R^2+1}}{2/P\text{-}1\text{-}R\text{-}\sqrt{R^2+1}}\right]} \qquad (12)$$

Where the capacity ratio, R, is defined as:

$$R=\frac{T_{s1}-T_{s2}}{T_{t2}-T_{t1}} \qquad (13)$$

The parameter may be given by the equation:

$$P=\frac{T_{t2}-T_{t1}}{T_{s1}-T_{s2}} \qquad (14)$$

Where, $\Delta T_2 = T_{ho} - T_{co}$ subscript i = inlet

$\Delta T_1 = T_{hi} - T_{ci}$ subscript o = outlet

The total tube outside heat transfer area;

$$Q = UA_O F_T\, \Delta T_{LM} \qquad (15)$$

The total tube heat transfer of length;

$$A_o=(\pi d_o+T_f N_f)N_t L_{cal}$$

(16)

B: Hydraulic Analysis:

1) Pressure Drop for Tube Side

The pressure drop encountered by the fluid making number of passes through the heat exchanger plus the additional pressure drop introduced by the change of direction in the passes are multiplied by the kinetic energy of the flow. Therefore the tube side pressure drop is calculated, by the formula:

$$\Delta p_t=\left(4f_t\frac{LN_p}{di}+4N_p\right)\frac{\rho_t u_t^2}{2} \qquad (17)$$

Where

$$f_t=\left[1.58\ln(Re_t)\text{-}3.28\right]^{-2} \qquad (18)$$

2) Shell side Pressure drop

The shell side pressure drop is calculated as a summation of the pressure drops for three components. They are

(1) Pressure drop in cross flow zone, Δp_c

(2) Pressure drop in window zone, Δp_w

(3) Pressure drop in end zone, Δp_e.

The total pressure drop for shell side is

$$\Delta p_s=\Delta p_c+\Delta p_e+\Delta p_w \qquad (19)$$

The zones covered by this pressure drop are the central baffle compartments for flow between the baffle cuts. The pressure drop in all the central baffle compartments is

$$\Delta p_c=\Delta p_{bl}(N_b\text{-})R_b R_l \qquad (20)$$

Where Δp_bi is the ideal bundle pressure drop for one baffle compartment of the N_b compartments and is based on the mass velocity defined earlier. The expression for Δp_{bi}

$$\Delta p_{bi}=0.002f_i N_{tcc}(\dot{m}_w)^2/\rho_s \qquad (21)$$

The pressure drop and mass velocity in all N_b window zones for turbulent flow (Re>100) are

$$\Delta p_w=N_b\left[\left(2+0.6N_{tcw}\right)\frac{0.001\dot{m}_w}{2\rho}\right]R_l \qquad (22)$$

The pressure drop Δp_e in the two end zones of the tube bundle is

$$\Delta p_e=\Delta p_{bi}\left[1+\frac{N_{tcw}}{N_{tcc}}\right]R_b R_s \qquad (23)$$

Where, N_b= the number baffle

R_b= the bypass correction factor

R_l =the linkage correction factor

R_s= the end zone correction factor.

III. RESULTS AND DISCUSSION

The stream analysis shell-side heat transfer coefficient for single-phase flow hs is used by Equation (9). In this equation JC, JL, JB, JR and JS are the correction factor and then calculated correction factor. Assume number of baffle Nb is 2, 3 and 4.As the baffle spacing is decreased the number of baffles will be increased which will lead to increase in shell side Reynolds's number that will lead to increase in overall heat transfer coefficient. The heat load is computed from Equation (10) and comparing with given heat load. If the calculated heat load is greater than the given heat load, it can be said that the design is satisfied. If not, by increasing Nt, the total number of tubes, and the whole procedure may be repeated till the above condition is satisfied. The result tables of heat load and heat transfer coefficient are shown in Table 1.

Table 1. Result Table of Heat Transfer Coefficient and Heat load

	$N_b = 2$	$N_b = 3$	$N_b = 4$
J_C	1.030960	1.030960	1.030960
J_L	0.786558	0.735269	0.690211
J_B	0.871091	0.908660	0.932108
J_S	0.974289	0.976656	0.977258
J_R	0.98	0.98	0.98
J_u	0.98	0.98	0.98
h_{is}	1049.477	1209.377	1353.992
h_s	693.6637	781.3488	842.8832
h_t	15904	15904	15904
U	618.54	687.32	734.49
Q	160.9	178.8	191.1

The pressure drop for shell side is computed from the Equation (19) and comparing with allowable pressure drop. If the calculated pressure drop is less than the given pressure drop (0.6bar), it can be said that the design is satisfied. If not, calculation can be repeated with a revised value of Nt till the above condition is satisfied. The result table for checking pressure drop for shell side is shown in Table 2.

Table2. Result Table of Pressure Drop for Shell Side versus Number of Baffle

	$N_b = 2$	$N_b = 3$	$N_b = 4$
ΔP_t	0.30590	0.30590	0.30590
ΔP_s	0.2344	0.5623	0.7293

To simplify numerical simulation, some basic characteristics of the process following assumption are made :

1. The shell side fluid is constant thermal properties
2. The fluid flow and heat transfer processes are turbulent and in steady state
3. The leak flows between tube and baffle and that between baffles and shell are neglected
4. The natural convection induced by the fluid density variation is neglected
5. The tube wall temperature kept constant in the whole shell side
6. The heat exchanger is well insulated hence the heat loss to the environment is totally neglected.

In all of the preliminary simulation, flow inside the shell is observed to be turbulent viscous model selected to be K-ε turbulent model [11]. The result is investigated using the heat exchanger model with $N_b = 2$, 3 and 4 baffle spacing for 25% baffle cut. In Figure2, 4 and 6, velocity path lines for four baffles are given for the shell side velocity flow of 1.2 m/sec, inlet boundary condition and outlet boundary condition is pressure, no viscous stress.

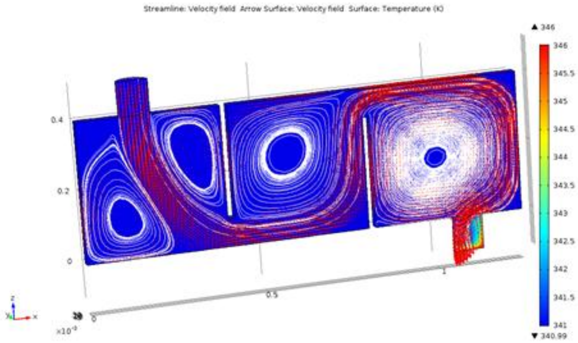

Figure 2.Velocity on Shell Side when N_b=2

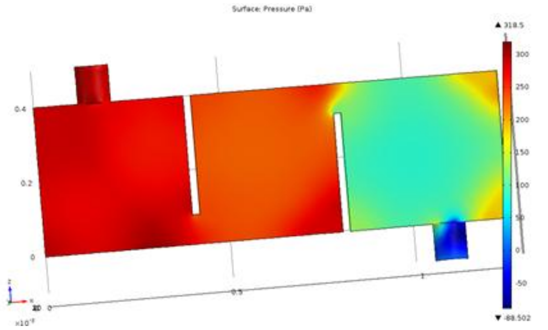

Figure 3.Pressure on Shell Side when N_b=2

The flows hit the baffle plate, and the direction of the flow is changed. In Figure 2, the shell space behind the baffle is not effectively used for cross flow, as marked with a circle. For this reason, the pressure drop occurs high mark and total pressure drop is 0.26 bar in Figure 3.

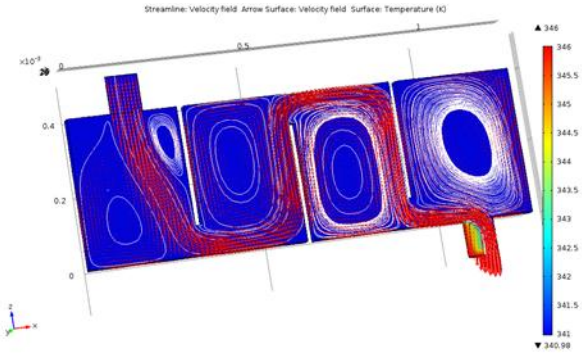

Figure 4.Velocity on Shell Side when N_b=3

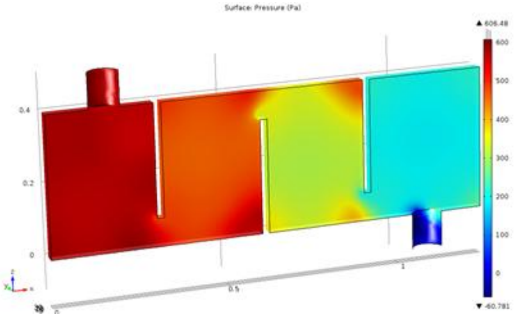

Figure 5.Pressure on Shell Side when N_b=3

In Figure 4, the flow is obseved to be well developed.The cross flow throughout the shell volume and the recirculation zone appears little. So, the pressure drop is effectively average shown in Figure 5, the simulation result is gained the pressure drop 0.58 bar.

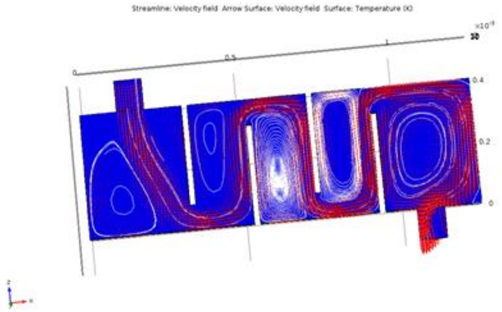

Figure 6.Velocity on Shell Side when N_b=4

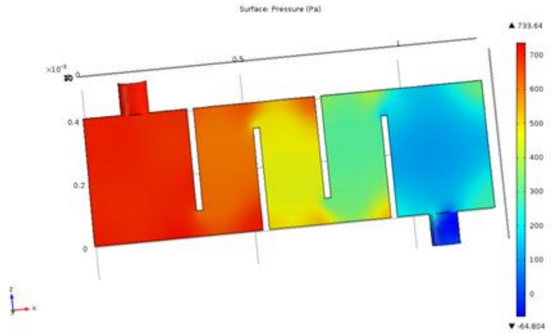

Figure 5.Pressure on Shell Side when N_b=4

In Figrue 6, the velocity stream line of two recirculation zones appear and also occur increasing pressure drop result is 0.72 bar in Figure 7. Hence, it is observed that three baffle gives better pressure drop compare with other baffle over allowable pressure drop for oil cooler for Locomotive. So, number of baffles (3) is good for theoretical and simulation.

Above the calculation result table, determined shell and tube heat exchanger design for diesel locomotive transmission system is shown in Table 3.

Table 3. Verification with Existing Data

Result	Industrial Data	Calculated Data
Shell diameter, m Tube length, m Total no. of tubes Tube diameter, m Number of baffle Heat load, kW Tube side pressure drop (bar) Shell side pressure drop (bar)	0.221 1.17 526 0.007 3 176.6 0.5 0.6	0.238 1.17 556 0.007 3 178.5 0.41261 0.5623

IV. CONCLUSION

In this research, in current numerical analysis, entire geometry to shell and tube heat exchanger including entrance and exist regions were considered as a domain of calculation, theoretical and numerical results have been compared with number of baffle.In this shell diameter (0.238m), number of tubes (556) and number of baffles (3), design heat load is greater than the limited heat load. Suitable tube sides are chosen by Standard Table-3. Design pressure drop for shell and tube side are less than limited pressure drop. This pressure drop is depending on nozzle diameter. The bigger the nozzle diameter, the less pressure drop becomes. The less pressure drop, the better the efficient it gets.

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REFERENCES

[1] J.P. Holman, 1963, Heat Transfer, 8thed, McGraw-Hill Book Company

[2] Bell, K., J., “Prelimimnary design of Shell and Tube Heat Exchanger,” Heat Exchanger Thermal Hydraulic Fundamental And Design, McGraw-Hill Book Co., New York, (1980).

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

[4] Halle H, Chenoweth JM, Wabsganas MW. Shell side water flow pressure drop distribution measurements in an industrial-sized test heat exchanger Eng 1994; 15: 42-56.

[5] Gaddis ES, Gnielinski V. Pressure drop on the shell side of the shell side of shell and tube heat exchangers with segmental baffles. Chem Eng Process 1997; 36: 149-59.

[6] Taborek, J., 1998. Shell and Tube Heat Exchangers in Single Phase Flow. In Handbook of Heat Exchanger Design. Begell House, New York.

[7] Shan, R.K. and D. Sekulic, 2003. Fundamentals of Heat Exchanger Design. Wiley New York.

[8] Professor John R. Thom, 2004, "Wolverine. Single Phase Shell Side Flows and Heat Transfer Data BookII”, Wolverine Tube Inc, www.Wolverine.com.

[9] R. Hesseini, A. Hesseini-Ghaffar, M. Soltani, “Experimental determination of shell side heat transfer coefficient and pressure drop for an oil cooler shell and tube heat exchanger with three different tube bundles”, Applied Therminal Engineering 27 (2007).

[10] M. M. E-Fawal, A. A. Fahmy and B. M, Taher, “ Modelling of Economical Design of Shell and Tube Heat Exchanger using Specificed Pressure Drop”, Journal of American Science, 2011.

[11] COMSOLAB, “Turbulent Flow through Reactor,” Heat transfer Module Library, Version: COMSOL 4.3b, 2012.