Heat Exchanger Design

Su Thet Mon Than, Khin Aung Lin, Mi Sandar Mon

Abstract—This paper is intended to assist anyone with some general technical experience, but perhaps limited specific knowledge of heat transfer equipment. 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 requirements or not. The purpose of this paper is how to design the oil cooler (heat exchanger) especially for shell-and-tube heat exchanger which is the majority type of liquid-to-liquid heat exchanger. General design considerations and design procedure are also illustrated in this paper and a flow diagram is provided as an aid of design procedure. In design calculation, the MatLAB and AutoCAD software are used. Fundamental heat transfer concepts and complex relationships involved in such exchanger are also presented in this paper. The primary aim of this design is to obtain a high heat transfer rate without exceeding the allowable pressure drop. This computer program is highly useful to design the shell-and-tube type heat exchanger and to modify existing deign.

Keywords—Shell-and-Tube Heat Exchanger, MatLAB and AutoCAD

I. INTRODUCTION

THE process of heat exchange between two fluids that are at different temperatures and separated by a solid wall occurs in many engineering applications. The device used to implement this exchange is termed a heat exchanger, and specific applications may be found in space heating and air-conditioning, power production, waste heat recovery, and chemical processing.

The problem of heat transmission is encountered in many industries and because of the diversity in the fields of application there exist countless difference in detail. However, the calculation principle underlying the problem to design a heat exchanger are everywhere the same, and it is the purpose of this paper to present design calculations rather than to deal with the details of individual problems and special cases [1].

A computer program in MatLAB has been written for shell and tube heat exchanger design. Normally, the heat exchanger design calculations are seems to be complex and repeated calculations are required with assumed and/or fixed data. So, many thermodynamic and fluid dynamic parameters are encountered into designs procedure and therefore this condition will satisfy and save the time by using a computer.

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With the help of this computer program, not only the shell and tube heat exchanger can be designed without knowing the detail design calculation but also plant engineer can modify the currently used heat exchanger for some necessary changes, such as blocking the blown out tubes without much effects to operation, changes in fluid flow rate according to other process, etc.

It is hoped that this paper will aid in classifying the many details questions that arise during design calculation and this will support practicing engineers to apply the formal backgrounds in fluid flow and heat transfer to the practical problems posed by the design, selection, testing, or installation of the shell and tube heat exchanger.

II. DESIGN CONSIDERATION

The designer must consider several factors that influence the shell-side heat transfer coefficient that, in turn, determine the rate of heat transfer in the shell-side.

1. When baffles are provided, the system directs the shell-fluid from axial flow to top-to-bottom flow or side-to-side flow with the effect that the heat transfer coefficient is higher than for undisturbed flow along the axes of the tubes [2].
2. Patterns of tube layout influence turbulence and hence heat transfer coefficient e.g. triangular pitch gives greater turbulence than square pitch. And under comparable conditions of flow and tube size the heat transfer coefficient for triangular pitch are roughly 25% greater than for square pitch [3].
3. Closer the baffle spacing, greater is the number of times the shell-fluid is to change its direction resulting in greater turbulence [4].
4. Shell-side coefficient is also affected by tube size, clearance and fluid-flow characteristics [5]
5. Shell-side flow area varies across the bundle diameter with the different number of tube clearances in each longitudinal row of tubes. That's why there is no true shell-side flow area by which the mass velocity of the shell-fluid can be computed. The correlation obtained for fluids flowing in tubes is obviously not applicable to fluids flowing over the tube bundles punctuated with segmental baffles [3].
6. There are some terms used in heat exchanger specification problems and their solutions, which are often confused. These are 'rating', 'design' and 'selection'. 'Rating' defines as the computational process in which the inlet flow rates and temperatures, the fluid properties, and the heat exchanger parameters are taken as input and the outlet temperatures and thermal duty (if the exchanger length is specified) or the required length of the heat exchanger are calculated as output [2].

'Design' defines as the process of determining all essential constructional dimensions of an exchanger that must perform a given heat duty and respect limitations on shell-side and tube-side pressure drop. 'Selection' defines choosing a heat exchanger from among a number of units already existing [2].

Most designers employ empirical relations with a cut-and-try approach that depends on their judgment and experience for convergence on a new design by extrapolation from tested units. However, an analytical approach is easier to follow for the less experienced designer, since it shows the basic relationships. Our primary concern in this paper shall be thermal analysis based on analytical approach developed by Wolverine.

III. INITIAL CONDITIONS AND REQUIREMENTS

In general, the design of most heat exchangers involved initial conditions in which the following variables are known and assumed data:

1. flow rate of fluids
2. temperature range of fluids
3. length/tube and arrangement of tubes

With this information, it must prepare a design for the optimum exchanger that will meet the required process conditions. Ordinarily, the following results must be determined.

1. Number of tubes and shell diameter
2. Heat transfer rate
3. Overall heat transfer coefficient
4. Tube side and shell side pressure drop

IV. DESIGN PROCEDURE

This computer program was divided into three main steps.
1. Calculation of total number of tubes
2. Calculation and Checking of heat load
3. Calculation and Checking of pressure drop for tube and shell side

A. Calculations of Total Number of Tubes

The suitable tube size, tube diameter (D_t) and tube material are chosen first. The tube configuration has to be decided. In our program the equilateral triangular pitch geometry is set since it is mostly used. No other configuration is available in our program. Total number of tubes must be calculated by using following equations.

$$N_t = CTP \frac{\pi D_s^2}{4CLP_t^2} \qquad (1)$$

$$N_t = CTP \frac{\pi D_{otl}^2}{4CLP_t^2} \qquad (2)$$

Fig.1 reproduced from Tabordk defines the principal heat exchanger dimensions. D_{otl} is the outer tube limit diameter and D_{ctl} is the centerline tube limit diameter ($D_{ctl} = D_{otl} - D_t$) where D_t is the outside diameter of the tubes). The baffle cut height is shown as a height L_{bch}; the value of the baffle cut B_c is (L_{bch}/D_s) x100%.

Fig. 1 Baffle and Tube Bundle Geometry [6]

The diametral clearance between the shell internal diameter D_s and outer tube limit diameter D_{otl} is L_{bb}. One half of L_{bb} is the width of this bypass channel. A pass partition lane is shown with a width of L_p. The diametral clearance between the shell internal diameter D_s and the diameter of the baffle D_b is L_{sb}, where the gap is equal to $L_{sb}/2$.

Fig. 2 Baffle Cut and Clearance [6]

The dimensions D_s, D_{otl}, baffle cut (% of D_s), and L_{bb} and L_{sb}, shown in Fig. 2, can be obtained from a tube layout drawing of the heat exchanger [6].

B. Calculation and Checking of Heat Load for U-Tube

Design procedures of heat load check with calculation of heat transfer coefficient of shell and tube side are as shown in Fig. 3.

1. Calculation of Tube-side Heat Transfer Coefficient

Because of total number of tube and total tube-side flow rate are known, tube side mass flow rate (G_1) can be solved as total flow rate divided by the tube-side flow passage area per pass. In our program, removable inserts are installed inside the tubes. Use of tube inserts (wire mesh or twisted tape types) is highly effective in laminar flows inside tubes. Installation of inserts on the tube-side of heat recovery units to increase energy recovery via a larger overall heat transfer coefficient and smaller temperature approaches. To overcome the fact that laminar flow heat transfer coefficients are almost independent of fluid velocity and performance is difficult to improve using plain tubes even resorting to large pressure drops, either inserts or external low fins may be the simple solution. Properly designed units with tube inserts normally are much

smaller in size and have smaller or equal pressure drops as conventional plain tube units [7].
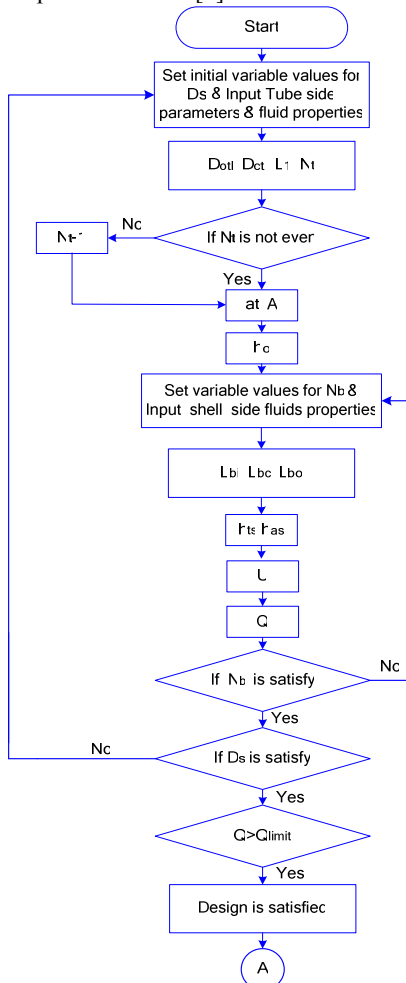

Fig. 3 Flow Chart for Heat Load

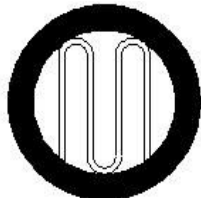
Fig. 4 Tube with Insert Plate

According to the Fig. 4, use the following equation for the tube-side flow passage area.

$$a_t = \pi d_i^2 / 4 \qquad (3)$$

$$a_f = [\, (l_f \, n_f) + (\pi \, d_f \times 3/2) \,] \, t_f \qquad (4)$$

$$a_n = a_t - a_f \qquad (5)$$

Where, a_t is tube cross flow area, a_f is tip area of plate fin, a_n is the net flow area for one tube. To calculate tube-side flow passage area for one pass, a_n is multiplied by N_t/one pass. And then, calculate tube side mass flow rate G_1.

After that, Reynold number of the flow inside tube can then be calculated by the following equations.

For Laminar flow,

$$Nu = \frac{hd}{k} = 1.86 \left(Re.Pr \frac{d}{L} \right)^{1/3} \varphi^{0.14} \qquad (6)$$

For Transistion flow,

$$Nu = \frac{hd}{k} = 0.116 (Re^{2/3} - 125) Pr^{1/3} \varphi^{0.14} \left[1 + \left(\frac{d}{L} \right)^{2/3} \right] \qquad (7)$$

For Turbulent flow,

$$Nu = \frac{hd}{k} = 0.023 \, Re^{0.80} . Pr^{1/3} \varphi^{0.14} \qquad (8)$$

The criterion of distinguishing between laminar and turbulent flow is the observed mixing action. Nusselt number (Nu) is a function of Reynolds number (Re) and Prandtl number (Pr). Nu of the flow inside tube can then be calculated by above Reynold number equation. Tube-Side Heat Transfer Coefficient is also calculated by above Reynold number equation [7].

2. Calculation of Shell-side Heat Transfer Coefficient

The stream analysis shell-side heat transfer coefficient for single-phase flow h_{as} (α_{ss}) is used by following equation.

$$\alpha_{ss} = (J_C \, J_L \, J_B \, J_R \, J_S \, J\mu \,) \, \alpha_1 \qquad (9)$$

In this expression, α_1 is the ideal tube bank heat transfer coefficient calculated for all the flow across the tube bundle and J_C, J_L, J_B, J_R and J_S are the correction factor. J_C is calculated by using (10).

$$J_C = 0.55 + 0.72 \, F_C \qquad (10)$$

$$F_C = 1 - 2F_W \qquad (11)$$

In this equation, the fraction of the cross-sectional area occupied by the window (F_w) is calculated from (12).

$$F_W = \frac{\theta_{ctl}}{360} - \frac{Sin \, \theta_{ctl}}{2\pi} \qquad (12)$$

For a well-designed unit, J_C typically ranges in value from 0.65 to 1.175. The maximum value of Baffle leakage correction J_L is 1.0. To calculate J_L, (13) is used.

$$J_L = 0.44(1-r_s) + [1-0.44(1-r_s)] exp(-2.2 r_{lm}) \qquad (13)$$

From this equation r_s and r_{lm} are calculated from 14 and 15.

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}} \qquad (14)$$

$$r_{lm} = \frac{S_{sb} + S_{tb}}{S_m} \qquad (15)$$

After that, J_L is calculated.
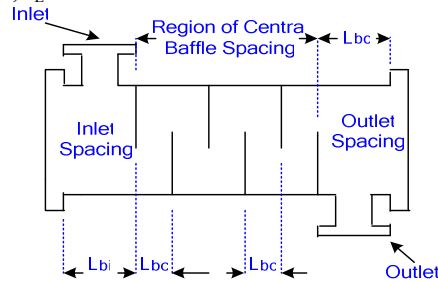
Fig. 5 Single-segmental Shell and Tube Heat Exchanger Showing Baffle Spacing [8]

Fig. 6 Tube Length Definition for U-Tube [8]

Fig. 5 depicts a single-segmental shell-and-tube bundle geometry with fixed tube sheets at both heads in which the shell-side flow makes one shell pass from one end of the tube bundle to the other with the flow directed across the tube bundle by the baffles. The inlet, central and outlet baffle spacing are shown and are identifies as L_{bi}, L_{bc} and L_{bo}, respectively. By changing N_b, baffle spacings L_{bi}, L_{bc}, L_{bo} are change.

L_{bi} and L_{bo}, are often equal in length to L_{bc}, expect when the first and last baffle compartments must be enlarged to allow for the placement of the respective shell-side nozzles. The baffle layout is determined from the inlet, central, and outlet baffle spacing and the effective tube length. The effective tube length L_{ta} is equal to the total tube length less the combined thickness of the two tube sheets. The number of baffles (an integer) and baffle spacing can be determined from these values. The effective length for determining the baffle spacing for U-tube exchanger includes the straight length of the tube plus $D_s/2$, where D_s is the shell internal diameter. Thus the baffle spacing at the U-Bend, shown in Fig.6, should include the tube straight length in this compartment plus $(D_s/2)$ or 0.3 D_{otl} [8].

By using the following Equation, baffle spacing are calculated.

$$L_{bi} = \{1.25/N_b+1\}+0.005 \tag{16}$$

$$L_{bo} = L_{bi} + 0.3\, D_{otl} \tag{17}$$

$$L_{bc} = \{1.25-2L_{bi}\}/\{N_b-1\} \tag{18}$$

The maximum limit of Bundle by pass correction factor J_B is 1.0 at $r_{ss} \geq \frac{1}{2}$. To calculate J_B, use F_{sbp}.

$$J_B = \exp[-C_{bh}F_{sbp}(1-\sqrt[3]{2r_{ss}})] \tag{19}$$

The empirical factor $C_{bh}=1.35$ for laminar flow ($100 \geq$ Re) and $C_{bh}=1.25$ for transition and turbulent flows (Re>100). To evaluate this expression, one requires the ratio of the bypass to the cross flow area F_{sbp}, and the ratio r_{ss} of the number of sealing strips N_{ss} (number of pairs if any) passed by the flow to the number of tube rows crossed between baffle tips in one baffle section N_{tcc}. To calculate F_{sbp}, use the following equation.

$$F_{sbp} = \frac{S_b}{S_m} \tag{20}$$

$$S_b = L_{bc}[(D_s-D_{otl})+L_{pl}] \tag{21}$$

In above expressions, S_b is the bypass area; L_{pp} represents the width of the bypass lane between tubes. For situations without a pass partition lane or for such a lane normal to the flow direction, set $L_{pl}=0$ while for a pass partition lane parallel to the flow direction L_{pl} is equal to $\frac{1}{2}$ the actual dimension of the lane or can be assumed to be equal to a tube diameter D_t.

$$r_{ss} = \frac{N_{ss}}{N_{tcc}} \tag{22}$$

$$N_{tcc} = \frac{D_s}{L_{pp}}\left[1-2\left(\frac{B_c}{100}\right)\right] \tag{23}$$

Where $L_{pp}=0.866L_{tp}$ for a 30° layout, $L_{pp}=L_{tp}$ for a 90° layout and $L_{pp}=0.707L_{tp}$ for a 45° layout. This expression has a maximum limit of $J_B=1$ at $r_{ss} \geq 1/2$ [8].

The unequal baffle spacing correction factor J_S accounts for the adverse effect of an inlet baffle spacing L_{bi} and/or outlet baffle spacing L_{bo} larger than the central baffle spacing L_{bc}. Some exchangers have larger baffle spacing in the inlet and outlet nozzle compartments compared to the central baffle spacing, allowing placement of the shell-side nozzles without interference with the body flanges and without overlapping the first baffle. The flow velocity in these compartments is thus lowered and has an adverse influence on heat transfer. The correction factor $J_S<1.0$ for larger inlet and outlet spacing than the central baffle spacing. For inlet and outlet baffle spacing equal to the central baffle spacing, no correction is required and $J_S=1.0$. The value for J_S is determined directly from the effect on the flow velocity and is given by the following expression.

$$J_s = \frac{(N_b-1)+(L_{bi}/L_{bc})^{1-n}+(L_{bo}/L_{bc})^{1-n}}{(N_b-1)+(L_{bi}/L_{bc})+(L_{bo}/L_{bc})} \tag{24}$$

Where n=0.6 for turbulent flow and n=1/3 for laminar flow. The number of baffle compartments N_b is determined from the effective tube length and the baffle spacing [6]-[8].

In laminar flows, heat transfer is reduced by the adverse temperature gradient formed in the boundary layer as the flow thermally develops along the flow channel. The laminar flow correction factor J_R accounts for this effect. For laminar shell-side flow $J_R<1.0$ (i.e. for $100 \geq$ Re) while for Re>100, no correction is needed and $J_R=1.0$.

$$J_R=(J_R)_{20}=\left(\frac{10}{N_c}\right)^{0.18} \quad (\text{For}20\geq\text{Re}) \tag{25}$$

$$N_c = (N_{tcc} + N_{tcw})(N_b+1) \tag{26}$$

Where N_c is the total number of tube rows crossed by the flow in the entire heat exchanger and N_{tcw} is the number of tube rows crossed in the window area. For Re>20 but Re>100, the value is prorated in (27). The minimum value of J_R in all cases is 0.4.

$$J_R = (J_R)_{20} + \left(\frac{20-\text{Re}}{80}\right)\left[(J_R)_{20}-1\right] \tag{27}$$

For laminar shell-side flow $J_R<1.0$ while for Re>100, no correction is needed. The correction factor J_μ is greater than 1.0 for heating the shell-side fluid and vice-versa for cooling the shell-side fluid.

To calculate ideal tube bank heat transfer coefficient h_s (α_1), use the following equation.

$$\alpha_1 = j_1 c_p m^\circ Pr^{-2/3} \tag{28}$$

$$j_1 = a_1 \left(\frac{1.33}{L_{tp}/D_t} \right)^a Re^{a_2} \tag{29}$$

$$a = \frac{a_3}{1 + 0.14 Re^{a_4}} \tag{30}$$

For heat load check, the overall heat transfer coefficient, U, can be achieved from (31).

$$U_o = \frac{1}{\left[\frac{r_o}{r_i} \right]\left(\frac{1}{h_i} + \frac{1}{h_{si}} \right) + \left[\frac{r_o}{k_w} \right] \ln\left(\frac{r_o}{r_i} \right) + \frac{1}{h_{so}} + \frac{1}{h_o}} \tag{31}$$

In this equation, fouling resistance (factor) associated with fluid outside tube R_{fo} ($1/h_{so}$) and fouling resistance (factor) associated with fluid inside tube R_{fi} ($1/h_{si}$). K is wall thermal conductivity. And then, calculate overall heat transfer coefficient.

The heat transfer area A is calculated by the following equation.

$$A = \left\{ (\pi d_o L_1) + 2\left[nL_f + \frac{\pi d_f x3}{2} \right] L_1 \right\} N_t \tag{32}$$

In this equation, next term is represented for insert plate heat transfer area. F_T is the correction factor. Log Mean Temperature Difference is calculated by following equation [7].

$$\Delta T_m = \Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1} \right)} \tag{33}$$

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

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

The heat load is computed from (34) and comparing with given heat load.

$$q = U A F_T \Delta T_{lm} \tag{34}$$

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 TN, the total number of tubes, and the whole procedure may be repeated till the above condition is satisfied.

C. Calculation and Checking of Pressure Drop

Design procedures of Pressure Drop check for shell and tube side are as shown in Fig .7.

The pressure drop for tube side is computed from the summation of (35) and (36) and comparing with allowable pressure drop. For the nozzle losses, it is usually sufficient to calculate the loss for each nozzle at about three times the velocity head in the nozzle [12].

$$\Delta p_{noz} = 3 \frac{\rho V_{noz}^2}{2g_c} \tag{35}$$

Where, v_{noz} is calculated at the smallest cross-section area for flow (i.e. highest velocity) in the nozzle. The combined header and tube entrance losses are estimated in a similar way, but using the velocity inside the tube, V_i.
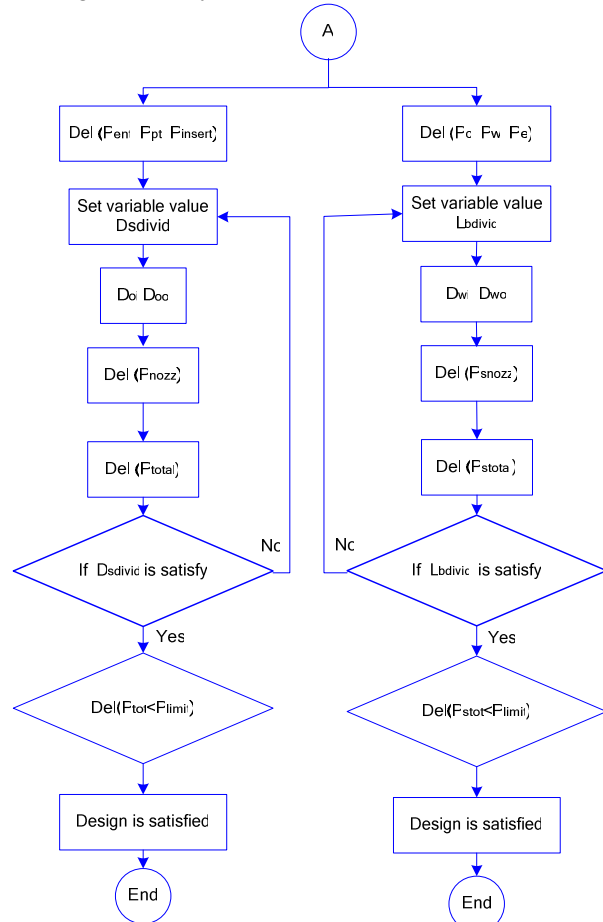

Fig. 7 Flow Chart for Pressure Drop on Shell and Tube Side

$$\Delta P_{ent} = 3\left(\frac{\rho V_i^2}{2g_c} \right) \tag{36}$$

The pressure drop for shell-side flow is equal to the sum of the inlet nozzle pressure drop, the bundle pressure drop and the outlet pressure drop. The inlet and outlet nozzle pressure drops can be approximated as being equal to two velocity heads each.

The bundle pressure drop is equal to the sum of the cross-flow pressure drops Δp_c, the window pressure drops Δp_w, and the two end zone pressure drops (first and last baffle compartments) Δp_e as illustrated in Fig. 8 [6]-[8].

So, the bundle pressure drop for shell side is computed from using (37) and comparing with allowable pressure drop.

$$\Delta p_{total} = \Delta p_c + \Delta p_w + \Delta p_e \tag{37}$$

$$\Delta p_c = \Delta p_{bl} (N_b - 1) R_B R_L \tag{38}$$

$$\Delta p_{bl} = 0.002\, f_l\, N_{tcc}\, m^{\cdot 2}/\rho\, R_\mu \tag{39}$$

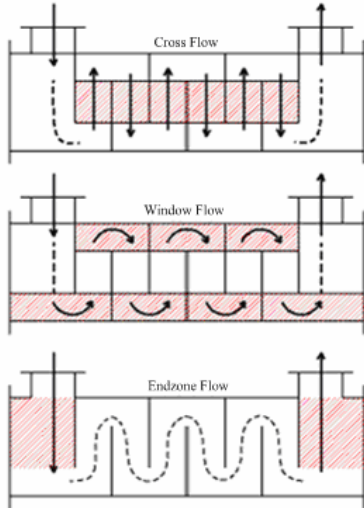$$f_l = b_1 \left(\frac{1.33}{L_{tp}/D_t} \right)^b Re^{b_2} \tag{40}$$

Fig. 8 Pressure Drop Regions in Shell-side Flow [8]

$$b = \frac{b_3}{1 + 0.14 Re^{b_4}} \tag{41}$$

$$R_\mu = \left[\frac{\mu}{\mu_w} \right]^{-m} \tag{42}$$

$$R_B = \exp[-C_{bp} F_{sbp}(1 - \sqrt[3]{2r_{ss}})] \tag{43}$$

$$R_L = \exp\left[-1.33(1 + r_s)r_{lm}^p \right] \tag{44}$$

$$P = -0.15(1+r_s) + 0.8 \tag{45}$$

Where Δp_{bl} is the ideal bundle pressure drop for one baffle compartment of the N_b compartments. f_l is the friction factor and the empirical constants. R_μ is the viscosity correction factor and R_B is the bypass correction factor. The limit of R_B is 1 for $r_{ss} \geq 1/2$. Use C_{bp} is 4.5 for laminar flow ($100 \geq Re$) and C_{bh} is 3.7 for transition and turbulent flows ($Re>100$). R_L is the leakage correction factor.

$$\dot{m}_w = \frac{M}{\sqrt{S_m S_w}} \tag{46}$$

$$\Delta p_w = N_b \left[(2 + 0.6 N_{tcw}) \frac{0.001 \dot{m}_w^2}{2\rho} \right] R_L R_\mu \tag{47}$$

Where M is the shell-side flow rate in kg/s and Δp_w and $m^\circ{}_w$ are the pressure drop and mass velocity in all N_b window zones for turbulent flow ($Re>100$).

$$\Delta p_e = \Delta p_{bl} \left[1 + \frac{N_{tcw}}{N_{tcc}} \right] R_B R_S \tag{48}$$

$$R_S = \left[\frac{L_{bc}}{L_{bo}} \right]^{2-n} + \left[\frac{L_{bc}}{L_{bi}} \right]^{2-n} \tag{49}$$

Δp_e is the pressure drop in the two end zones of the tube bundle and R_s is the pressure drop correction for unequal baffle spacing at the inlet and/or outlet with respect to the central baffle spacing. For all baffle spacing of equal length, R_s is 2.0. 'n' is 1 for laminar flow and n is 0.2 for turbulent flow [6]-[8].

V. CASE STUDY

A transmission oil cooler is designed according to the MatLAB program discussed above.

TABLE I
INPUT DATA FOR TRANSMISSION OIL COOLER FOR DIESEL
HYDRAULIC LOCOMOTIVE

	Tube side	Shell side
Fluid	Hydraulic oil	water
Inlet Temperature (˚C)	105	77
Outlet Temperature (˚C)	80.7	79.9
Flow rate (m³/hr)	27.7	84
Heat load (kW)	282	282
Pressure drop (bar)	0.6	0.3
Max working pressure (bar)	15	2

The following figures are obtained by using the MatLAB Program.
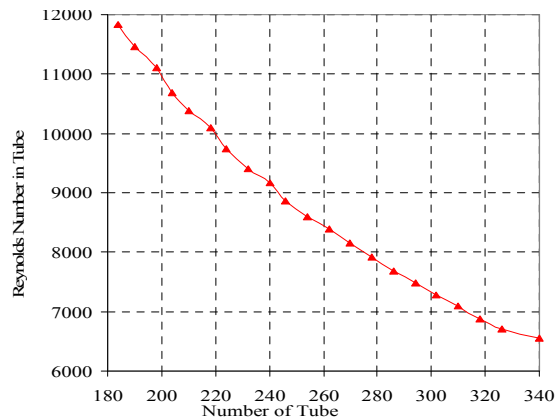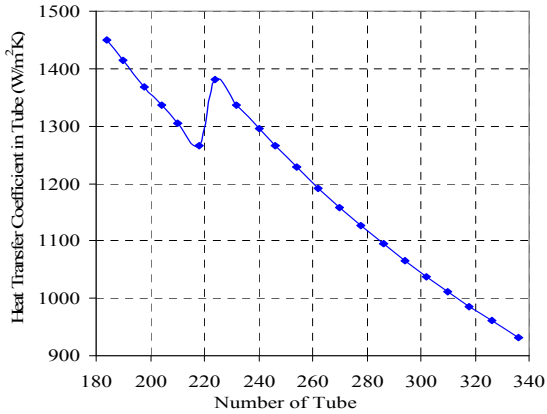

Fig. 9 A fact of Reynolds Number on Number of Tubes

Fig. 10 A fact of Heat Transfer Coefficient on Number of Tube

It can be seen that in Fig. 9 and Fig. 10 shows that the Re and h_o are gradually decreased corresponding to as high as N_t. Because of mass flow rate are constant, velocity are increase and Re, h_o are also decrease. Between total number of tube 220 and 240, there are a slight rise in heat transfer coefficient because of changing from turbulent flow to transition flow.
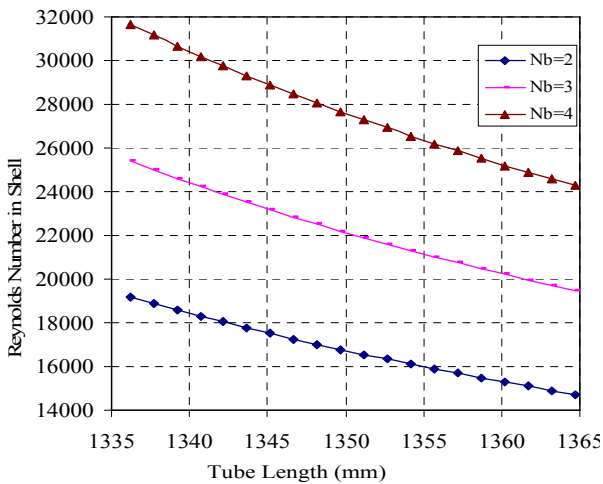

Fig. 11 A fact of Reynolds Number on Number of Baffles and Length of Tube

The decreasing pattern of curves of Reynold Number and heat transfer coefficient shown in Fig. 11 and Fig. 12 shows that the Re and h are gradually decreased corresponding as high as Tube Effective Length (L_1). However, this graph also describes, due to Number of Baffles increase, Reynold Number increase.

The result table for Heat Load is shown in Fig. 13. Between shell diameter 320m and 330m, there are also a slight rise in heat transfer coefficient because of changing from turbulent flow to transition flow.
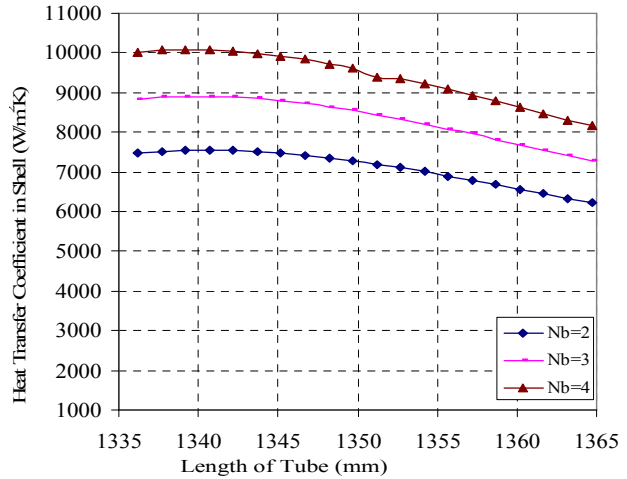

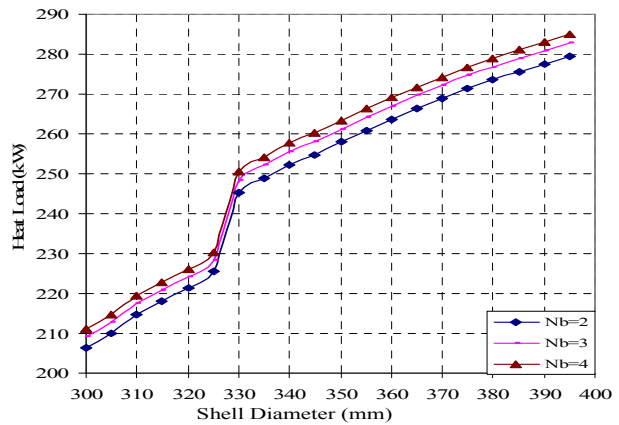Fig. 12 A fact of Heat Transfer Coefficient on Number of Baffles and Length of Tube

Fig. 13 A fact of Heat Load on Shell Diameter

TABLE II COMPARISON WITH INDUSTRIAL DATA

Result	Industrial Data	Calculated Data
Shell diameter, m	0.385	0.39
Tube length, m	1.25	1.25
Total no. of tubes	248	318
Tube diameter, m	0.014	0.0127
Number of Baffle	4	4
Heat Load, kW	282	283.02
Tube side pressure drop	0.6	0.4032
Shell side pressure drop	0.3	0.1652

V. DISCUSSION AND CONCLUSION

There are few limitations to be considered in program. To reduce size and cost of heat exchanger, U-Bend Exchangers

are chosen. The arrangements of tubes are only for a widely used triangular pattern which permits the use of more tubes. The limitation of shell diameter is within the minimum 0.3 m and maximum 0.4 m. It is still assumed that there is no advantage in varying less than the allowable pressure drop and baffles will be employed within the minimum (one fifth of the shell diameter) and maximum (inside diameter of the shell) spacing. These limitations stem from the fact that at wider spacing the flow tends to be axial rather than across the bundle and at closer spacing there is excessive leakage between baffles and the shell. If the shell side pressure drop is higher than the acceptable limits, the baffles spacing is increased slightly, the shell side pressure drop will be satisfactory. If the tube side pressure is not within acceptable limits, the calculation can be repeated with a revised value of total number of tube because of the pressure drop depends on the square of the velocity.

There are many types heat baffle. Among them, segmental baffle is selected in this design. This type of baffle is probably the most popular. It is a circle of near shell diameter from which a horizontal or vertical portion has been cut. The cut-out portion which represents the free flow area for the shell side fluid is usually from 20 to 50 percent of the open shell area. In this design, 25 percent baffle cut is used according to the method of Wolverine. The limitation of number of baffle is 2, 3 and 4.

Shell diameter for calculated design is nearly the same the existing design. But total numbers of tubes are more than existing design because of tube diameter changes. Furthermore, use simple insert flat plate in tubes. By using of tube inserts (flat plate) are highly effective in laminar flow inside tubes. Tube inserts flat plate in existing design is more confused than calculated flat plate. In this program, tube diameter is limited to 12.7 mm, so this diameter is less than existing tube diameter 14mm. Because of these facts, increase total number of tube, heat transfer area and heat load is more than limited heat load (283.2 kW). The tube and shell pressure drop for acceptable limits is 0.6 and 0.3 bar. The calculated pressure drop for tube and shell are 0.4032 and 0.1652 bar. The design is satisfied because the pressure drop for both sides is lower than the limited pressure drop. So this computer program is highly useful to design the liquid-to-liquid shell and tube type heat exchanger.

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