

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

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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 device that 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, heat recovery, alternate fuels and other industries. The relation was formulated by Newton and is called Newton's law of cooling, which is given by

$$Q = h \cdot A \cdot \Delta T$$

Where, h is heat transfer coefficient [W/m²K], A is the heat transfer area [m²], and T is the temperature difference [K].

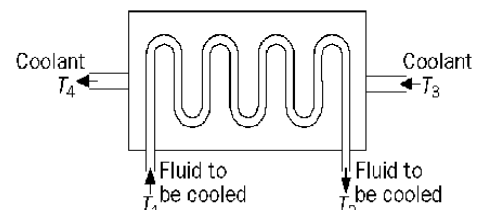


Fig.1: The basic heat transfer mechanism [18]

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.

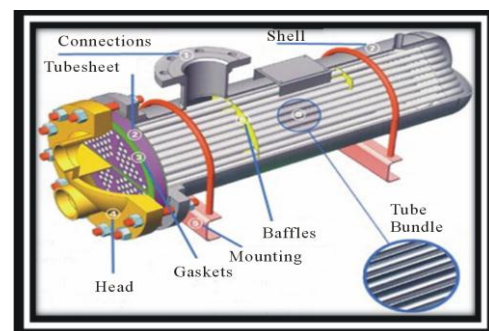


Fig.2: The basic Components Shell and Tube Heat Exchanger [19]

TEMA Standards

STHE are classified and constructed incorporate with the widely used TEMA (Tubular Exchanger Manufacturers Association) standards. Notation system used in TEMA to designate major types of combination, the first letter 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

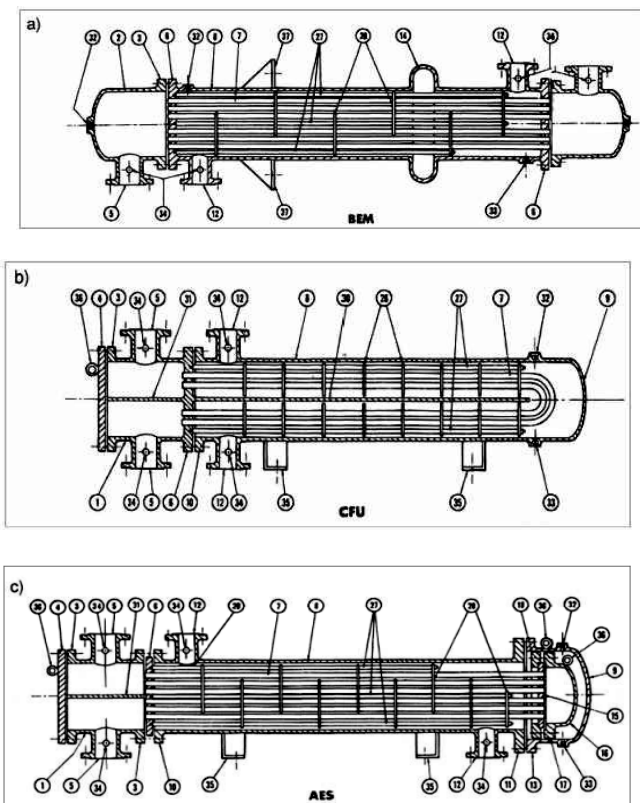


Fig 3: Construction Parts and Connections [10]

- | | |
|-------------------------------------|---|
| 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 et al.[6]In this paper data is evaluated for heat transfer area and pressure drop and checking whether the assumed design satisfies all requirement or not. The primary aim of this design is to obtain a high heat transfer rate without exceeding the allowable pressure drop.

The decreasing pattern of curves of Reynolds Number 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 et al. [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 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.

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 segmental 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 [16][1],

The Data regarding hydraulic oil temperature, pressure, are measured from existing industrial hydraulic power pack machine. Using this data, heat potential available in the Hydraulic oil is calculated using energy balance equation.

In order to develop relationships between the heat transfer rate Q, surface area A, fluid terminal 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, The mass flow rate of oil is calculated by energy balance equation

$$Q_o = \dot{m}_h C_{ph} (T_{h1} - T_{h2})$$

$$Q_w = \dot{m}_c C_{pc} (T_{c2} - T_{c1})$$

Heat loss from wall of shell is 5-10%

$$Q_w = 0.9Q_o$$

Flow arrangement selected is counter flow type and accordingly LMTD for this stage is F=0.95)

$$\Delta T_m = \frac{(Th1 - Tc2) - (Th2 - Tc1)}{\ln \frac{(Th1 - Tc2)}{(Th2 - Tc1)}} \times F$$

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 temperature difference 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 ,

$$Q_{act} = U \times A \times F_d T_{LM}$$

The correction factor charts are available from many sources these 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.

Total surface area of heat exchanger is calculated .Assume overall heat transfer coefficient ($U_{0\text{ ass}}$),

$$A = \frac{Q}{U_o \cdot \Delta T_m}$$

Total length of tube,

$$L = \frac{A}{\pi d_i}$$

Mass flow rate of oil is calculated from energy balance equation,

$$\dot{m}_h = \frac{\dot{m}_c C_{pc} (T_{c2} - T_{c1})}{C_{ph} (Th1 - Th2)}$$

Inner diameter of tube,

$$d_i = d_o - 2t$$

Water side film heat transfer coefficient,

Water side film heat transfer coefficient is calculated using following equation,

Area available for gas flow,

$$A_{fl.w} = \frac{\pi}{4} \cdot d_i^2 \cdot NT$$

Velocity of Water,

$$V_w = \frac{\dot{m}_w}{\rho_w \cdot A_{fl.w}}$$

Reynolds number of water,

$$Re_{eg} = \frac{\rho_w \times V_w \times d_i}{\mu_g}$$

Using following correlations for Nusselt for diesel exhaust flow [18]

If Reynolds number (Re_w) < 2300

For developing flow,

$$Nu_w = 1.86 \left[\left(\frac{D}{L} \right) Re_w \cdot Pr_w \right]^{1/3} \times \left[\frac{\mu_w}{\mu_s} \right]^{0.14}$$

For developed flow,

$$Nu_w = 3.66 + \frac{0.0668 \left(\frac{d_i}{L}\right) Re_w \cdot Pr_w}{(1 + 0.04 \left(\frac{D}{L}\right) Re_w \cdot Pr_w)^{2/3}}$$

$$\text{If } 2300 < (Re_w) < 5 \times 10^6$$

The Geielinskiequation[1]

$$Nu_w = \frac{\left(\frac{f}{8}\right) [(Re_w - 1000) Pr_w]}{(1 + 12.7 \left(\frac{f}{8}\right)^{1/2}) [(Pr_w)^{2/3} - 1]} \left(1 + \frac{d_o}{L}\right)^{2/3}$$

$$f = (0.79 \ln Re_w - 1.64)^{-2}$$

Heat transfer coefficient of Water,

$$h_i = \frac{Nu_w \cdot K_w}{d_i}$$

Tube Side Pressure Drop:[10]

The tube side pressure drop can be calculated by knowing the number of tube passes (Np) and length (L) of heat exchanger; the pressure drop for the tube side fluid is given by equation, The change of direction in the passes introduction in the passes introduction an additional pressure drop due to sudden expansions and contractions that the tube fluid experiences during a return that is accounted for allowing four velocity head per pass The total pressure drop of the side becomes:

$$\Delta P = \left(\frac{4fLV_w^2 \rho_w Np}{2d_i}\right) + \frac{4\rho_w NpV_w^2}{2}$$

Oil side film heat transfer coefficient,

The oil side heat transfer is calculated using following equations.

The shell side equivalent diameter (hydraulic diameter), for an equilateral triangular pitch arrangement [1],

$$D_e = \frac{4\left[\frac{P_t^2 \cdot \sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right]}{\frac{\pi d_o}{2}}$$

For a square pitch arrangement,

$$D_e = \frac{4\left[\frac{P_s^2}{4} - \frac{\pi d_o^2}{4}\right]}{\pi d_o^2}$$

Clearance between adjacent tubes,

$$C = P_t - d_o$$

The tube bundle cross flow area, at the center of the shell,

$$A_s = \frac{D_i \cdot C \cdot B}{P_t}$$

Baffle spacing,

$$B = \frac{1}{5} D_i$$

The shell side mass velocity,

$$G_s = \frac{\dot{m}_h}{A_s}$$

Reynolds number of oil,

$$Re_w = \frac{G_s D_e}{\mu_o}$$

Using same correlations for Nusselt number for oil side which are used for water side

Heat transfer coefficient of oil side,

$$h_o = \frac{Nu_o \cdot K_o}{D_e}$$

Overall heat transfer coefficient without fouling,

$$U_{oth} = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k} + \frac{1}{h_o}}$$

Overall heat transfer coefficient with fouling,

$$U_{oth} = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k} + \frac{1}{h_o} + Rf_o + \left(\frac{d_o}{d_i}\right) \times Rf_i}$$

Shell Side Pressure Drop: [10]

The calculation of shell side pressure drop is significantly more complicated as the shell side flow path is considerably more complex. For our purposes, we will use a correlation presented by which can be taken to an appropriate chart and used to get a friction factor. Note that the chart provides a dimensional friction factor (unlike the dimensionless values used for pipe flow). It crosses between the baffles, so the cross will be one more than the number of baffles, N_b. The number of baffles can be determined using the baffle spacing:

$$N_b + 1 = \frac{L}{B_s}$$

The pressure drop is then determined using the Equivalent diameter, cross flow velocity, friction factor, number of crosses, and fluid properties:

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

Where,

$$\phi_s = (\mu_o + \mu_w)^{0.14}$$

N_b = Number of baffles,

(N_b + 1) = Number of times fluid passes to the tube bundle,

Friction factor (f) calculated from:

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

Experimental Overall heat transfer coefficient, U_{oexp}

$$Q_{avg} = Q_{exp} = \frac{Q_w + Q_g}{2}$$

$$Q_{exp} = U_{o exp} \cdot A \cdot \Delta T_m$$

$$U_{o exp} = \left(\frac{Q_{exp}}{A \cdot \Delta T_m}\right)$$

Deviation between theoretical and experiment overall heat transfer coefficient,

$$\% \text{ Deviation} = \frac{U_{o exp} - U_{oth}}{U_{o exp}} \times 100$$

IV FABRICATION AND TESTING OF HEAT EXCHANGER

Design Input Data

Problem taken from the industrial application i.e. heat rises of hydraulic oil in the hydraulic power pack machine. Following are the operating parameters while designing the shell & tube heat exchanger:

1. Inlet temperature of Oil, T_{h1} = 90 °C
2. Outlet temperature of hot Oil T_{h2} = 44 °C
3. Inlet temperature of water, T_{c1} = 30 °C
4. Mass flow rate of water, m_c = 0.042 kg/s
5. Mass flow rate of Oil, m_h = 0.024 kg/s

Table 4.1: Shell and tube exchanger design data– physical properties [7]

Physical Properties	Shell side	Tube side
Fluid	oil	Water
Flow rate (kg/s)	0.024	0.042
Fluid density (kg/m ³)	998	1000
Heat capacity (J/kg. K)	1849	4187
Viscosity (Pa. s)	3.25	1
Thermal conductivity (W/m. K)	0.118	0.61
Prandtl number	3.5	4.187

Final Specifications of Shell and Tube Heat Exchanger Based on the methodology the calculated results for fabrication of STHE are summarized below;

Table 4.2: Calculated Results for specifications of STHE

Design Parameter	Value	unit of measurement
Cold outlet temperature (Tc ₂)	41	^o C
Heat duty (Q _w)	1912.001667	W
LMTD (ΔT _m)	25.94863678	^o C
Heat Transfer Area by LMTD (A)	0.409356051	m ²
Numbers of tubes (N _t)	26	nos
Shell diameter (D _s)	0.154607034	m
Pitch (P _T)	0.02500	m
Baffle spacing (B _s)	0.077303517	m
Numbers of baffles (N _b)	4.174408816	nos
Baffle Cut (B _c)	0.04638211	m
Bundle To Shell Clearance (L _{bb})	0.012000773	m
Outer Tube Limit Diameter (D _{out})	0.142606261	m
Centerline Tube Limit Diameter (D _{clt})	0.129906261	m
Diametric clearance between shell diameter and baffle diameter (L _{sb})	0.003100618	m
Baffle Diameter (D _b)	0.151506416	m
Equivalent Diameter (D _e)	0.052960079	m
Clearance Between Tubes (C)	0.0148	m
Bundle Cross Flow Area (A _s)	0.00643217	m ²
Shell Side Mass Velocity (U _s)	3.749641693	kg/s.m ²
Shell Side Reynolds Number (Re _s)	2920.313538	unitless
Approximate Wall Temperature (T _w)	48.5	deg C
Shell side heat transfer coefficient (h _o)	49.04374442	W/m ² .K
Average velocity in tube (V _w)	0.047012905	m/s
Tube Side Reynolds Number (Re _w)	610.845999	unitless
Tube side Friction factor (f _w)	0.021277835	unitless
Nusselt number (Nu _w)	3.5	unitless
Tube side heat transfer coefficient (h _i)	232.0101596	W/m ² .K
Overall heat transfer coefficient (U _{oth})	38.08694531	W/m ² .K
Tube Diameter (d _o)	0.0127	m
Tube Inner Dia.(d _i)	0.0095	m
Length Of Tube(L)	0.4	m
Shell inside diameter (D _i)	0.144607034	m

Above results which are helps to fabricate 1-2 pass shell and tube heat exchanger for maximum cooling of oil and According to above calculated values fabricated the shell & Tube Heat Exchanger.

Experimental Set-up

A shell and tube heat exchanger (two-pass) was fabricated and tested for its performance using the hydraulic power pack hot oil. The recovered heat is used to heating the water.

Figure 4.1 shows the block diagram of the experimental set up.

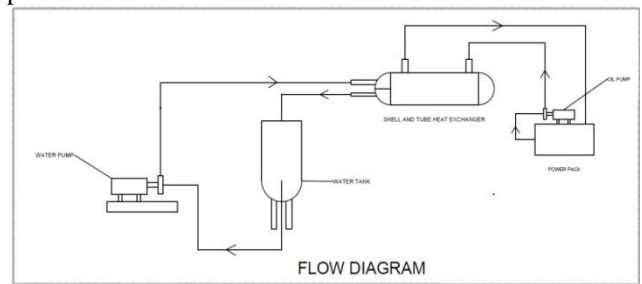


Fig. 4.1:- Block Diagram of Experimental Set up

Actual Experimental Set-up

The actual experiment set up consist of from manufacturing of each components of shell and tube heat exchanger up to the total assembly of set up. Below photographs gives all details of actual experimental set up.



Fig.4.6:- Shell & Tube Heat Exchanger before Trial



Fig.4.7:- Shell & Tube Heat Exchanger during trial

Actual Trial on Experimental Set-up



Fig.4.8:- Temperature Readings Taken During Experimental Test

V RESULTS & DISCUSSION

Fabricated model of shell and tube type heat exchanger was tested for heat transfer performance by using hydraulic power pack. The test was conducted on this shell and tube type heat exchanger for various mass flow rates of oil as well as ass flow rate of water. First kept mass flow rate of the oil

constant and we changed the mass flow rate of the water. During these reading taken the temperature of oil used 65°C, 75°C, 85°C and 95°C. We were used different mass flow rate of oil such as 0.0333, 0.036667, 0.066667 and 0.133333 kg/sec. For each mass rate of oil we are taken four mass flow rates water and taken 16 trails. Total trails on hydraulic power pack oil were taken 64.

Temperature of oil from 65°C to 75 °C the overall theoretical overall heat transfer coefficient (U_{oth}) increases and then above 75 °C it decreases. As the temperature of the oil increases theoretical overall heat transfer coefficient (U_{oth}) decreases gradually. As the mass flow rate of water increases from 0.033333 Kg/s to 0.10000 Kg/s then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

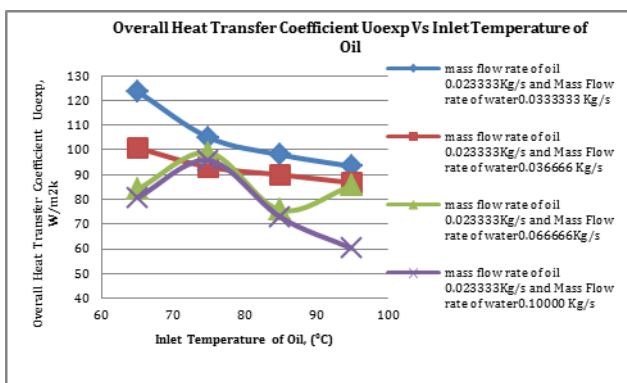


Fig:-5.1 Overall Heat Transfer Coefficient U_{oexp} Vs Inlet Temperature of Oil

Figure 5.1 indicates as the temperature of the oil increases experimental overall heat transfer coefficient (U_{oexp}) decreases gradually. Also mass flow rate of water increases from 0.033333 Kg/s to 0.10000 Kg/s at constant mass flow rate oil i.e. 0.023333 Kg/s, experimental overall heat transfer coefficient (U_{oexp}). If the inlet temperature of oil kept constant then experimental overall heat transfer coefficient (U_{oexp}) decreases. Temperature of oil from 65°C to 75 °C the experimental overall heat transfer coefficient (U_{oexp}) increases and then above 75°C it decreases.

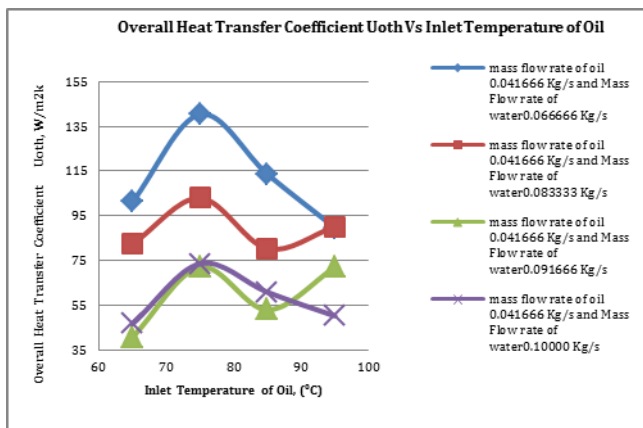


Fig:-5.7 Overall Heat Transfer Coefficient U_{oth} Vs Inlet Temperature of Oil

Figure 5.7 indicates as the temperature of the oil increases theoretical overall heat transfer coefficient (U_{oth}) decreases gradually. Also mass flow rate of water increases from 0.066666 Kg/s to 0.10000 Kg/s at constant mass flow rate oil i.e. 0.083333 Kg/s, theoretical overall heat transfer coefficient (U_{oth}) decrease. If the inlet temperature of oil kept constant then overall theoretical overall heat transfer coefficient (U_{oth}) decreases. Temperature of oil from 65°C to 75 °C the overall theoretical overall heat transfer coefficient (U_{oth}) increases and then above 75°C it decreases.

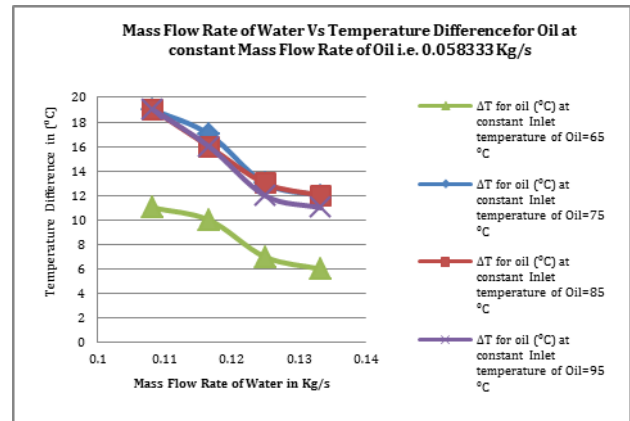


Fig:-5.12 Mass Flow Rate of Water Vs Temperature Difference for Oil at constant Mass Flow Rate of Oil i.e. 0.058333 Kg/s

Figure 5.11 indicates the graphical representation of Mass Flow Rate of Water Vs Temperature Difference for Oil at constant Mass Flow Rate of Oil i.e. 0.058333 Kg/s. As the mass flow rate of water increases from 0.108333 Kg/s to 0.133333 Kg/s then temperature difference of oil decreases. Also as the inlet temperature of oil increases from 65 °C to 95 °C then the temperature difference of oil decreases at constant Mass Flow Rate of Oil and at constant Mass Flow Rate of water.

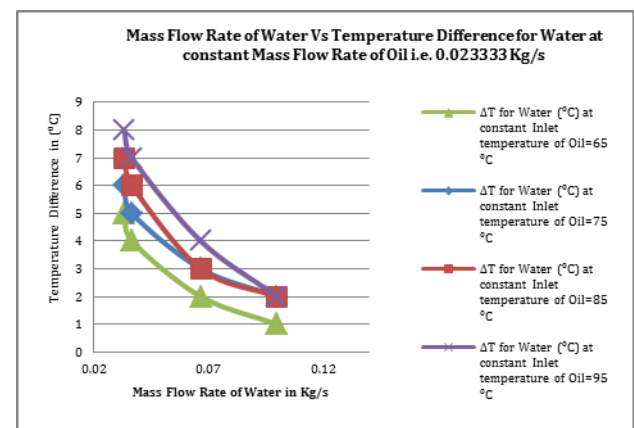


Fig:-5.13 Mass Flow Rate of Water Vs Temperature Difference for water at constant Mass Flow Rate of Oil i.e. 0.023333 Kg/s

Figure 5.13 indicates the graphical representation of Mass Flow Rate of Water Vs Temperature Difference for water at constant Mass Flow Rate of Oil i.e. 0.023333 Kg/s. As the mass flow rate of water increases from 0.033333 Kg/s to

0.10000 Kg/s then temperature difference of water decreases . Also as the inlet temperature of oil increases from 65 °C to 95 °C then the temperature difference of water decreases at constant Mass Flow Rate of Oil and at constant Mass Flow Rate of water.

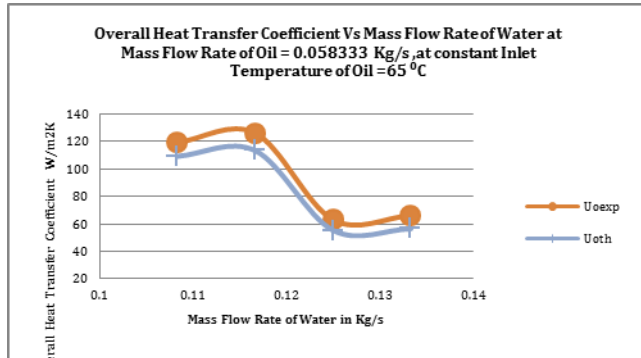


Fig:-5.20 Overall Heat Transfer Coefficient Vs Mass Flow Rate of Water at Mass Flow Rate of Oil = 0.058333 Kg/s, at constant Inlet Temperature of Oil =65 °C

Figure 5.20 indicates the graphical representation of overall heat transfer coefficient vs. mass flow rate of water at mass flow rate of oil = 0.041666 kg/s, at constant inlet temperature of oil =65 °c. As the mass flow rate of water increases from 0.108333 Kg/s to 0.133333 Kg/s then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

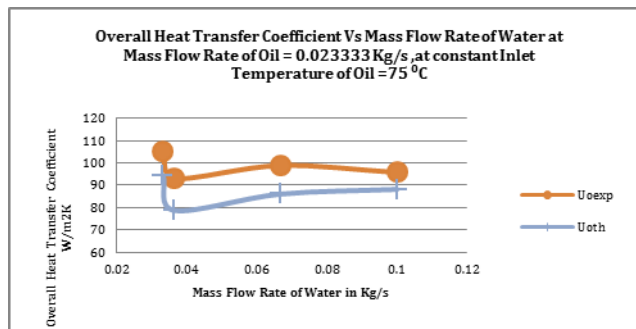


Fig:-5.21 Overall Heat Transfer Coefficient Vs Mass Flow Rate of Water at Mass Flow Rate of Oil = 0.023333 Kg/s, at constant Inlet Temperature of Oil =75 °C

Figure 5.21 indicates the graphical representation of overall heat transfer coefficient vs mass flow rate of water at mass flow rate of oil = 0.023333 kg/s, at constant inlet temperature of oil =75 °c. As the mass flow rate of water increases from 0.033333 Kg/s to 0.10000 Kg/s then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

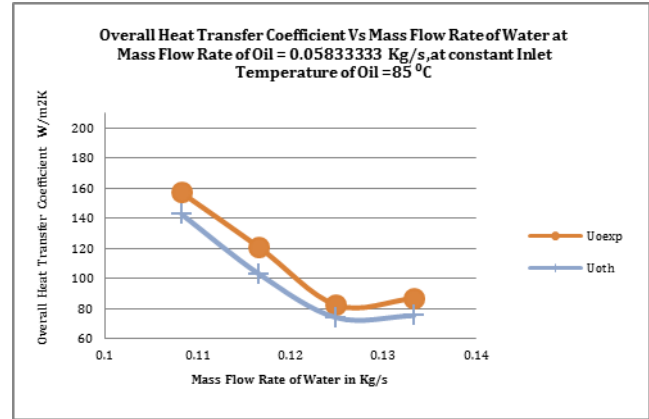


Fig:-5.28 Overall Heat Transfer Coefficient Vs Mass Flow Rate of Water at Mass Flow Rate of Oil = 0.0583333 Kg/s, at constant Inlet Temperature of Oil =85 °C

Figure 5.28 indicates the graphical representation of overall heat transfer coefficient vs mass flow rate of water at mass flow rate of oil = 0.041666 kg/s, at constant inlet temperature of oil =85 °c. As the mass flow rate of water increases from 0.108333 Kg/s to 0.133333 Kg/s then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

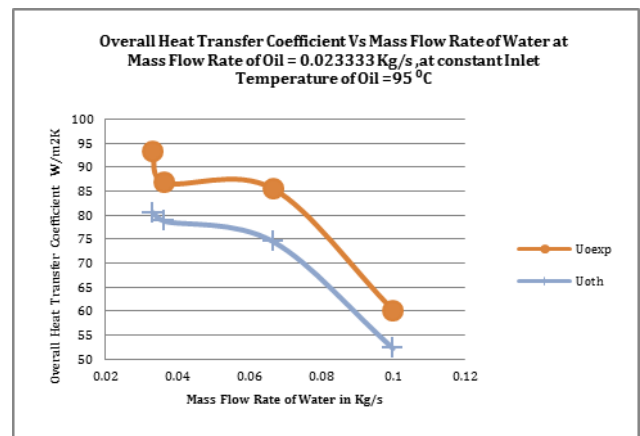


Fig:-5.29 Overall Heat Transfer Coefficient Vs Mass Flow Rate of Water at Mass Flow Rate of Oil = 0.023333 Kg/s, at constant Inlet Temperature of Oil =95 °C

Figure 5.29 indicates as the mass flow rate of water increases from 0.033333 Kg/s to 0.10000 Kg/s then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

VI CONCLUSION

Methodology was developed to carry out design calculation for optimum design of the shell and tube heat exchanger for heat recovery from hydraulic oil. In order to validate the design, small 1-2 Shell and tube heat exchanger was fabricated and tested for its performance. Recover heat from hydraulic oil was used for water heating. During these reading taken the temperature of oil used 65°C, 75 °C, 85 °C and 95°C. We were used different mass flow rate of oil such as 0.0333, 0.036667, 0.066667 and 0.133333 Kg/sec. For each

mass rate of oil we are taken four mass flow rates water and taken 16 trails. Total 64 tests were conducted on hydraulic power pack to check the accuracy of the design methodologies developed for shell and tube heat exchanger.

Temperature of oil from 65°C to 75 °C the overall theoretical overall heat transfer coefficient (U_{oth}) increases and then above 75 °C it decreases. As the temperature of the oil increases theoretical overall heat transfer coefficient (U_{oth}) decreases gradually. The percentage deviation in overall heat transfer coefficients obtained by using correlations and using experimental results. As the mass flow rate of water increases then both Theoretical and experimental overall heat transfer coefficient decreases and throughout percentage deviation is same for all mass flow rates of water.

During experimental study , the overall heat transfer coefficient i.e. 205.19 W/m²K and heat extracted i.e.1814.37 W is maximum at 0.058333 Kg/s mass flow rate of oil, 0.108333 Kg/s mass flow rate of water and 75 °C inlet temperature of oil. Also maximum temperature difference for water i.e.8 °C is at 0.023333 Kg/s mass flow rate of oil, 0.033333 Kg/s mass flow rate of water and 95 °C inlet temperature of oil.

If the mass flow rate of water increases then temperature difference of water decreases. Also as the inlet temperature of oil increases from 65 °C to 95 °C then the temperature difference of water decreases at constant Mass Flow Rate of Oil and at constant Mass Flow Rate of water.

The total manufacturing cost of heat exchanger is Rs. 22700. The value indicates that it is economical to use shell and tube heat exchanger for heat recovery from hydraulic oil.

Also the result of the study ends up with the final conclusion that the use of the mathematical 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.

The methodologies developed for design of shell and tube heat exchanger is reasonably accurate for carrying out studies.

Shell and tube heat exchanger is economical and technically viable for waste heat recovery application.

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