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Effective Design Of Shell And Tube Condenser In A CHP Using TEMA Code-A Case Study

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Abstract: Heat exchangers are the thermal devices responsible for effective heat transfer amidst two fluids with distinct temperatures .Shell and tube heat exchangers (STHX) are most popularly known in thermal power plant, process industries and in different chemical processes .It is very much important for the practicing engineers to understand its thermal analysis ,design and construction .The STHX is quite suitable at high pressure and high temperature .The present research explains a methodological approach for thermal and mechanical design of a steam condenser used in a thermal power plant .The thermal design includes the mathematical determination of mean temperature difference ,no of tubes ,overall heat transfer coefficient ,heat transfer area and the mechanical design involves the computation of shell parameters, baffle design and its configuration. The design procedure follows TEMA .A suitable algorithm has been developed for thermal analysis, mechanical design and computing different performance parameters of heat exchanger.

Index Terms - STHX, TEMA, MATLAB, FLOWCHART, BAFFLES, OHTC

1. INTRODUCTION

The condenser has a wide application in thermal power plant, refrigeration engineering and air conditioning engineering .The researchers around the globe are focusing their attention to make the condenser more compact with better performance .This requires proper understanding of the concept of heat exchanger .

Systematic study of flow and temperature inside a shell and tube type has been done by Mohammed et al .In their analysis ansys is used to compare the performance at different baffle configuration [1].A review on different expertise studies on the design performance and simulation of heat transfer performed to provide minimum heat transfer area for particular heat load as described by Acharya et al. In his method of analysis, he has used Ansys software [2]. The thermal analysis of the STHX with different cross section of the tube performed by various researchers has been reviewed by Gupta et al .In their analysis they have used Ansys software and they have found when circular cross section is replaced by the flat surface enclosure enhances the heat transfer rate [3] . A safest design of least corrosion and safer surface temperature of a STHX has been done by Shukla et al .In his design he has assumed the SS-304 as tube and shell material [4] .An experimental study involving parametric investigation in a pipe heat exchanger has been performed by Saraiya et al .In their

analysis the different characteristics of the heat exchanger at distinct mass flow rates has been addressed [5].Different performance parameters of STHX were evaluated considering different fluids and compared in the area of waste recovery applications by Nag raj Et al .In his study he carried out Cfd analysis with and without fins using Ansys 13.0 [6].A validated program of UHX of ASME Sec VIII Div 1 with TEMA standards has been proposed by Shirode Et.al. He found that FE analysis is closer to exact solution [7]. Kern method was used for designing the heat exchanger and different graphs were plotted for fluid combinations by Shukla et.al .In his study temperatures of inside the shell and cold fluid are considered as parameters with a fixed arrangement of tube bundle and heat transfer analysis was done [8].Thermal analysis of condenser type was done by Sahu et al in which the helix angle of the baffle was varied from 0 to 200 degree and found that there was a increase in overall heat transfer coefficient against the pressure drop used in condensers[9].Condenser type was designed by Vijay et al with temperature distribution and stress analysis using CFD as a tool and validated the data collected from the source based on TEMA and ASME standards[10].Theory of condenser type with curved tubes along with nanofluids was proposed by Fredric et al .They found the coefficient of nanofluids had more value than the base fluid at the identical mass flow rate and inlet temperature due to increase of surface area [11].Cotton wool and tape have given the highest values for effectiveness by Dubey et al .He found that insulation increased the rate of heat transfer if used below critical level of thickness and there is no direct relation between turbulence and effectiveness of heat exchanger [12].Design of the condenser type with design considerations was done by Patel et al .HTRI software was used to validate manually calculated result [13].Optimization of the shell and tube heat exchanger was done by Bhatt et al . In this study tube metallurgy and baffle spacing were changed and the effect on coefficient of heat transferwas considered [14].Optimization of the outlet temperature of the cold fluid was done by Bogale et al .Optimization of a part of geometry was done , investigation of natural frequency and vortex shedding was also conducted [15].

In the present research, the steam condenser of a mega thermal power plant has been properly analyzed and designed and its performance has been computed and compared with the performance of the said industry .In the design process TEMA and ASME Sec VIII div has been used to make the design process safe and acceptable by the industry.

2. SHELL AND TUBE HEAT EXCHANGER

A shell and tube type heat exchanger consists of a shell and a bundle of tubes in which one of the fluid flows through tube while another fluid flows over the tubes inside the shell for transfer of heat between two fluids. The tubes may be straight or bent in U shape called U tubes .The simplicity of this model makes it an ideal cooling solution for most of the applications, in engine, transmission lines and are easy to maintain, particularly with models where floating tube is available.

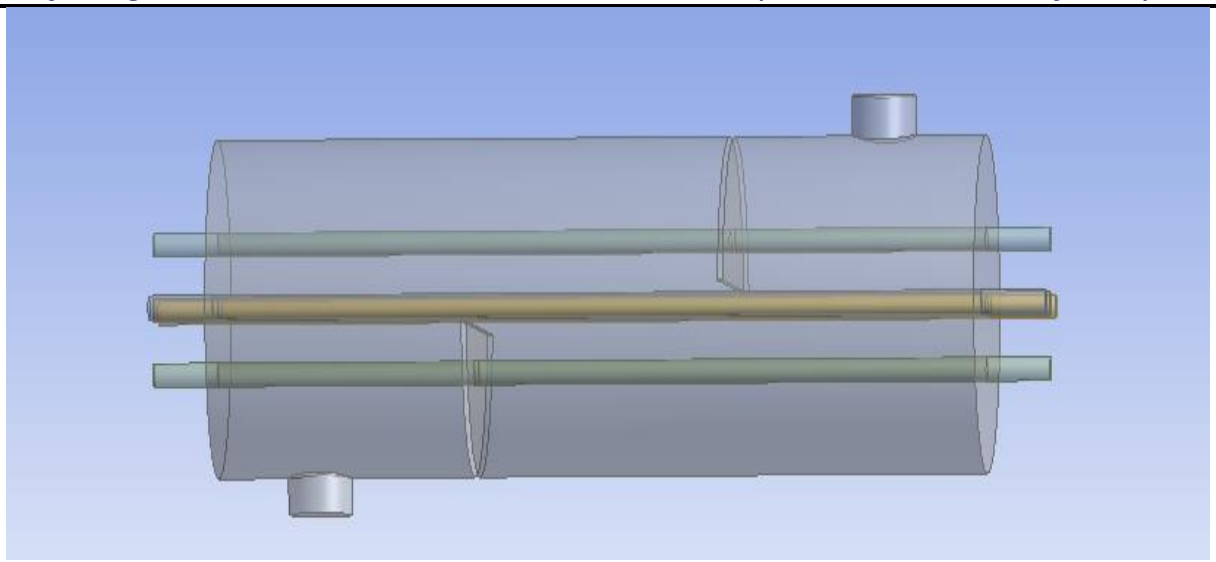


Fig 2.1 Schematic diagram of Shell and Tube Heat Exchanger

2.1Thermodynamic Analysis of STHX

2.1.1 Industrial data input:-

Inlet temperature of hot fluid - 49.06 ^{O}C
Outlet temperature hot fluid - 49.06 ^{O}C
Inlet temperature of cold fluid - 35^{O}C
Outlet temperature of cold fluid - 43^{O}C
Hot fluid fouling factor -0.00009 m^{2}K/W
Cold fluid fouling factor -0.00018 m^{2}K/W
P_{inlet} (for hot fluid) -5.5 bar
P_{inlet} (for cold fluid) -0.12 bar
ΔP_t -6 psi
ΔP_s -4 psi
Mass flow rate of cold fluid - 4200 × 10^{3}kg/hr

2.1.2 Assumptions:-

1. The heat exchanger parameters are not a function of time.
2. Both fluids are considered to be flowing at the same rate.
3. The properties of fluid such as temperature and velocity are considered to be constant.
4. Difference in kinetic energy and potential energy are considered to be negligible.
5. Specific heat of a fluid generally changes with temperature but within a specified temperature range it can be assumed to be constant with some average value with little loss in accuracy.
6. The outside is perfectly insulated.
7. Axial heat conduction is neglected.

Nomenclature:-

Sym	Name	Sym	Name
$Q^{\cdot}$	Heat transfer rate	d	Equivalent diameter
U_m	Heat transfer coefficient	P	Internal Design Pressure
A_h	Area required for heat to be transferred	K_e	Joint efficiency longitudinal joint
ΔT_m	Logarithmic average heat transfer rate	S_a	Allowable stress
C_t	Specific heat of tube side fluid	n_j	Joint Efficiency
C_s	Specific heat of shell side fluid	t_e	Element thickness
$m^{\cdot}_t$	mass flow rate of tube side fluid	t_s	Tube sheet thickness
$m^{\cdot}_s$	mass flow rate of shell side fluid	K_f	Constant Factor
T_i	Inlet temperature of shell side fluid	m''_t	Mass velocity of tube
T_o	Outlet temperature of shell side fluid	m''_s	Shell –side mass velocity
t_i	Inlet temperature of tube side fluid	v	Velocity inside the shell
t_o	Outlet temperature of tube side fluid	μ_s	Dynamic Viscosity of steam
$\Delta T_m'$	True Temperature Difference	K_s	Thermal conductivity of steam
F	Correction factor	K_w	Thermal conductivity of water
R	Heat Capacity Ratio	j_h	Colburn factor
E	Thermal Effectiveness of tube side fluid	n_p	Number of passes
D_o	Outside diameter	P_{ex}	External pressure
D_i	Internal diameter	n_{cj}	Joint efficiency circumferential joint
P	Tube pitch	F_s	Fouling factor of steam
h_T	Convective coefficient on tube side	F_t	Fouling factor of water
h_S	Convective coefficient on shell side	f_t	Tube side friction factor
n	Number of tubes	f_s	Shell side friction factor
Re	Reynolds number	g	Acceleration due to gravity
l	Tube Length	RD	Specific gravity
D_B	Bundle diameter	A_d	Design heat transfer area
D	Shell Id	A_r	Total heating area required
b	Baffle spacing	Nu	Nusselt number
A	Shell side area	ΔP_t	Pressure Drop in tube side
Pr	Prandtl number	C_t	Clearance between tubes
u	Velocity inside the tube	μ_w	Dynamic viscosity of water

2.1.3 Step by Step design:-

1. Thermo properties of fluids are obtained at average temperature.
2. Energy balance equation is carried out and heat transferred by the heat exchanger is calculated.
   $Q^{\cdot}=m_t^{\cdot}C_t(t_o-t_i)=m_s^{\cdot}C_s(T_i-T_o)$
3. U value is arbitrarily taken.
4. No of shell and tube passes is decided, LMTD and correction factor are determined.
   $\Delta T_m= (T_i-t_o)-(T_o-t_i)/\ln (T_i-t_o/T_o-t_i)$
   $F = (\bar{R}+1)^{0.5}\ln (1-E/1-E\bar{R})/ (\bar{R}-1) \ln (2-E (\bar{R}+1-(\bar{R}^2+1)^{0.5}/2-E (\bar{R}+1+ (\bar{R}+1)^{0.5})$
   Where $\bar{R}=\frac{T_i-T_o}{t_o-t_i}$ and $E=\frac{t_o-t_i}{T_i-t_i}$
   $\Delta T_m' =F\times \Delta T_m$
5. Area for the requisite heat transfer is calculated.
   $A_h=Q^{\cdot}/U_m\Delta T_m'$
6. Tube material, tube diameter, its wall thickness and tube length of the tube l is selected. The tubes required is calculated
   $n=A_h/\Pi D_o l$
7. Velocity in tube side is calculated by
   $u=4m^{\cdot}_t(n_p/n)/ \Pi\rho d_i^2$
   If u <1 m/sec, then no of passes is fixed so that Re>=10^4

8. Type of shell and tube heat exchanger is selected (fixed tube sheet, U-tube) .The tube pitch and shell diameter is selected that can allow all the tubes.

   $P=1.25\ D_o$ $D= D_B+BDC$ $D_B=D_o(n/K_1)^{1/n_1}$

9. Baffle type (segmental, doughnut etc.), size (percentage cut), spacing and number is selected.

   $b=0.4\ D$

10. Hot side and cold side coefficient is calculated according to the Re number.

    Hot side and cold side heat transfer dirt factor is selected.

11. Coefficient is calculated according to outer surface

    $\bar{U}_{o,cal}=(1/h_s+R_s+A_o/A_i(d_o-d_i/2K_w)+A_o/A_i(1/h_t)+A_o/A_i\times R_t)^{-1}$

12. If $0<\frac{\bar{U}_{o,cal}-\bar{U}_{o,assumed}}{\bar{U}_{assumed}}<30$ % then next step is followed other heat transfer area is calculated using $U_{o,cal}$. If the calculated heat transfer coefficient on the shell side is too low then baffle spacing is assumed to be 0.2 D and shell side heat transfer coefficient is again calculated.

13. Percentage of overdesign is calculated.

    % overdesign $=\frac{A_d-A_r}{A_r}$

14. Cold side pressure drop is calculated

    i. Pressure drop in the straight section of the tube (frictional loss)

    $\Delta P_t=f_t\times G_t^2\times l\times n_p/2\times g\times\rho\times\varphi_t\times D_i$

    ii. Return loss due to change of direction of fluid in multi pass exchanger

    $\Delta P_r=4n_p\times u^2\times 62.5\ /RD\times 2g\times 144$

    Total tube side pressure drop $(\Delta P_T)=\Delta P_t+\Delta P_r$

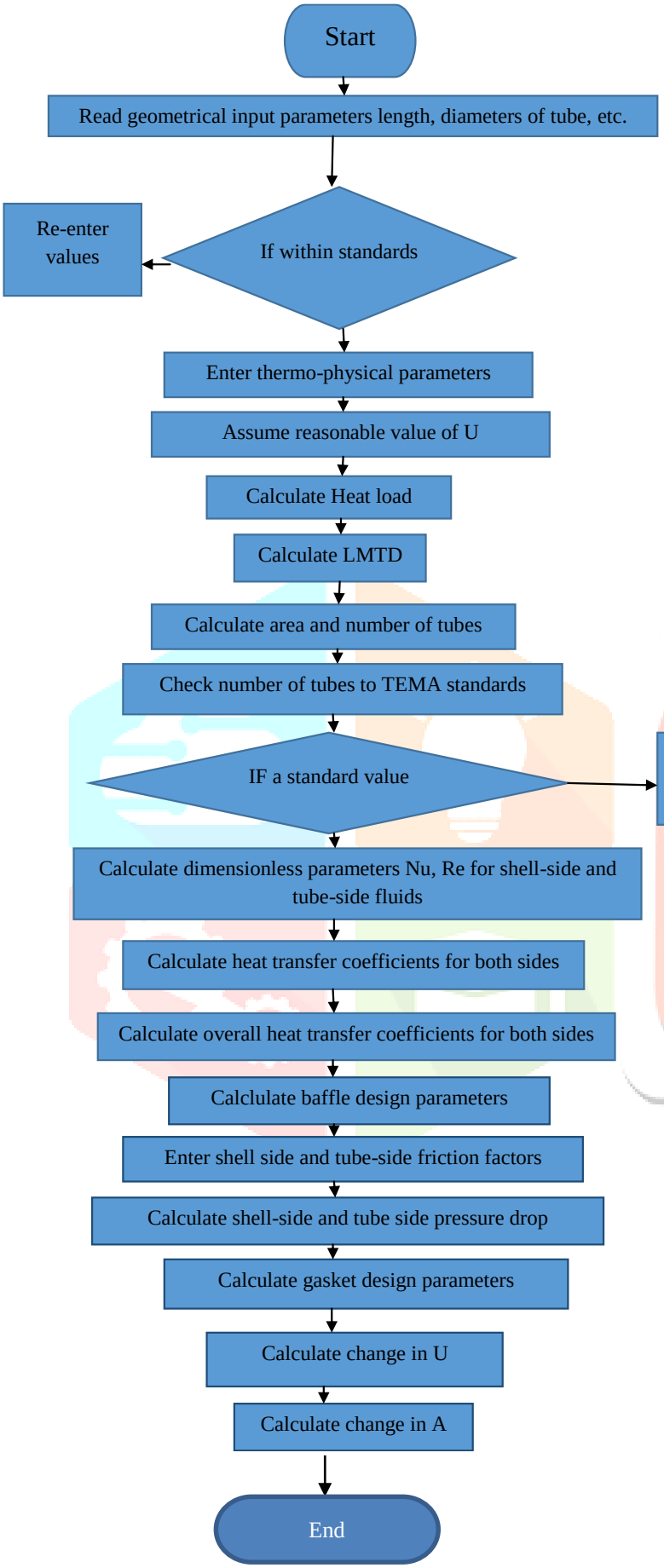
15. Hot side pressure drop is calculated

    a. Pressure drop for flow across the tube bundle (frictional loss)

    $\Delta P_s= f\times G_s^2\times D^2(n_b+1)/7.5\times 10^{12}\times D_e\times S_k\times\varphi_k$

    b. Return loss due to change of direction of fluid=0 , Total shell side pressure drop $(\Delta P_S)=\Delta P_s$

2.1.4 FLOWCHART FOR THERMAL ANALYSIS:-



2.2 Design Analysis of STHX:-

2.2.1 Main Shell Design

Industrial data input:-

Inside pressure – 1.1 bar
Outside pressure – 3.5 kg/cm^2
Shell outside diameter -3000 mm
Efficiency of longitudinal joint - 0.75
Efficiency of circumferential joint -0.85
Stress allowed -49 MPa
Thickness – 12 mm
Efficiency of outer longitudinal joint -0.85
Outside diameter of shell with allowance- 3003 mm

2.2.2 Step by step design:-

Design under inside pressure

1. Thickness required:-
   $tr = P_s \times D \times K/(2 \times S \times E - 0.2 \times Ps)$
2. Maximum pressure allowed :-
   $= (2 \times S \times E \times t)/ (K \times D + 0.2 \times t)$
3. Maximum pressure required for new and cold
   $= 2 \times S \times E \times t/K \times D + 0.2 \times t$
4. Actual stress
   $= (P_s \times R)/ (S \times E - 0.6 \times P_s)$
5. Maximum working pressure for straight flange
   $= (S \times E \times t)/(R + 0.6 \times t)$

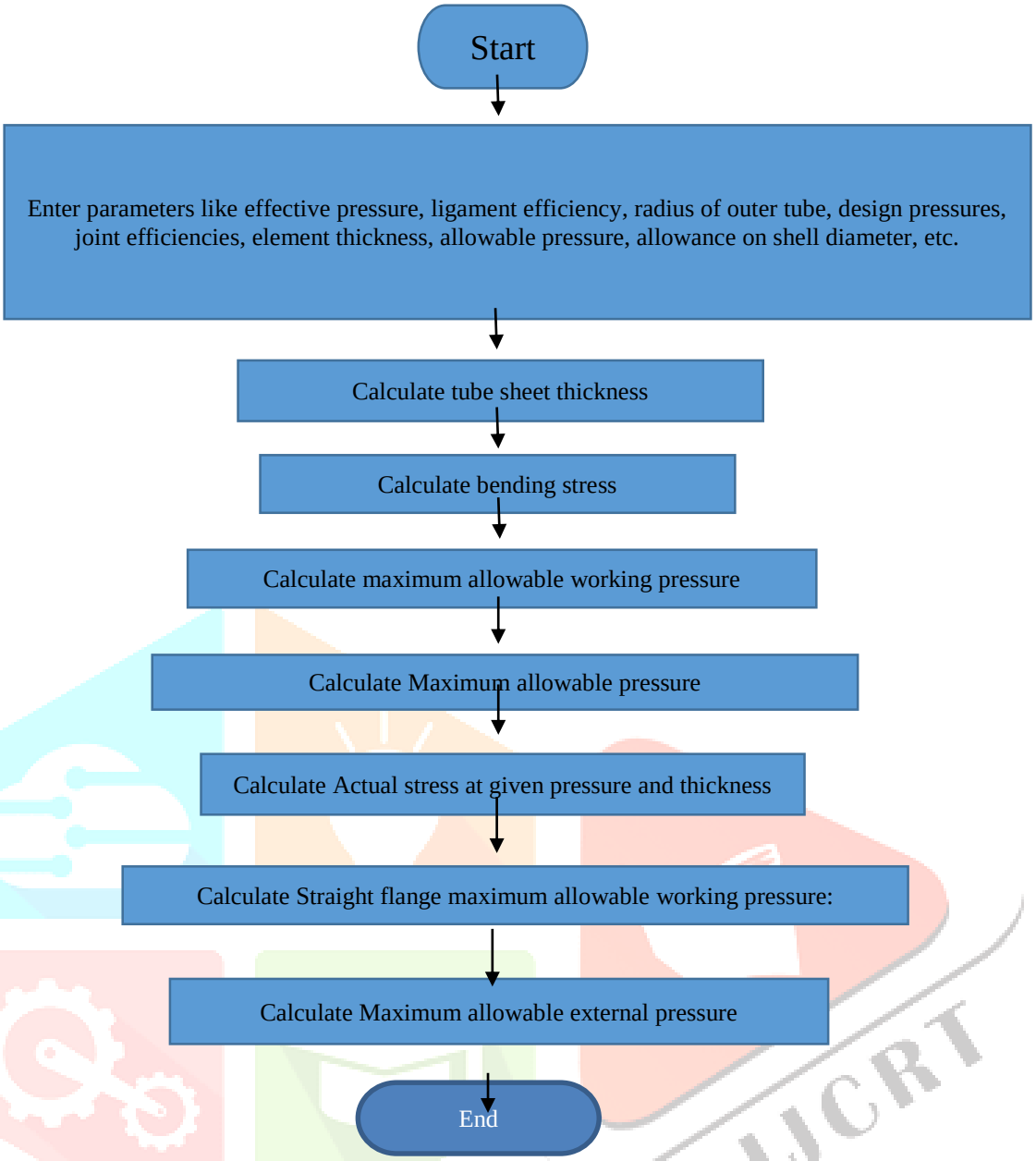
Design under external pressure

Maximum allowable external pressure $= MAEP = B/ (K_o \times D_{so}/t)$

2.2.4 Gasket Design

The minimum width of peripheral ring gaskets for external joints shall be 3/8'' (9.5 mm) for shell sizes through 23'' (584 mm) ID and 1/2'' (12.7 mm) for all large sizes.

2.2.5 FLOWCHART FOR MECHANICAL DESIGN:-



3. RESULTS AND DISCUSSION

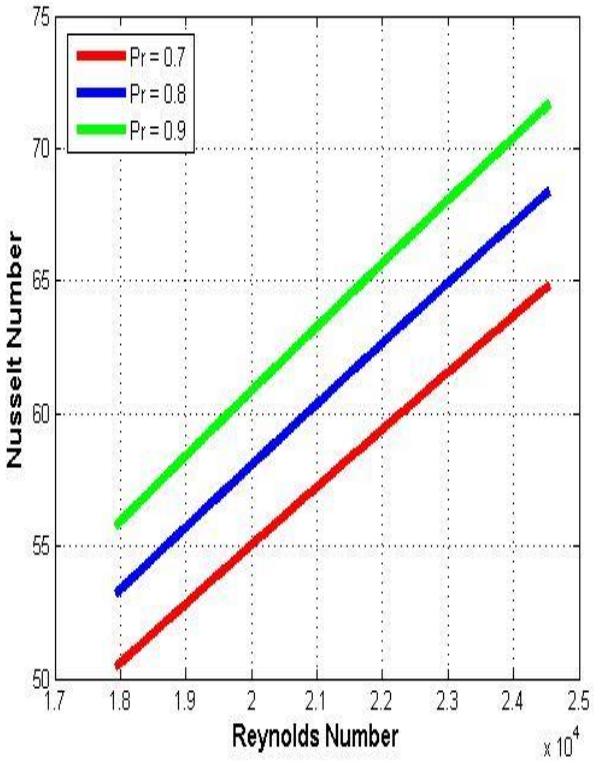


Fig 3.1 Nusselt number variation with Reynolds number at different Prandtl number.

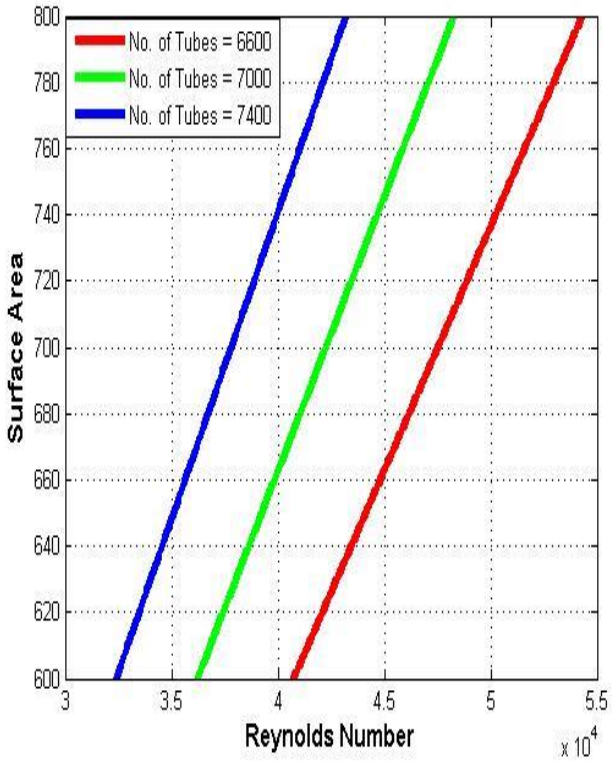


Fig 3.2 Surface area variation with Reynolds number at different length of tubes.

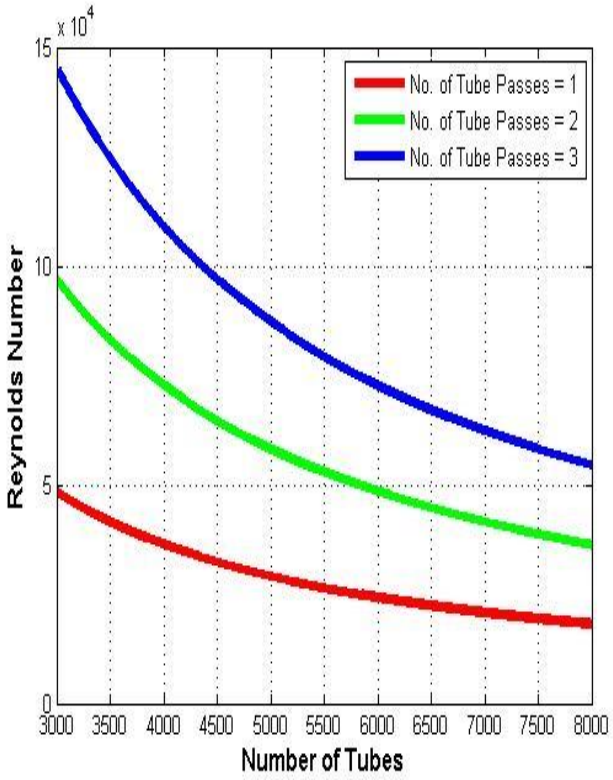


Fig 3.3 Reynolds number variation with number of tubes at different passes of tube.

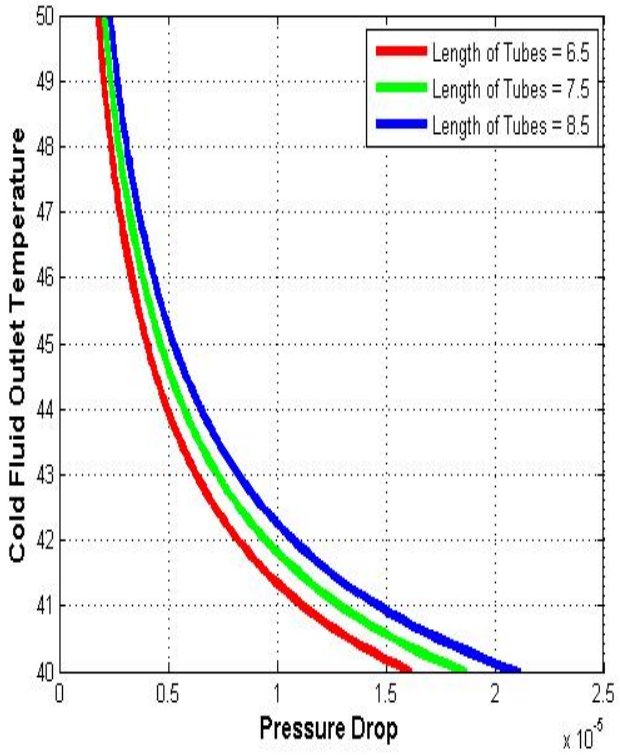


Fig 3.4 Cold fluid outlet temperature variation with pressure drop at different length of tubes.

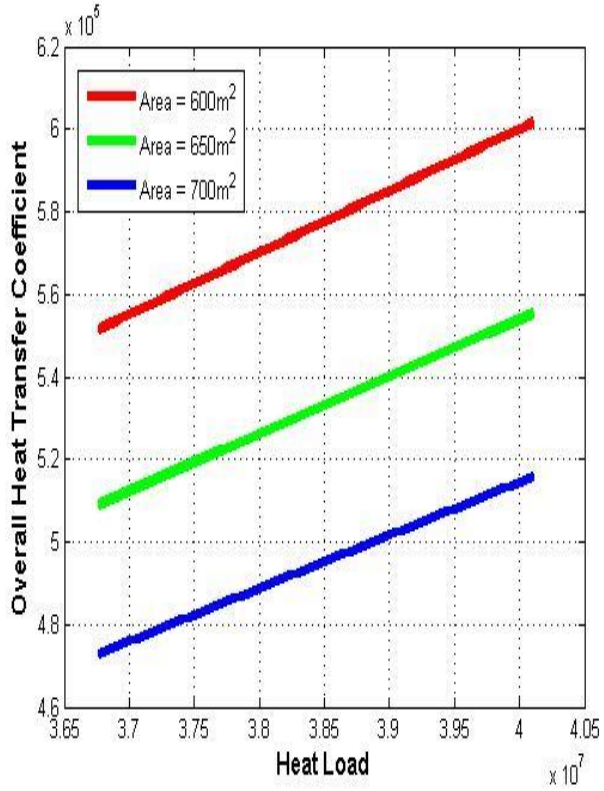


Fig 3.5 Overall heat transfer coefficient variation with heat load at different area.

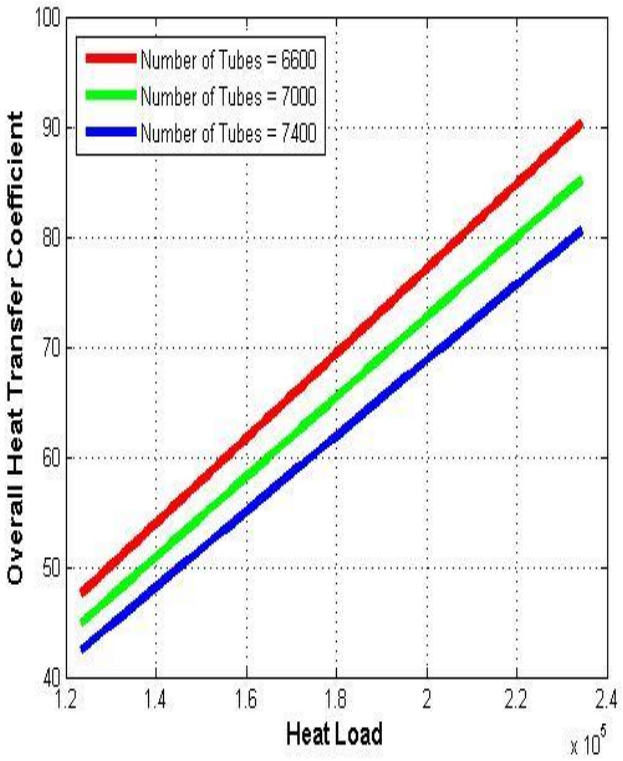


Fig 3.6 overall heat transfer coefficient variation with heat load at different number of tubes.

Fig 3.1 depicts the deviation of Nusselt number with Reynolds number at different Prandtl number .It is seen from the graph that with increase in Reynolds number, the Nusselt number increases linearly. At a particular Nusselt number, Prandtl number increases with increase in Reynolds number. At a particular Reynolds number with increase in Prandtl number Nusselt number increases.

Fig 3.2 represents the change in Reynolds number with surface area at different lengths of tube. It is found that with increase in surface area of the heat exchanger, the Reynolds number decreases.

Fig 3.3 shows the alteration of Reynolds number with number of tubes for different passes of tube. It is observed that Reynolds number decreases with increase in number of tubes.

Fig 3.4 expresses the distinction of cold fluid inlet temperature with pressure drop inside the tube at different lengths of tube. It is viewed that for a particular length of the heat exchanger with the increase in the pressure drop inside the water tubes the cold fluid outlet temperature decreases .Also it is clear from the graph that with increase in length of the heat exchanger the pressure drop increases with no effect on the cold fluid outlet temperature .This figure also states that at a particular pressure drop the increase in the length of the heat exchanger increases the cold fluid outlet temperature.

Fig 3.5 identified the unconformity of overall heat transfer coefficient with heat load at different area. It can be viewed that with increase in heat load the overall heat transfer coefficient increases. In addition, this figure suggests that at a particular heat load, with increase in surface area the overall heat transfer coefficient increases.

Fig 3.6 recognized the fluctuation of overall heat transfer coefficient with heat load at different number of tubes .It is found that heat transfer coefficient increases with increase in heat load. Also it is seen that overall heat transfer coefficient remains constant with increase in number of tubes and with increase in heat load, surface area increases.

4. CONCLUSION:-

Thorough thermal and mechanical analysis of industrial data was concluded with the MATLAB code for each, which strictly followed TEMA code and different performance parameters of condenser type heat exchanger were studied in the form of graphs .Thus any shell and tube heat exchanger can be designed from the above code.

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