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Numerical Analysis Of A Surface Condenser Design

Prachi Bhatnagar
M.Tech. Scholar
Department of Mechanical Engineering LNCT, Bhopal (MP), India

Dr. V. N. Bartaria
Professor & Head
Department of Mechanical Engineering LNCT, Bhopal (MP), India

Abstract:

Power plant condenser (a huge complex shell-and-tube heat exchanger) is one of the most important auxiliary equipments in a power plant. The thermal efficiency of a power plant can be increased by enhancing the heat transfer rate of a condenser, this can be achieved by reducing the turbine exhaust pressure. Considering this, the designing of surface condenser is an important aspect towards making thermal power plant efficient. In our present work, the author has considered a conventional design procedure and empirical correlations of surface condenser for the analysis of critical parameters like pressure drop, dirt factor, overall heat transfer coefficient, velocity of circulating water, rate of condensation, outlet temperature of water, number of tubes, tube pitch, outer diameter of tube, weight flow of steam and water, etc. These parameters directly affect the performance of a condenser or the unit where this condenser is employed. The results are calculated by the programme made in computer codes and are represented graphically.

Keywords: *Surface Condenser, Condenser Design, Performance Analysis, Heat Exchanger.*

Introduction

Steam surface condensers are the most commonly used condensers in modern power plants. The exhaust steam from the turbine flows on the shellside (under vacuum) of the condenser, while the plant's circulating water flows in the tubeside. The source of the circulating water can be either a closed-loop (i.e. cooling tower, spray pond, etc.) or once through (i.e. from a lake, ocean, or river). The condensed steam from the turbine, called condensate, is collected in the bottom of the condenser, which is called a hotwell. The condensate is then pumped back to the steam generator to repeat the cycle. The main heat transfer mechanisms in a surface condenser are the condensing of saturated steam on the outside of the tubes and the heating of the circulating water inside the tubes. Thus for a given circulating water flow rate, the water inlet temperature to the condenser determines the operating pressure of the condenser. As this temperature is decreased, the condenser pressure will also decrease. As described above, this decrease in the pressure will increase the plant output and efficiency.

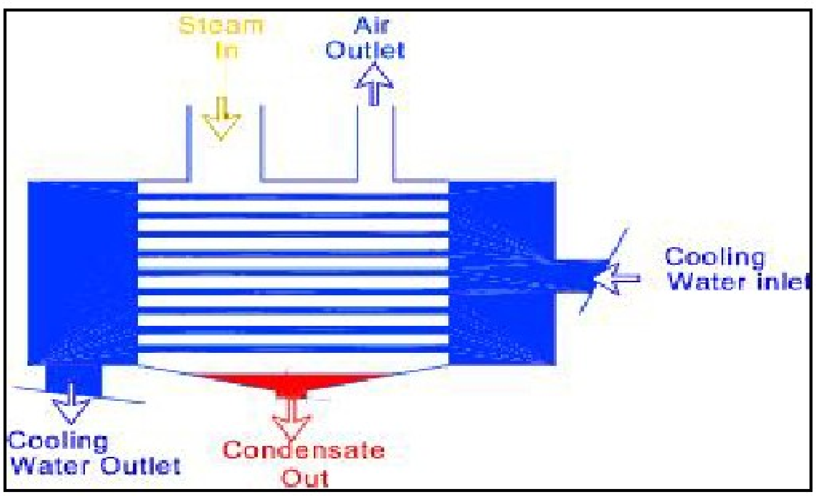


Figure 1: Surface condenser

Due to the fact that a surface condenser operates under vacuum, non condensable gases will migrate towards the condenser. The non condensable gases consist of mostly air that has leaked into the cycle from components that are operating below atmospheric pressure (like the condenser). These gases can also result from caused by the decomposition of water into oxygen and hydrogen by thermal or chemical reactions. These gases must be vented from the condenser otherwise the gases will increase the operating pressure of the condenser. Since the total pressure of the condenser will be the sum of partial pressures of the steam and the gases, as more gas is leaked into the system, the condenser pressure will rise. This rise in pressure will decrease the turbine output and efficiency. The gases will blanket the outer surface of the tubes. This will severely decrease the heat transfer of the steam to the circulating water. Again, the pressure in the condenser will increase The corrosiveness of the condensate in the condenser increases as the oxygen content increases. Oxygen causes corrosion, mostly in the steam generator. Thus, these gases must be removed in order to extend the life of cycle components. The two main devices that are used to vent the non condensable gases are Steam Jet Air Ejectors and Liquid Ring Vacuum Pumps. Steam Jet Air Ejectors (SJAE) use high-pressure motive steam to evacuate the non condensable from the condenser (Jet Pump). Liquid Ring Vacuum Pumps use a liquid compressant to compress the evacuated non -condensable and then discharges them to the atmosphere.

In cross flow condensers, fluids flow perpendicular to each other. Cooling water flows through the tubes and steam flows around these tubes at an angle of 90°. Basically a single type of steam condenser is not suitable in most of the thermal power plants, rather a combination of two or all the types among parallel, counter and cross flow is preferred. The reason for the combination of the various types is to maximize the efficiency of the heat exchanger within the restrictions placed on the design. That is, size, cost, weight, required efficiency, type of fluids, operating pressures, and temperatures. All these factors help in determining the complexity of a specific heat exchanger.

Design Parameters & Empirical Correlation

Heat flow in the condenser corresponding to the tube side (water) as well as shell side (steam) is equal because for the condensation of steam same amount of heat in the form of latent heat is rejected to the water, so heat gained is equal to the heat lost and heat flow in the condenser is a function of overall design heat transfer coefficient, heat transfer surface and LMTD (ΔT).

For the heat balance, $Q = W\ h_{fg} = w\ c_p\ (T_2 - T_1) = U_D\ A\ \Delta T$, (Btu/hr)

LMTD (log mean temperature difference), $\Delta T = (T_2 - T_1) / \log((T_s - T_1) / (T_s - T_2))$, (°F)

Number of tubes calculated by the formula or for the exact value of N_t **corresponding to the situation see in the HEI standards.** $N_t = A / (L\ a''_t)$ **,for the value of** a''_t **referring HEI standards**

Flow area of tube and shell , $a_t = N_t\ a'_t /\ 144$ n, (sq ft) and a_s= (ID) C` B / (144 P_T), (sq ft)

for the value of a'_t & ID referring HEI standards , Baffle spacing, B comes in the rang e of 0.2 ID to ID, Clearance is calculated by the formula C`= P_t - (ODT), (in.)

Mass velocity of water and steam, G_t = w / a_t , (lb/hr sqft) and G_s= W / a_s, (lb/hr-sq ft)

Condensate loading for tubes, G``= W/ L $N_t^{0.66}$, (lb/hr-lin ft), L is length of tube in ft.

Velocity of circulating water is the function of mass velocity of water V = G_t / 3600 * 62.5 (ft per sec)

Average temperature of water, t_a= (t_1 + t_2) /2, (°F)

The value of heat transfer coefficient for water when referred to the tube outside diameter,

h_{io}= h_i (ID)/(OD), (Btu/hr-sq ft-°F) and the average value of the condensing film coefficient or heat transfer coefficient for the outside steam is, h_o and its value can be seen from the HEI standards.

The tube wall temperature, $t_w = t_a + (h_o / (h_{io} + h_o)) (T_s - t_a)$, (°F)

The film temperature, $t_f = (T_s + t_w) / 2$, (°F)

Number of crosses, N + 1 = 12 L / B and the specific gravity for steam, s = 1 / (v_s * 62.5)

The Reynolds Number for shell side fluid i.e. steam side, Re_s = (ODS) G_s / μ

Shell side pressure drop mostly depends on baffle spacing or pressure drop can be controlled by few parameters like number of tubes, length of tubes, tube pitch, but baffles are the most suitable one . ΔP_s= (f G_s^2(IDS)(N+1)) / (2*5.22*10^{10} (ODS) s),(psi)

Total tube side pressure drop is the summation of tube side pressure drop and return pressure drop. Return pressure drop is the pressure drop for return losses.

$\Delta P_t = 0.0067\ (V)^{1.84} / (ID)^{1.16}$, (psi)

$\Delta P_r = (4\ n / s) (V^2 / 2\ g\ \square)$, (psi)

$\Delta P_T = \Delta P_t + \Delta P_r$, (psi)

The clean overall heat transfer coefficient is depends on the cleanliness factor or the overall coefficient which is dirt free. $U_c = h_{io} h_o / (h_{io} + h_o)$, (Btu/hr-sq ft-°F)

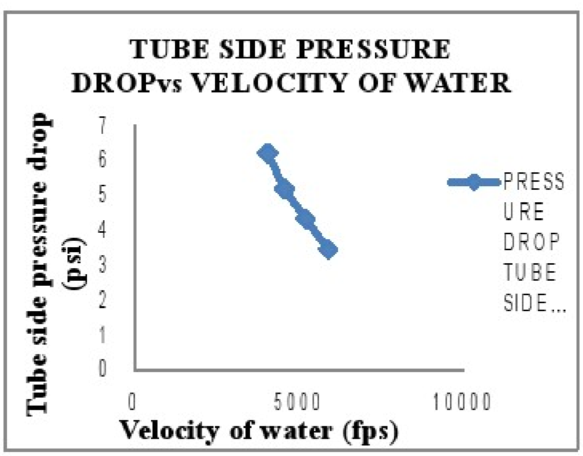
Dirt factors are defined as a percentage of the clean overall coefficients (cleanliness factor).thus a cleanliness factor of 85% means that the design overall coefficientwill be 85% of the clean overall coefficient. $R_d = (U_C - U_D) / U_C\ U_D$, (hr-sq ft- °F / Btu)

Numerical Analysis

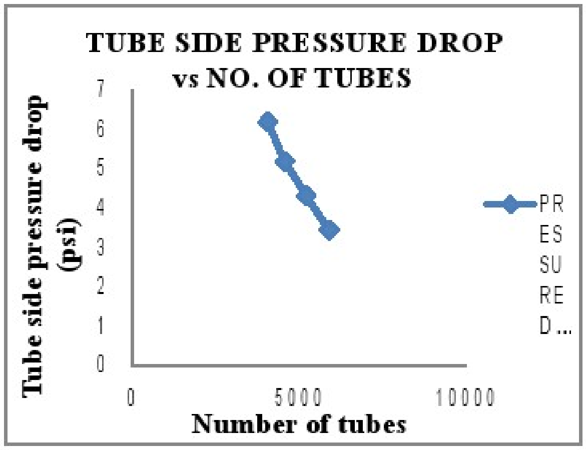
The results are obtained by the above empirical correletions, the values of designing parameters is calculated by the aid of a programme on C++ and run it for different values by assuming few parameters constant, by doing many trials for the satisfactory design and found relationships between many critical parameters which directly affects the pressure drop and then the performance of the plant where condenser is employed. This numerical analysis part of our work is the summary of the designing procedure of surface condenser , give relationship between most important critical parameters i.e. pressure drop , overall heat transfer coefficient, tube geometry, rate of condensation, outlet water temperature, dirt factor, velocity of circulating water, etc.

Result

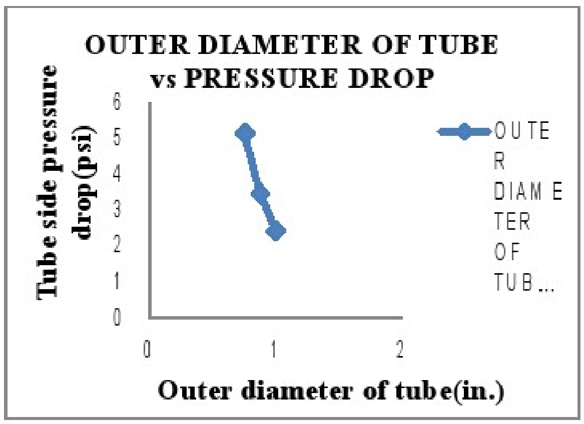
The results are shown graphically, In graph 1,2 and 6 the relationship between tubeside (water) pressure drop - velocity of circulating water, tubeside pressure drop - number of tubes and tube side pressure drop - overall design coefficient shows at constant outer diameter of tube, heat flow, weight flow, temperatures and length of the tube. In graph 3, the relationship between outer diameter of tube and tube side pressure drop obtain for constant overall design coefficient , heat flow, weight flow, temperatures, surface area and length of the tube. In graph 4, the relationship between outlet water temperature and weight flow of steam for constant weight flow of water and inlet temperature of water and varying outlet temperature of water. In graph 5, the relationship between inner diameter of shell and tube pitch for the constant values of number of passes and number of tubes are shown.



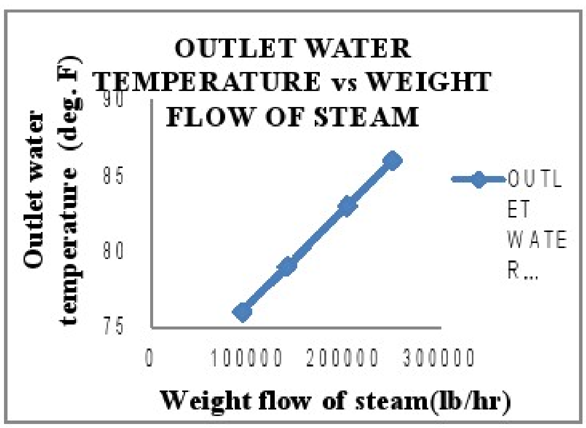
Graph 1



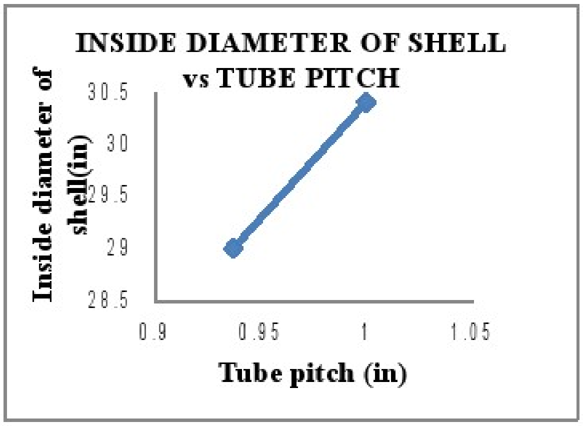
Graph 2



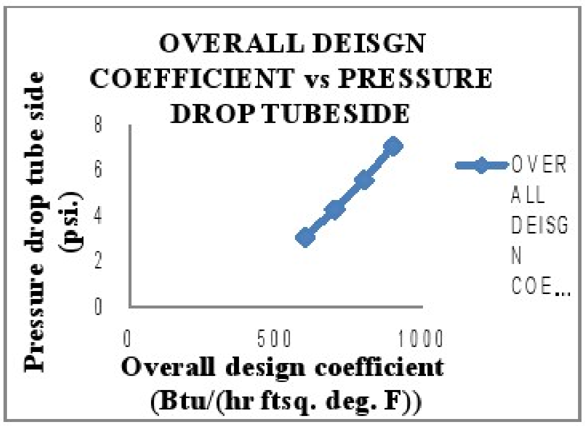
Graph 3



Graph 4



Graph 5



Graph 6

Conclusion

By considering surface condenser design procedure and empirical correlations we have obtain the critical parameters by calculation on the programme made in C++ and relationship in graphical form between them which give true facts for the analysis purpose. The performance of steam surface condenser power plant depends on the performance of condenser or pressure drop from the condenser, so here graphical results are shown above, in graph 1, 2 and 6 the pressure drop reduces by reducing the velocity of circulating water, by reducing the overall design coefficient and by increasing the number of tubes at constant outer diameter of tube. In graph 3, at constant overall design coefficient the pressure drop reduces by increasing the outer diameter of tube. In graph 5, by increasing the value of tube pitch at constant number of passes and constant number of tubes, the inside diameter of shell is also increases. In graph 4 the outlet water temperature at inlet pressure of steam for constant weight flow of water and constant inlet temperature of water, increases by increasing the weight flow of steam.

Nomenclature

Q = Heat flow, Btu/hr

W, w = Weight flow of steam, water, lb/hr

T_s, T_1, T_2 = Temperature of steam, inlet water, outlet water, °F

U_D, U_C = Design, Clean overall heat transfer coefficient, Btu/hr-sq.ft°F

A = Heat transfer surface, sq.ft

ΔT= Log Mean Temperature Difference, °F

c_p= Specific heat of water, Btu/ lb °F

N_t= Number of tubes

L= Length of tube, ft

a''_t= External surface per linear foot, ft

a_t, a_s = Flow area of tube, shell, sq in.

n = Number of tube passes

a'_t= Flow area per tube, sq in.

G_t, G_s = Mass Velocity of water, steam, lb/hr sq ft.

ΔP_T ,ΔP_t , ΔP_r ΔP_s = Total, tube, return, shell pressure drop, respectively, psi

h_{io},h_i,h_o = Heat transfer coefficient corresponding to outer diameter of tube, for inside water,for outside steam, Btu/hr-sq.ft°F

R_d = Dirt factor, hr-sq ft- °F / Btu

C `, B, P_T = Clearance between tube, Baffle spacing, Tube pitch, in.

f , s, v_s = Friction factor, sq. ft. / sq. in. , specific gravity , specific volume of steam, ft^3/ lb

N, N+1 = Number of shell side baffles, number of crosses.

Re_s, μ = Reynolds number of steam, viscosity of steam lb/ft-hr

G`` = Condensate loading for tubes, lb/hr-ft

t_f, t_a,t_w = Film temperature, Average temperature of water, Tube wall temperature, °F

ODT, IDT & ODS, IDS = Outer, inner diameter of tube & Outer, inner diameter of shell.

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