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Surface Condenser Design – A Review

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Abstract: *Surface condenser is also known as power plant condenser and is one of the most important auxiliary equipment 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 to have a design review of surface condenser with all the critical parameters which directly or indirectly affects the condenser performance. The author also focuses the effects of key parameters on surface condenser performance.*

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

Introduction

The most efficient thus most popular condensing systems are the water cooled surface condenser, popular in areas where a large amount of cooling water is available. Surface condenser is a shell & tube type indirect contact heat exchanger in which steam flows inside the shell & cooling water flows inside the tubes, the hot fluid must be steam & the cold fluid must be water. 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.

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.

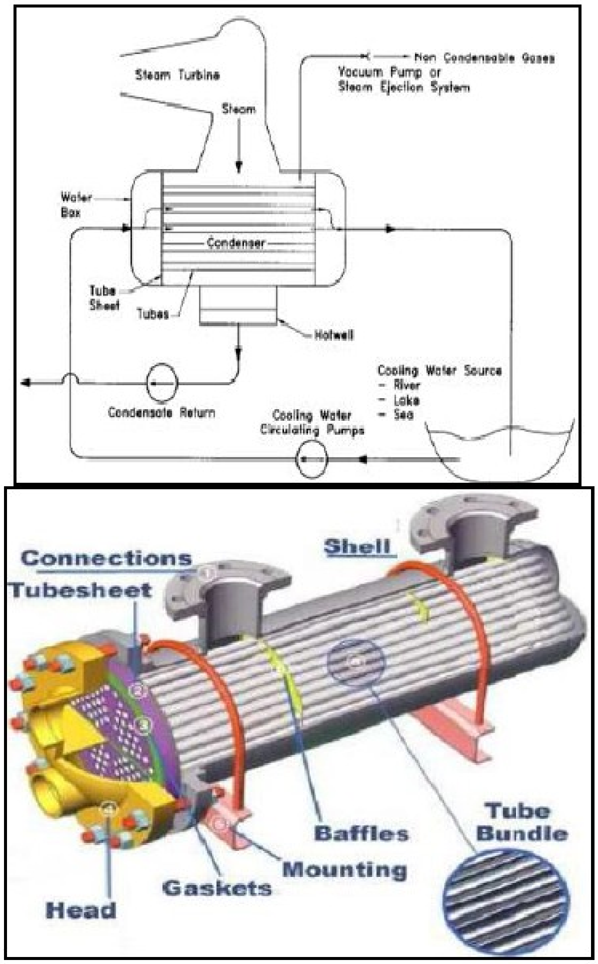

Figure 1: Surface Condenser

Components of Surface Condenser

Tube

Heat-exchanger tubes are also referred to as condenser tubes and should not be confused with steel pipes or other types of pipes which are extruded to iron pipe sizes. The outside diameter of heat exchanger or condenser tubes is the actual outside diameter in inches within a very strict tolerance. Heat-exchanger tubes are available in variety of metals which includes steel, copper, admiralty, muntz metal, brass, 70-30 copper-nickel, aluminum bronze, aluminum and the stainless steel. They are obtainable in a number of different wall thicknesses defined by the Birmingham wire gage, which is usually referred to as the BWG or gage of the tube. The sizes of tubes can be referred from HEI standards. OD is most common in heat-exchanger design.

Shell

Shells are fabricated from steel pipe with nominal IPS diameter up to 12 in. as given. Above 12 and including 24 in. the actual outside diameter and the nominal pipe diameter are the same. The standard wall thickness for shells with inside diameters from 12 to 24 in. inclusive is $^3/_8$ in. which is satisfactory for shell-side operating pressure up to 300 psi. Greater wall thicknesses may be obtained for greater pressures. Shells above 24 in. in diameter are fabricated by rolling steel plate.

Tube Pitch

Tube holes can be drilled very close together, since too small a width of metal between adjacent tubes structurally weakens the tube sheet. The shortest distance between two adjacent tube holes is the clearance or ligament, and these are now fairly standard. Tubes are laid out on either square or triangular patterns as. The advantage of square pitch is that the tubes are accessible for external cleaning and cause a lower pressure drop when fluid flow in the direction indicated in figure. The tube pitch P_T is the shortest center-to-center distance between adjacent tubes. The common pitches for square layouts are $^3/_4$ in. OD on 1-in. square pitch and 1-in. OD on $1\,^1/_4$ -in. square pitch. For triangular layout these are $^3/_4$ in. OD on $^{15}/_{16}$ -in. triangular pitch, $^3/_4$ in. OD on 1-in.triangular pitch, and 1-in. OD on $1\ ^1/_4$ -in. triangular pitch.

Tube-sheet Layouts and Tube Counts

A typical example of the layout of tubes for an exchanger with a split-ring floating head. The actual layout is for a $13\,^1/_4$ in. IDS with 1 in. ODT on $1\,^1/_4$ -in. Triangular pitch arranged for six tube passes. The partition arrangement is also shown for the channel and floating-head cover along with the orientation of the passes. Tubes are not usually laid out symmetrically in the tube sheet. Extra entry space is usually allowed in the shell by omitting tubes directly under the inlet nozzle so as to minimize the contraction effect of the fluid entering the shell. When tubes are laid out with minimum space allowances between partitions and adjoining tubes and within a diameter free of obstruction called the outer tube limit, the number of tubes in the layout is the tube count. It is not always possible to have an equal number of tubes in each pass, although in large exchangers the unbalance should not be more than about 5 per cent.

Baffles

It is apparent that higher heat-transfer coefficient result when a liquid is maintained in a state of turbulence. To induce turbulence outside the tubes it is customary to employ baffles which cause the liquid to flow through the shell at right angles to the axes of the tubes. This causes considerable turbulence even when a small quantity of liquid flows through the shell. The center-to-center distance between baffles is called the baffle pitch or baffle spacing. The baffle spacing is usually not greater than a distance equal to the inside diameter of the shell or closer than a distance equal to one-fifth the inside diameter of the shell. There are several types of baffles which are employed in heat-exchangers, but by far the most common are segmental baffles. Segmental baffles are drilled plates with heights which are generally 75% of the inside diameter of the shell. These are known as 25 per cent cut baffles.

Design Procedure Of Surface Condenser

Known parameters

- Weight flow of steam , W (lb/hr)
- Inlet presuure of steam, P_s (psig)
- Temperature of steam, T_s (°F)
- Inlet temperature of water, T_1 (°F)
- Outer diameter of tube, ODT (in)
- Gage of the tube, BWG
- Tube pitch, P_T (in)
- Length of the tube, L (ft)
- Required dirt factor, R_{dr} (hr-ft^2-°F / Btu)

- Allowable pressure drop for steam, ΔP_{sa} **(psi)**
- Allowable pressure drop for water, ΔP_{wa} **(psi)**

Design Procedure

The value of T_2 is seen from HEI Standards corresponding to P_s, for h_{fg} refer HEI Standards corresponding to T_s, For the heat balance

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

w can be computed from heat balance formula., LMTD (log mean temperature difference)

$$\Delta T = (T_2 - T_1) / \log((T_s - T_1) / (T_s - T_2)), \quad (°F)$$

Assume a tentative value of U_D in the range of 175 to 1050 (Btu/hr-sq ft-°F), here we can also decide the approximate range of U_C and h_o with the aid of required dirt factor formula, so that we can get a idea to proceed further with less number of iterations. Compute the surface area, Number of tubes calculated by the formula, for the value of a''_t refer HEI Standards corresponding to ODT.

$$A = Q / (UD\ \Delta T), \quad (ft^2)$$

$$N_t = A / (L\ a''_t)$$

Now by the aid of computed value of N_t see the nearest value of N_t from table 5 or from HEI standards and corresponding to that approximate value of N_t get the values of number of tube passes n and inner diameter of shell IDS. Now the number of tubes is the value of new N_t and corresponding to that compute real surface area and overall design coefficient U_D

$$A = N_t\ L\ a''_t, \quad (ft^2)$$

$$U_D = Q / (A\ \Delta T), \quad \text{(Btu/hr-sq ft-°F)}$$

The value of computed U_D must be nearest to the value of assumed U_D, if not then assume the new value of U_D follow same steps until U_D will come satisfactory then proceed further.

Tube side (water side) calculations

For the value of a'_t corresponding to ODT & BWG refer HEI standards. Flow area of tube, Mass velocity of water and steam, Velocity of circulating water is the function of mass velocity of water, Average temperature of water

$a_t = N_t\, a'_t / \ 144\, n$, (ft^2)

$G_t = w / a_t$, (lb/hr sqft)

$V = G_t / 3600 * 62.5$ (ft per sec)

$t_a = (t_1 + t_2)/2$, (°F)

For the value of heat transfer coefficient of tube inside water h_i (Btu/hr-sq ft-°F) , refer fig. 4 corresponding to t_a. The value of heat transfer coefficient for water when referred to the tube outside diameter, Here IDT & ODT should be in ft.

$$h_{io} = h_i\, (IDT)/(ODT), \quad \text{(Btu/hr-sq ft-°F)} ,$$

Shell side (steam side) calculations

Assume baffle spacing B in the range of 1/5 IDS to IDS. Clearance, Flow area of shell, Mass velocity of steam, Condensate loading for tubes is calculated by the formulae's **below:**

$$C` = P_t - (ODT), \quad \text{(in.)}$$

$$a_s = (ID)\, C`\, B / (144\, P_T) \quad \text{(sq ft)}$$

$$G_s = W / a_s \quad \text{(lb/hr-sq ft)}$$

$$G`` = W / L\, N_t^{0.66} \quad \text{(lb/hr-lin ft)}$$

Assume the value of h_o should be greater than the value of U_D and should lie in the decided range, The tube wall temperature, film temperature is calculated by the formulae's below:

$$t_w = t_a + (h_o / (h_{io} + h_o))\,(T_s - t_a), \quad \text{(°F)}$$

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$$t_f = (T_s + t_w) / 2, \quad \text{(°F)}$$

For the value of K_f & μ_f refer HEI Standards corresponding to t_f , For the value of S_f refer HEI Standards. The average value of the condensing film coefficient or heat transfer coefficient for the outside steam is, h_o. From HEI Standards get the value of h_o **corresponding to the values of K_f, μ_f and S_f or by the formula,**

$$(h_o\ \mu_f^2) / (K_f^3\, \rho_f^2\, g) = 1.5\, (4\, G`` / \mu_f)^{-0.33}$$

Check for pressure drop calculation

The value of computed h_o must be nearest to the value of assumed $h_{o,}$ if not then assume the new value of h_o follow same steps until h_o will come satisfactory then proceed further for pressure drop calculation

Tube side pressure drop calculation:

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)**

Shell side pressure drop calculation

For the value of μ_s corresponding to $T_{s,}$ For the value of ODS corresponding ODT & P_T refer HEI Standards. The Reynolds Number for shell side fluid i.e. steam side, take ODS in ft.

$$Re_s = (ODS)\ G_s / \mu$$

For the value of friction factor f corresponding to Re_s , specific volume v_s of steam refer HEI Standards, Number of crosses, specific gravity for steam computed by the formulae's below:

$$N + 1 = 12\ L / B$$

$$s = 1 / (v_s * 62.5)$$

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. Take IDS in ft.

$$\Delta P_s = (f\ G_s^2 (IDS)(N+1)) / (2*5.22*10^{10}\ (ODS)\ s) \quad \text{(psi)}$$

The clean overall heat transfer coefficient

$$U_c = h_{io} h_o / (h_{io} + h_o) \quad \text{(Btu/hr-sq ft-°F)}$$

Dirt factor

$$R_d = (U_C - U_D) / U_C\ U_D \quad \text{(hr-sq ft- °F / Btu)}$$

The computed value of dirt factor must be nearest to the value of required dirt factor and the computed value of tube side as well as shell side pressure drop should not exceed the value of allowable pressure drop of tube side as well as shell side, if not then assume the new value of U_D and follow the whole design procedure until the values get satisfied. If the above condition get fulfill then the design is satisfactory.

SHELL SIDE	TUBE SIDE
Inner diameter of shell IDS (in.)	Outer diameter of tube ODT (in.)
Shell passes n_s	Tube passes n
Baffle spacing B (in.)	Number of tubes N_t
	Length of tube L (ft.)
	Birmingham Wire Gage BWG
	Tube pitch P_T (in.)

Table 1: Design Summary

Effects Of Key Parameters on Surface Condenser Design

The Pressure Drop is a very important parameter in design of surface condenser. 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. The condensation process occurs on constant temperature & pressure. The shell side (steam) pressure drop has to be constant but pressure drop occurs due to the reason that vacuum cannot be properly maintained inside the shell. The tube side pressure drop signifies that if the improper heat transfer occurs than pressure drop rises. Here the author done manual calculation by following the above design procedure for different values of inputs and got the key parameters which will directly or indirectly affects the pressure drop.

Velocity Of Water

The pressure drop reduces by reducing the velocity of circulating water. As if the velocity of circulating water is higher than the heat transfer rate is high. By Bernoulli's equation we know that when liquid flows from higher pressure to lower pressure than the velocity of flowing liquid will be inversely proportional to the heat transfer rate. The author also done manual calculations then the same result obtained.

Overall Design Coefficient

By reducing the overall design heat transfer coefficient the pressure drop reduces. This signifies that when overall design coefficient will be lower then the tube count will be higher and pressure drop reduces. The author validated this point by doing manual calculations.

Number Of Tubes

The pressure drop reduces by increasing the number of tubes. If the heat transfer surface area is large (i.e. number of tubes) this means that the heat transfer will be higher and pressure drop reduces for the same input parameters. The author validated this point also by doing manual calculations.

Outer Diameter Of tubes

By keeping the value of outer diameter of tube higher then the pressure drop reduces. This statement is also validated by doing manual calculation by author.

Outlet Water Temperature

For the higher value of weight flow of steam, the temperature of outlet water will be higher

Tube Pitch

By increasing the tube pitch, the inner diameter of shell increases.

Conclusion

The author considers empirical correlations and suggests a manual design procedure of a surface condenser for considering the effects of all critical parameters which directly or indirectly related to the condenser performance. This systematic design procedure gives tube side as well as shell side designing outputs. The author does manual calculations by adopting the same design procedure for different values of inputs and got the relationship between the critical parameters, i.e. pressure drop, velocity of circulating water, tube layout, baffle spacing, overall design heat transfer coefficient & outlet temperature of water. The effects of these parameters as shown above.

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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