OPTIMIZATION AND ENHANCEMENT OF HYBRID COOLING SYSTEM

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Abstract: An alternative is to use air as the cooling medium but their performance drops in hot weather conditions. Maximization of overall efficiency is as vital as the cost and availability of water. Therefore there is a need to integrate the technologies of air cooled and water cooled condensers. This paper will focus on parallel condensing where steam from the turbine is ducted in parallel to both air cooled and water cooled condensers. To optimize the parallel condensing capacities of air cooled and water cooled condensers considering parameters such as ambient temperature, pressure, availability of water, fan power, pumping costs.

Key Words: Optimization; Enhancement; Hybrid Cooling System.

1.0 INTRODUCTION:

Exhaust steam from the steam turbine is separated into two streams. One stream flows into a water cooled surface condenser while the other is directed to an air-cooled condenser. The heated cooling water is cooled as it flows through a cooling tower, where air is forced through the tower by mechanical or natural draft.

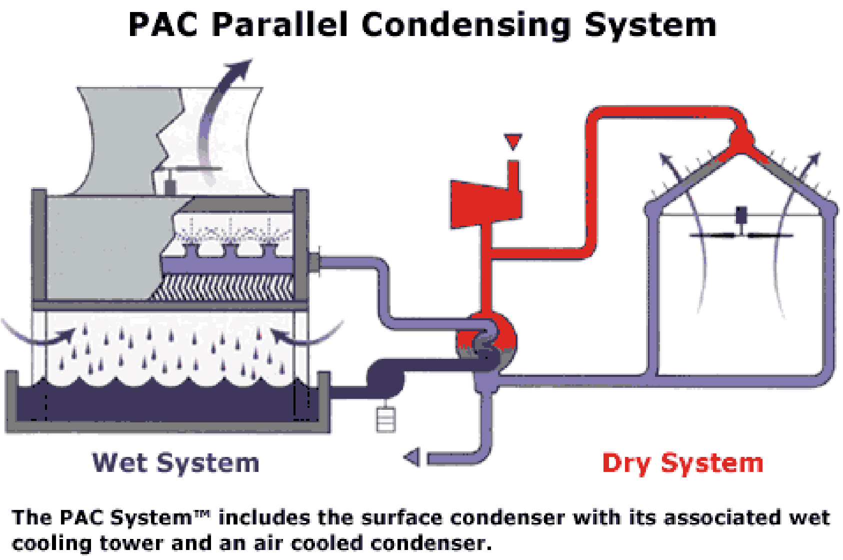


Fig .1

Condensate from the water cooled and air-cooled condenser can be collected in a common hotwell. Water consumption is controlled by the distribution of the heat load between the two condensers.
The PAC System should not be confused with a "hybrid" cooling tower, which is used primarily to reduce visible plume from a wet cooling tower. A "hybrid" cooling tower has practical limits to the amount of heat that can be rejected in the dry section, since the latter is sized for plume abatement only. With the PAC System there is complete flexibility in the amount of heat rejected in the dry section.
The dry section of the PAC System employs direct condensation in contrast to most "hybrid" systems, which are indirect condensing systems, i.e. water is cooled through both the wet and dry sections and is then pumped through a common condenser. As a result, the dry section of the PAC System can efficiently reject a substantial amount of heat even on hot days, thereby reducing peak water usage. During cooler periods, the amount of heat rejected in the dry section can be increased up to 100% if so designed, thus further reducing the plant's water consumption. The amount of steam entering each condenser at any given time depends on ambient conditions,

plant load and availability of make up water. Both condensers operate at the same condensing pressure all the time. Parallel condensing systems have been developed to save water while avoiding the high cost of dry cooling systems and to ensure a relatively low steam turbine back pressure at high ambient conditions.
An excessive rise in turbine back pressure during periods of peak ambient temperatures and demand will result in lesser efficiency. In such a case, the dry section of the system may be designed to reject the total heat load at a low ambient temperature while at high temperatures we can maintain the turbine back pressure within specified limits usig the wet part of the system. An additional benefit of the PAC System is the reduction of plume. Plume can be reduced or eliminated entirely when danger of icing exists, simply by shutting off the wet section.
A parallel system combines some positive features of dry and wet cooling systems – the water consumption is reduced compared to a 100% wet system, performance is improved compared to a 100% dry system and the capital cost decreases as the proportion of wet system is increased.

2.0 WATER COOLED CONDENSERS:

The cooling medium used is water whose source might be sea, river or ground water. This is the most efficient and frequently used type of condenser. The condenser may be single pass or two pass. Two pass condensers have higher temperature rise and require greater heat transfer surface for a performance equal to that of single pass. However we can obtain lower pressure drops using two pass condensers. Using more than two passes usually results in an uneconomical operation.
A wet cooling system has a cooling tower where the warm water is cooled in the tower using mechanical or natural draft. The water carried due to evaporation, drift and blow down has to be replenished.

Fig .2 Sketch of a water cooled condenser

A WCC is of cylindrical shape and is a shell and tube heat exchanger. The cooling water flows through the tubes and steam flows over the tubes and the latent heat of stem is rejected to the cooling water. The surface condenser consists of a casing or shell with a chamber at each end referred to as water boxes. Tube sheets separate the water boxes from the centre steam space. Banks of tubes connect the water boxes by piercing the tube sheets. Circulating pumps force the cooling water through the water boxes and connecting tubes. The non condensable gases are cooled in air cooling zones which are separated from the main tube nest by a shroud and through which coldest water passes to cool the noncondensable gases plus vapour mixture to the maximum possible extent. The tube nest is provided with liberal steam lanes for reduced pressure drop on shell side to avoid undercooling and ensure proper steam distribution in the tube nest.
The condenser needs auxiliary equipment to move cooling water through the tubes and to remove air from the steam space and condensate from the hot well. The equipment includes a steam jet air ejector or mechanical vacuum pumps, an atmospheric relief valve, and circulating and condensate pumps.

2.1 Main parts of a condenser are:

1. Shell
   It houses the tube banks and tube support plates.
2. Tube Sheets
   There are two tube sheets which have accurately drilled and reamed in to which the tubes are roller expanded.
3. Tubes

These are long and thin heat exchanger tubes in which cooling water flows and steam condenses over these tubes.

4. Tube support plates
   These are provided within the shell to prevent vibration damage of the tubes.

5. Water chambers
   There are two water chambers each at the either end of the cell. One chamber is called inlet chamber and houses cooling water inlet and outlet branches. The other chamber is called return water box.
1. Hot well
   It is provided at the bottom of condenser cell and it is used to collect the condensate. It serves the purpose of sump for condensate extraction pumps.
2. Neck
   It is provided at the top to distribute the steam throughout the length of the tube nest.

2.2 Materials for condenser tubes

The material used depends on the type of cooling water used – fresh or sea water. For fresh water – stainless steel (19%Cr, 9%Ni)

– 90/10 cupronickel

For clean sea water – 90/10 cupronickel

When ammoniacal attack is a possibility – 70/30 cupronickel is used.

For polluted sea/brackish water – 90/10 cupronickel or titanium is used.

- 90/10 cupronickel is excellent for corrosion resistance in all environments.
- Stainless steel (19%Cr, 9% Ni) is also good but is susceptible to chloride stress corrosion and biofouling.

3.0 DESIGN PROCEDURE FOR AIR COOLED CONDENSER:

In air cooled condensers where air flowing over a tube bundle cools steam and condenses it inside the tubes, the air side heat transfer co-efficient governs the overall heat transfer co-efficient. Even an error of + or – 50% on the condensation heat transfer co-efficient hardly affects the results.

There are a large number of heat transfer co relations available for predicting the heat transfer for fluids flowing in cross flow over finned tube banks. The co relations that are being used in our calculations have been developed from experimental data.

Assumptions made in thermal design of air cooled condenser :

- Free stream velocity of air = 6 to 8 m/s
- Staggered arrangement of tubes.
- Helically wound fins which are welded on tubes are used.
- Fouling factor for fluid flowing inside tubes is 0.0005 ft^2hr F/Btu
- Carbon steel is used for fins.
- Length of tube (5 to14 m) = 12m

Steps involved in the calculation are as follows :

Step 1: find the heat duty of the condenser

$$Q = w_s (x) h_{fg}$$

Step 2: find the mass flow rate of air

$$w_a = Q/ C_{pa} (t_{ao} – t_{ai})$$

Step 3: Find the number of tubes

$$N= (4w_a)/(\pi d_i^2 \rho_a v)$$

Step 4: Find maximum air velocity

$$v_{max} = Max(v_1 ,v_2)$$

$$v_1 = S_t v_a /(S_t – d_0)$$

$$v_2 = S_l v_a /2(S_d - d_0)$$

$$S_d = (S_l^2 + S_t^2 /4)^{0.5}$$

Maximum velocity of air in a staggered tube arrangement can occur either in the transverse plane or longitudinal plane depending on the values of transverse and longitudinal pitch. For determining the Reynolds number we need to use the maximum of the above obtained values. For an inline arrangement maximum velocity always occurs in the transverse plane.

Step 5: Find Nusselt number

$$Nu = 0.19\ Pr^{0.33}\ Re^{0.65}\ (Y/H)^{0.2}$$

The above correlation can be used only with helically wound fins welded on tube.

Step 6: Find the fin side heat transfer co-efficient

$$h_c' = 12Nu\ k / d_0$$

Step 7: Determination of fin efficiency

$$m = (24\ h_c' /(k_m\ b))^{0.5}$$

$$\Phi = (1 + m\ b\ (2(d_0+2H)/d_0)^{0.5})^{-1}$$

$$\eta = 1-(1- \Phi)A_f /A_t$$

Step 8: Finding the corrected fin side heat transfer co-efficient

$$h_c = \eta\ h_c'$$

Step 9: Find inside heat transfer co-efficient

$$h_i = Z\ (n_b\ L / W_s)^{0.333}$$

Step 10: Find overall heat transfer co-efficient

$$1/U = (1/h_i\ (A_t /A_i)) + ff\ (A_t /A_i\ ) +(A_t /A_i\ \ (d_0 /24k_m\ ) \ln(d_0 /d_i)+1/h_c$$

The first term represents the resistance offered by the fluid flowing inside the tubes. Second term represents the resistance due to the fouling factor. The last two terms indicate the resistance due to the thermal conductivity of tube material and due to fluid flowing outside the tubes respectively.

Step 11: Find heat transfer area

$$A = Q / (U\ \Delta T_{LM})$$

Step 12: Find fin side pressure drop

$$\Delta p = f\ G^2\ n / (\rho_a(S_t /d_0\ ) (Y/H)^{0.71}\ )$$

The friction factor f is found from a graph between Reynold's number and friction factor. If Δp is within 12mm of water column, the design is acceptable.

Step 13: Find fan power

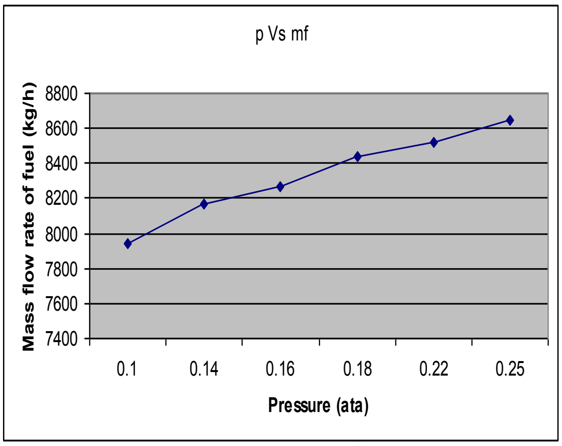
$$P_{fan} = Qh_f / (102\eta_f\ )$$

The above mentioned procedure was followed and the following results were obtained:

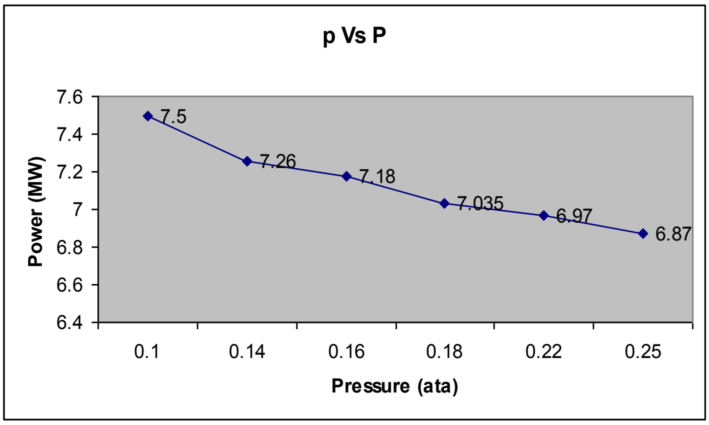
Pressure of steam,p	0.22ata
Steam condensed,w_s	29814 kg/hr
Air temperature inlet,t_1 outlet,t_2	42^0C 57^0C
Flow rate of air required, w_c	1179.2 kg/s
Air velocity,v	6 m/s
Number of tubes, N	2694
Tube size	1.9 in O.D, 1.61 in I.D.
Overall heat transfer coefficient ,U	37.27 W/m^2k
Heat transfer area required,A	45562 m^2
Length of tubes,L	12 m
Fin height,H	0.625 in
Fin thickness,b	0.064 in
Fin clearance, A	0.136 in
Fins per inch	5
Transverse pitch,S_t	130 mm
Longitudinal pitch,S_l	70 mm
Fin side pressure drop, Δp	10 mm of water
Fan power	168.16 KW

4.0ANALYSIS OF OPERATING PARAMETERS:

1. Effect of turbine exhaust pressure on fuel consumption



2. Effect of turbine exhaust pressure on power output



5.0 CONCLUSION:

From the above methods of analysis it is clear that dry cooling is comparable to wet cooling at the assumed rates of water and power. If water is available in sufficient quantities at a cheaper rate, wet cooling is economical. When water is available but only for certain months in the year we can opt for a parallel condensing system. In this case, the capital cost decreases and performance is improved compared to 100% dry cooling system. The percentage of dry cooling and wet cooling in a parallel condensing system depends on the water availability.

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