Performance Analysis of Cooling Tower

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Abstract *Power plants, some other industries produce a large quantity of waste heat in the form of hot water. In the present scenario, in most of the places, the water supply is limited and thermal pollution is also a serious concern. Considering the recent increase of interest in analysing these problems and solving them for the well being of the environment, this work is an attempt to deal with the technology, applications of cooling towers. In this present study, the factors affecting the performance like environmental conditions, cooling water quality have been studied on Induced draft cooling tower of 32 Mw thermal power plant. The performance parameters like range, approach, cooling capacity, evaporation loss liquid to gas ratio have been evaluated when the plant is operated at full load and part load under the same water flow rates..*

Keywords — *Power plant, cooling tower, induced draft, range, approach, environmental conditions, cooling capacity, evaporation loss, L/G ratio, water flow rates.*

1. Introduction

The Cooling tower is a direct contact type heat exchanger. It is a semi-closed, evaporative cooler. The water is sprayed through a certain height and an air current passes over it. Some water evaporates and the heat of evaporation is extracted from falling water and surrounding air, thus water cool.

NECESSITY OF COOLING THE CONDENSER WATER:

The cooling water system is one of the most important systems of power plant and its availability predominantly decides the plant site. The high cost of water makes it necessary to use cooling towers for water cooled condensers.

As the cooling water absorb the latent heat of steam in the condenser, the temperature of the water increases. The hot water coming out of the condenser cannot be used again in a closed system without pre reaches near to T_s saturation temperature of steam at condenser pressure and the condenser vacuum cannot be maintained. Therefore it is absolutely necessary to pre cool the water coming out of condenser before using again.

The cooling water requirement in an open system is about 50 times the flow of steam to the condenser. Even with closed cooling system using cooling towers, the requirement for cooling water is also considerably large as 5 to 8 kg/kW hr. This means a 1000 MW station will require about 100 thousand tunes of circulating water per day even with the use of cooling towers.

2. Merkel Theory:

Dr. Merkel developed a cooling tower theory for the mass (evaporation of a small portion of water) and sensible heat transfer between the air and water in a counter flow cooling tower. The theory considers the flow of mass and energy from the bulk water to an interface, and then from the interface to the surrounding air mass. The flow crosses these two boundaries, each offering resistance resulting in gradients in temperature, enthalpy, and humidity ratio. Merkel demonstrated that the total heat transfer is directly proportional to the difference between the enthalpy of saturated air at the water temperature and the enthalpy of air at the point of contact with water.

$Q = K \times S \times (h_w - h_a)$

Where,

l Q = total heat transfer Btu/h

l K = overall enthalpy transfer coefficient lb/hr.ft_2

l S = heat transfer surface ft_2.

$$S = a \times V,$$

Which "a" means area of transfer surface per unit of tower volume. (ft_2/ft_3), and V means an effective tower volume (ft_3).

l h_w = enthalpy of air-water vapor mixture at the bulk water temperature, Btu/Lb dry air

l h_a = enthalpy of air-water vapor mixture at the wet bulb temperature, Btu/Lb dry air

The water temperature and air enthalpy are being changed along the fill and Merkel relation can only be applied to a small element of heat transfer surface dS.

$dQ = d[K \times S \times (h_w - h_a)] = K \times (h_w - h_a) \times dS$

The heat transfer rate from water side is $Q = C_w \times L \times$ Cooling Range, Where C_w = specific heat of water = 1,

L = water flow rate.
Therefore, $dQ = d[C_w \times L \times (tw_2 - tw_1)] = C_w \times L \times dtw$.

Also, the heat transfer rate from air side is $Q = G \times (ha_2 - ha_1)$, where G = air mass flow rate Therefore, $dQ = d[G \times (ha_2 - ha_1)] = G \times dha$.

Then, the relation of $K \times (h_w - h_a) \times dS = G \times dha$ or $K \times (h_w - h_a) \times dS = C_w \times L \times dtw$ are established, and these can be rewritten in
$K \times dS = G / (h_w - h_a) \times dha$ or $K \times dS / L = C_w / (h_w - h_a) \times dtw$.
By integration,

$$\frac{KS}{L} = \frac{KaV}{L} = \frac{G}{L}\int_{ha1}^{ha2} \frac{dh}{hw - ha} \qquad \frac{KS}{L} = \frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw - ha}$$

This basic heat transfer equation is integrated by the four point Tchebycheff, which uses values of y at predetermined values of x within the interval a to b in numerically evaluating the integral $\int_a^b y\,dx$ The sum of these values of y multiplied by a constant times the interval (b - a) gives the desired value of the integral. In its four-point form the values of y so selected are taken at values of x of 0.102673.., 0.406204.., 0.593796.., and 0.897327..of the interval (b - a). For the determination of KaV/L, rounding off these values to the nearest tenth is entirely adequate. The approximate formula becomes:

$$\int_a^b y\,dx = (b - a) \times (y_1 + y_2 + y_3 + y_4) / 4$$

$$\text{where, } y_1 = \text{value of } y \text{ at } x = a + 0.1 \times (b - a) = CWT + 0.1 \times Range$$

$$y_2 = \text{value of } y \text{ at } x = a + 0.4 \times (b - a) = CWT + 0.4 \times Range$$

$$y_3 = \text{value of } y \text{ at } x = b - 0.4 \times (b - a) \text{ or } x = a + 0.6 \times (b - a) = CWT + 0.6 \times Range$$

$$y_4 = \text{value of } y \text{ at } x = b - 0.1 \times (b - a) \text{ or } x = a + 0.9 \times (b - a) = CWT + 0.9 \times Range$$

For the evaluation of KaV/L,

$$\frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw - ha} = (tw_2 - tw_1) \times [(1 / Dh_1) + (1 / Dh_2) + (1 / Dh_3) + (1 / Dh_4)]$$

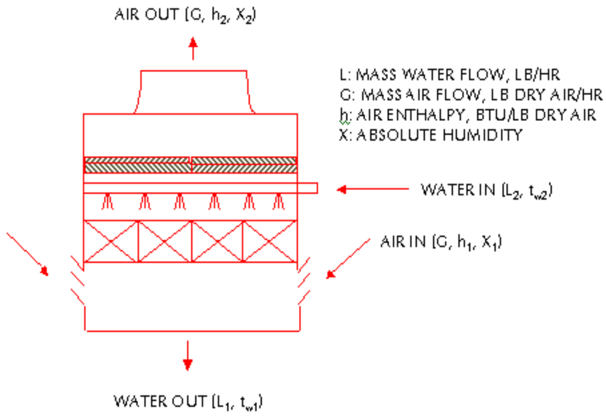
where, Dh_1 = value of (hw - ha) at a temperature of CWT + 0.1 x Range

Dh_2 = value of (hw - ha) at a temperature of CWT + 0.4 x Range

Dh_3 = value of (hw - ha) at a temperature of CWT + 0.6 x Range

Dh_4 = value of (hw - ha) at a temperature of CWT + 0.9 x Range

3. Heat Balance



HEAT in = HEAT out
WATER HEAT in + AIR HEAT in = WATER HEAT out + AIR HEAT out
$C_w L_2 tw_2 + G h_{a1} = C_w L_1 tw_1 + G h_{a2}$

The difference between L_2 (entering water flow rate) and L_1 (leaving water flow rate) is a loss of water due to the evaporation in the direct contact of water and air. This evaporation loss is a result of difference in the water vapor content between the inlet air and exit air of cooling tower. Evaporation Loss is expressed in $G \times (w_2 - w_1)$ and is equal to $L_2 - L_1$. Therefore, $L_1 = L_2 - G \times (w_2 - w_1)$ is established.

Let's replace the term of L_1 in the right side of Eq. 2-1 with the equation of $L_1 = L_2 - G \times (w_2 - w_1)$ and rewrite.
Then, $C_w L_2 tw_2 + G h_{a1} = C_w [L_2 - G \times (w_2 - w_1)] \times tw_1 + G h_{a2}$ is obtained.
This equation could be rewritten in
$C_w \times L_2 \times (tw_2 - tw_1) = G \times (h_{a2} - h_{a1}) - C_w \times t_{w1} \times G \times (w_2 - w_1)$.

In general, the 2nd term of right side is ignored to simplify the calculation under the assumption of $G \times (w_2 - w_1) = 0$.
Finally, the relationship of $C_w \times L_2 \times (tw_2 - tw_1) = G \times (h_{a2} - h_{a1})$ is established and this can be expressed to

$C_w \times L \times (t_{w2} - t_{w1}) = G \times (h_{a2} - h_{a1})$ again.
Therefore, the enthalpy of exit air,

$h_{a2} = h_{a1} + C_w \times L / G \times (t_{w2} - t_{w1})$ is obtained.

The value of specific heat of water is Eq. 2-1 and the term of t_{w2} (entering water temperature) - t_{w1}

(leaving water temperature) is called the cooling range.

Simply, $h_{a2} = h_{a1} + L/G \times Range$

Consequently, the enthalpy of exit air is a summation of the enthalpy of entering air and the addition of enthalpy from water to air (this is a value of L/G x Range).

4. Mechanical draft towers:

Mechanical draft towers are available in the following airflow arrangements:

a. Counter flow forced draft.
b. Counter flow induced draft.
c. Cross flow induced draft.

In the counter flow induced draft design, hot water enters at the top, while the air is introduced at the bottom and exits at the top. Both forced and induced draft fans are used

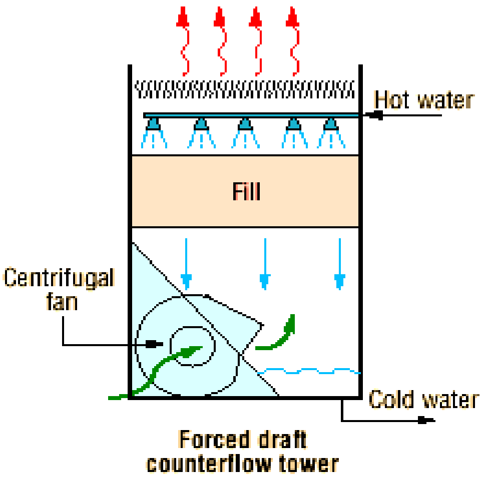


Fig. 2.5 Counter flow forced draft

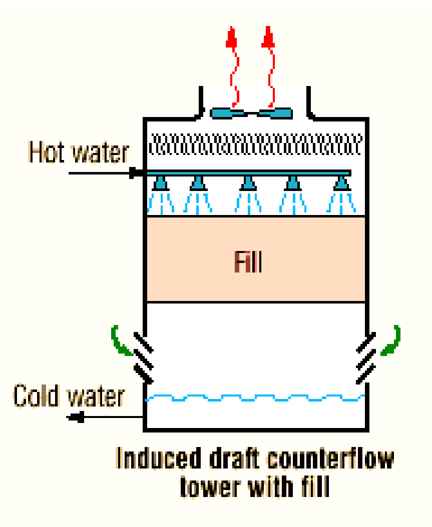


Fig. 2.6 Counter flow induced draft

In cross flow induced draft towers, the water enters at the top and passes over the fill. The air, however, is introduced at the side either on one side (single-flow tower) or opposite sides (double-flow tower). An induced draft fan draws the air across the wetted fill and expels it through the top of the structure.

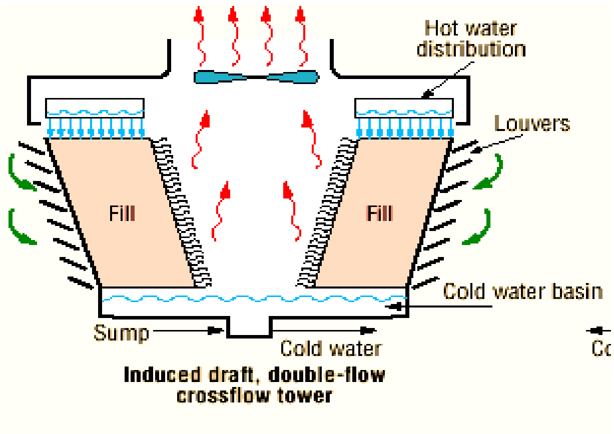


Fig. 2.7 Cross flow induced draft.

5. Performance Parameters:

The measured parameters are used to determine the cooling tower performance in several ways. These are:

❖**Range:** This is the difference between the cooling tower water inlet and outlet temperature. A high CT Range means that the cooling tower has been able to reduce the water temperature effectively, and is thus performing well. The formula is:

CT Range (°C) = [CW inlet temp (°C) – CW outlet temp (°C)]

❖ **Approach:** This is the difference between the cooling tower outlet coldwater temperature and ambient wet bulb temperature. The lower the approach the better the cooling tower performance. Although, both range and approach should be monitored, the `Approach' is a better indicator of cooling tower performance.

CT Approach (°C) = [CW outlet temp (°C) – Wet bulb temp (°C)]

❖**Effectiveness:** This is the ratio between the range and the ideal range (in percentage), i.e. difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = Range / (Range + Approach). The higher this ratio, the higher the cooling tower effectiveness.

CT Effectiveness (%) = 100 x (CW temp – CW out temp) / (CW in temp – WB temp)

- **Cooling capacity:** This is the heat rejected in kCal/hr or TR, given as product of mass flow rate of water, specific heat and temperature difference.
- **Evaporation loss:** This is the water quantity evaporated for cooling duty. Theoretically the evaporation quantity works out to 1.8 m^3 for every 1,000,000 kCal heat rejected. The following formula can be used (Perry):

Evaporation loss (m^3/hr) = 0.00085 x 1.8 x circulation rate (m^3/hr) x (T_1-T_2)

T_1 - T_2 = temperature difference between inlet and outlet water

- **Cycles of concentration (C.O.C):** This is the ratio of dissolved solids in circulating water to the dissolved solids in make up water.

- **Blow down:** Blow down losses depend upon cycles of concentration and the evaporation losses and is given by formula:

Blow down = Evaporation loss / (C.O.C. – 1)

- **Liquid/Gas (L/G) ratio:** The L/G ratio of a cooling tower is the ratio between the water and the air mass flow rates. Cooling towers have certain design values, but seasonal variations require adjustment and tuning of water and air flow rates to get the best cooling tower effectiveness. Adjustments can be made by water box loading changes or blade angle adjustments. Thermodynamic rules also dictate that the heat removed from the water must be equal to the heat absorbed by the surrounding air. Therefore the following formulae can be used:

$$L(T_1 – T_2) = G(h_2 – h1)$$
$$L/G = (h_2 – h_1) / (T_1 – T2)$$

Where:

L/G = liquid to gas mass flow ratio (kg/kg)
T_1 = hot water temperature (^{0}C)
T_2 = cold-water temperature (^{0}C)
h_2 = enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (same units as above)
h_1 = enthalpy of air-water vapor mixture at inlet wet-bulb temperature (same units as above).

6. TECHNICAL DATA OF THE COOLING TOWER:

Type of cooling tower: Induced draft counter flow	
Number of cells	: 3
Cooling water flow	: 7000 m^3/hr
Hot water inlet temperature: 43^{0}C	
Wet bulb temperature	: 27.5^{0}C
Re-cooled water temperature: 33^{0}C	
Make-up water temperature (Based on COC-4)	: 41.22 m^3/hr
Inlet air enthalpy	: 18.10kCal/kg
Exit air enthalpy	: 35.55kCal/kg
Approach to inlet WBT	: 5^{0}C
Cooling range	:10^{0}C
Air flow	: 8628.1 kg/m^2-hr
L/G ratio	: 1.65
Guaranteed drift losses	: 3.50 m^3/hr-(0.05%)
Evaporation losses based on 60% RH: 1.59%	
Blow down for concentration of 4	: 0.48%

6.1 AT 100% LOAD OF GENERATING STATION (I.E. AT 32 MW):

Data at 32 MW:

Circulating water flow = 7000 m^3/hr
=1.944kg/sec

Circulating water inlet temperature = 43^{0}C

Circulating water outlet temperature= 33^{0}C

Air inlet temperatures T_1 = DBT=32^{0}C
WBT=28^{0}C (i.e., RH is 74%)

Air outlet temperatures T_2 = DBT = 40^{0}C
WBT = 39^{0}C (i.e., RH is 95%)

Dissolved solids in circulating water =800ppm

Dissolved solids in make-up water =120ppm

Specific humidity of entering air ω_1:

Vapor pressure of the entering air pv_1:

$$Pv_1 = (P_{vsat})\,wb - \frac{[P- (P_{vsat})\,wb]\,[DBT-WBT]\,1.8}{2854 – 1.325\,[(1.8\,DBT) + 32]}$$

P_{vsat} at 28^{0}C WBT =0.03778 bar

$$= 0.03778 - \frac{[1.0132 – 0.03778]\,[32-28]\,1.8}{2854 – 1.325[(1.8x32) + 32]}$$

= 0.035212 bar

Absolute humidity or specific humidity of entering air ω_1:

$$\omega_1 = 0.622 \frac{P_v}{P_a - P_v}$$

$$\omega_1 = 0.622 \frac{0.035212}{1.0132 – 0.035212}$$

= 0.022 kg/kg of dry air

Specific humidity of leaving air ω_3:

Vapor pressure of the leaving air pv_3:

$$Pv_3 = (P_{vsat})\,wb - \frac{[P - (P_{vsat})\,wb]\,[DBT - WBT]\,1.8}{2854 - 1.325\,[(1.8\,DBT) + 32]}$$

P_{vsat} at 39^0C WBT =0.0991 bar

$$= 0.06991 - \frac{[1.0132 - 0.06991]\,[40-39]\,1.8}{2854 - 1.325[(1.8x32) + 32]}$$

= 0.069847 bar

Absolute humidity or specific humidity of leaving air ω_3:

$$\omega_3 = 0.622\,\frac{P_v}{P_a - P_v}$$

$$\omega_3 = 0.622\,\frac{0.069847}{1.0132 - 0.069847}$$

= 0.04605 kg/kg of dry air

Range:

CT Range (°C) = [CW inlet temp (°C) – CW outlet temp (°C)]

= 43-33

=10^0C

Approach:

CT Approach (°C) = [CW outlet temp (°C) – Wet bulb temp (°C)]

= 33-28

= 5^0C

Effectiveness:

CT Effectiveness (%) =100 x (CW in temp – CW out temp) / (CW in temp – WBT)

= 100 x (43-33) / (43-38)

= 66.67%

Cooling capacity:

This is the heat rejected in kCal/hr or TR, given as product of mass flow rate of water, specific heat (4.2kJ/kg) and temperature difference.

Cooling capacity (Q) = mass flow rate of water X spe.heat X temp. difference

= 1.944 X 4.2 X (43-33)

= 81.648kJ/sec

=19.5kCal/sec

Evaporation loss:

Evaporation loss(m^3/hr)=0.00085 x 1.8 x circulation rate(m^3/hr) (T1-T2)

= 0.00085 X 1.8 X 7000 X (43-33)

= 107.1m^3/hr

Percentage of evaporation = (107.1/7000) x 100

= 1.53%

Cycles of concentration (COC):

$$COC = \frac{\text{Dissolved solids in circulating water}}{\text{Dissolved solids in makeup water}}$$

= 800/120

= 6.67

Blow down = Evaporation loss / (C.O.C. – 1)

= 107.1 / (6.67-1)

= 18.89m^3/hr

= 0.27%

Make-up water required:

Total losses in circulating water=Make-up water Evaporation losses + Blow down losses + Drift losses

= 107.1+18.89+3.5

= 129.49m^3/hr

= 1.85%

Liquid/Gas (L/G) ratio:

$$L(T_1 - T_2) = G(h_3 - h_1)$$

$$L/G = \frac{(h_3 - h_1)}{(T_1 - T_2)}$$

Where:

L/G = liquid to gas mass flow ratio (kg/kg)

T_1 = hot water temperature (^{0}C)

T_2 = cold-water temperature (^{0}C)

h_3 =enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (i.e. at 39^0C) from steam tables in kCal/kg = $(CpT_{3wb} + \omega_3 hv_{3wb})$

= (1.005 x 39) + (0.04605 x 2572.6)

= 157.66kJ/kg =37.54 kCal/kg

h_1=enthalpy of air-water vapor mixture at inlet wet-bulb temperature (i.e. at 28^0C in kCal/kg = $(CpT_{1wb} + \omega_1 hv_{1wb})$

= (1.005 x 28) + (0.022 x 2552.7)

= 84.3kJ/kg =20.07 kCal/kg

$$L/G = \frac{(37.54 - 20.07)}{(43 - 33)}$$
$= 1.747.$

6.2 AT 70% LOAD OF GENERATING STATION (I.E. AT 22.4 MW):
Data at 22.4MW :

Circulating water inlet temperature $= 40^0C$

Circulating water outlet temperature $= 30^0C$

Air inlet temperatures T_1 $= DBT=32^0C$

$WBT=28^0C$ (i.e. RH is 78.7%)

Air outlet temperatures T_2 $= DBT = 38^0C$

$WBT = 37^0C$ (i.e.RH is 98.9%)

Specific humidity of entering air $\omega_{1:}$

From previous calculation;

Vapor pressure of the entering air pv_1= 0.035212 bar

Absolute humidity or specific humidity of entering air
ω_1 =0.022kg/kg of dry air

Specific humidity of leaving air ω_3:
Vapor pressure of the leaving air pv_3:

$$Pv_3= (P_{vsat})\ wb - \frac{[P- (P_{vsat})\ wb]\ [DBT-WBT]\ 1.8}{2854 - 1.325\ [(1.8\ DBT) + 32]}$$

P_{vsat} at 37^0C WBT =0.06991 bar

$$=0.06274 - \frac{[1.0132 - 0.06274]\ [38 - 37]\ 1.8}{2854 - 1.325[(1.8x38) + 32]}$$
= 0.06211 bar

Absolute humidity or specific humidity of leaving air ω_3:

$$\omega_3 = 0.622 \frac{P_v}{P_a - P_v}$$

$$\omega_3 = 0.622 \frac{0.06211}{1.0132 - 0.06211}$$

= 0.0406 kg/kg of dry air

Range:
CT Range (°C) = [CW inlet temp (°C) – CW outlet temp (°C)]
= 40-30
$=10^0C$

Approach:
CT Approach (°C) = [CW outlet temp (°C) – Wet bulb temp (°C)]
= 30-28
$= 2^0C$

Effectiveness:
CT Effectiveness (%) =100 x (CW in temp – CW out temp) / (CW in temp – WBT)
= 100 x (40-30) / (40-28)
= 83.33%

Cooling capacity:
This is the heat rejected in kCal/hr or TR, given as product of mass flow (1.944kg/sec) rate of water, specific heat (4.2kJ/kg) and temperature difference (40-30).
Cooling capacity (Q)= mass flow rate of water X spe.heat X temp. Difference
= 1.944 X 4.2 X (40-30)
= 81.648kJ/sec
= 19.5 kCal/sec

Evaporation loss:

Evaporation loss (m^3/hr)

= 0.00085x1.8x circulation rate (m^3/hr)x(T1-T2)

= 0.00085 X 1.8 X 7000 X(40-30)

= $107.1m^3$/hr

Percentage of evaporation = (107.1/7000) x 100
=1.53%

Liquid/Gas (L/G) ratio:

$L(T_1 – T_2)= G(h3 – h1)$
Where:
T_1 = hot water temperature (^0C)=40^0C
T_2 = cold-water temperature (^0C)=30^0C

h_3=enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (i.e. at 37^0C) from steam tables in kCal/kg = $(CpT_{3wb} + \omega_3 hv_{3wb})$
= (1.005 x 37) + (0.0406 x 2569.0)
= 141.486kJ/kg =33.7 kCal/kg
h_1=enthalpy of air-water vapor mixture at inlet wet-bulb temperature (i.e. at 28^0C in kCal/kg = $(CpT_{1wb} + \omega_1 hv_{1wb})$
= (1.005 x 28) + (0.022 x 2552.7)
= 84.3kJ/kg =20.07 kCal/kg

$$L/G = \frac{(33.7\text{-}20.07)}{(40 - 30)}$$
= 1.363

7. DISCUSSION OF RESULTS

- For the same ambient air inlet conditions the range of the cooling tower is maintaining at the same value even though at part load.
- Approach has been improved (i.e. value is decreased) in part load operation of the cooling tower.
- Effectiveness or efficiency of the cooling tower has been improved during part load operation. This increase in effectiveness may be due to increase in potentiality to absorb heat from

cooling water, because of more difference between hot water and wet bulb temperature of entering air.

- Cooling capacity of the cooling tower is same for both full load and part load condition because the temperature differential is same in both the conditions and also the water flow rate is kept constant.
- Evaporation loss in both cases is same. It is evident that, during part load condition, maintaining cooling water flow at the same rate as in full load condition may not be economical, because for the same power output evaporation losses are seems to be more.

It can be observed that in part load condition liquid to gas ratio is decreased even for same heat removal and same temperature differential as in full load condition. We know that L/G ratio is depends on inlet and outlet air enthalpies at their wet bulb temperatures and differential temperature of cooling water and however as outlet air enthalpy at WBT is decreased obviously the L/G ratio decreased.

CONCLUSION

The climatic conditions like air dry bulb and wet bulb temperature, relative humidity will affect the performance of the cooling tower.

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