Measurement of Air Flow Requirement, Alter the Fan Blade Angle for Optimum Performance and Energy Consumption of a Cooling Tower

M.V.H.Satish Kumar
Department of Mechanical Engineering
P V P Siddhartha Institute of Technology, Kanuru, Vijayawada – 7. Andhra Pradesh, India

Abstract - Over a couple of decades, great attempts are made in improving the performance of cooling towers. Since the studies on optimum conditional operations of cooling towers are seldom found in literature, this work is an attempt to provide some information in this direction. This study deals with the technology, application of cooling towers. In this work, studies are made on cooling tower fan of induced draft counter flow type cooling tower of 32 MW thermal power plant. In this present study, the air flow requirements have been calculated for different climatic conditions and made selection of suitable fan blade angle to evaluate the shaft power, static pressure, static efficiency and energy savings.

Keywords – Cooling tower, Air flow requirements, fan blade angle, shaft power, static pressure, static efficiency

I. INTRODUCTION

In a cooling tower, water is made to trickle drop by drop so that it comes in contact with the air moving in the opposite direction. As a result of this some water is evaporated and is taken away with air. In evaporation, the heat is taken away from the bulk of water, which is thus cooled.

.

1.1 WATER COOLING METHOD USED IN STEAM CONDENCER OF POWER PLANT:

The cooling of water coming out from the condenser is done by either cooling pond or cooling tower. The hot water gets cooled when exposed to atmospheric air, that is because, the atmospheric air has definite capacity to absorb water vapour at given temperature and the heat required for water vapourization is taken from the remaining water and water gets cooled.

Induced draft cooling towers are compact in size with low height. The air movement through the tower is created by induced draft fans, located at the top of the tower. The performances of the induced draft fans are important in optimum operation of a cooling tower.

Induced draft cooling towers can control the temperature of out let water precisely. However, the induced draft towers enable better control on heat transfer process, which is favorable for varying load on power plant and changing ambient conditions.

1.2 BASICS OF COOLING TOWERS:

The demand for electric power has increased to such an extent that it is now no longer justifiable to site the large power station near river side's. The present trend is to locate the station near the load center with the use of large and highly efficient cooling towers.

The cooling towers are desired when positive control on the temperature of water is required, the space occupied by the cooling system is considerable factor and the plant is situated near load center and far away from the adequate natural resources of cooling water.

The principle of cooling the in cooling tower is similar to the evaporative condenser or spray pond. The rate of evaporation of water in cooling tower and subsequent reduction in water temperature depends upon the following factors:

a. Amount of water surface area exposed.
b. The time of exposure.
c. The relative velocity of air passing over the water droplets formed in cooling tower.
d. The relative humidity of air and difference between the inlet air WBT and water inlet temperature.
e. The direction of air flow relative to water.

Higher the surface area more time of exposure, lower relative humidity, higher difference between WBT of air and water inlet temperature, and cross flow give effective cooling and reduce the tower size.

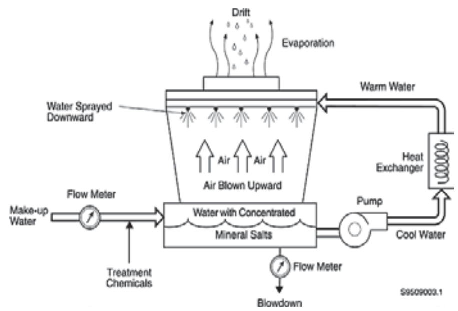


Fig.1.2 Schematic diagram of a cooling water system

A cooling tower is a semi-enclosed device as shown in the figure for evaporative cooling of water by contact with air. It is a wooden, steel or concrete structure and corrugated surfaces and trough or baffles or perforated trays are provided inside the tower for uniform distribution and better atomization of water in the tower. The hot water coming out from the condenser is fed to the tower on the top and allowed to tickle in from of thin sheets or drops. The air flows from bottom of the tower or perpendicular to the direction of water flow and then exhausts to the atmosphere after effective cooling. To prevent the escape of water particles with air, draft eliminators are provided at the top of the tower.

2.1 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.

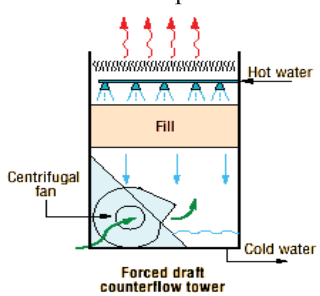


Fig. 2.1 Counter flow forced draft

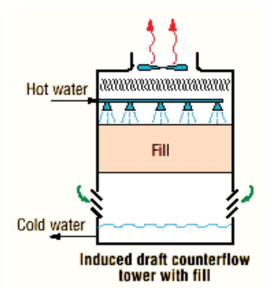


Fig. 2.2 Counter flow induced draft

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.

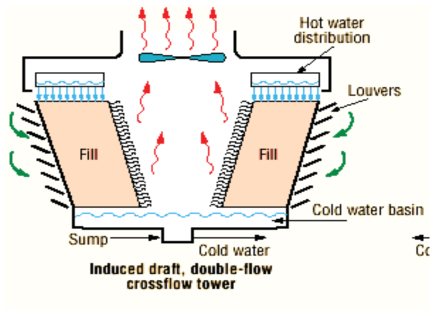


Fig. 2.3 Cross flow induced draft.

III. OBSERVATIONS AND CALUCULATION PROCEDURES FOR OPTIMAL PERFORMANCE ENERGY CONSUMPTIONS OF COOLING TOWERS

TECHNICAL DATA OF THE COOLING TOWER:

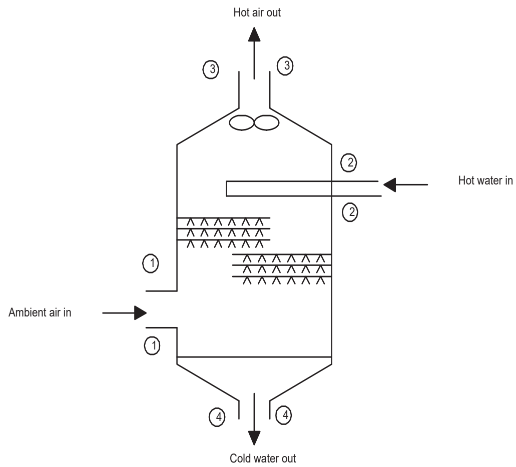
Type of cooling tower	: Induced draft counter flow
Number of cells	: 3
Cooling water flow	: 7000 m^3/hr
Hot water inlet temperature	: 43^0C
Wet bulb temperature	: 27.5^0C
Re-cooled water temperature	: 33^0C
Make-up water temperature (Based on COC-4)	: 41.22 m^3/hr
Inlet air enthalpy	: 18.10kCal/kg
Exit air enthalpy	: 35.55kCal/kg
Approach to inlet WBT	: 5^0C
Cooling range	:10^0C
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%

OBSERVATIONS AND DATA COLLECTION

During winter season where the ambient temperature can be maintained at lower than that of in summer, the air flow requirement for a cooling tower is decreased. By calculating the optimal air flow and altering the blade of the cooling fan according to the requirement will lead to saving in power being consumed by the fans.

It is collected, the daily average data from the cooling tower plant at two different seasons to calculate the optimal air flow requirement. The optimum operating points of the fan has been taken from the performance characteristic curves of the fan.

Calculation Procedure



Case 1:

Water flow Q = 7000 m^3/Hr
Water inlet temperature = 43^0C
Water outlet temperature = 33^0C
Ambient pressure = 1.0132 bar
Ambient air density = 1.102 kg/m^3
Air inlet temperatures T_1 = DBT =32^0C
WBT =28^0C (i.e., RH is 74.08%)
Air outlet temperatures T_2 = DBT = 40^0C
WBT = 39^0C (i.e., RH is 94.7%)

Vapor pressure of the entering air Pv1:

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

P_{vsat} at 28^0C 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

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 =0.06991 bar

[1.0132 – 0.06991] [40 - 39] 1.8

$$=0.06991 - \frac{}{2854 - 1.325[(1.8x40) + 32]}$$

= 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

From steam tables:

h_{v1} at 32^0C = 2560.0 kJ/kg
h_{v3} at 40^0C = 2574.4 kJ/kg
hf_3 = 167.5 kJ/kg
hf_4 = 138.2 kJ/kg

Now from energy balance

$[h_{a1} + \omega_3 hv_1] + m_{w2}$ x h_{f2} = $[h_{a3} + \omega_3 h_{v3}] + m_{w4}$ x h_{f4}

$C_p(T_1\text{-}T_3)+ \omega_1 h_{v1}+m_{w2}$ = $\omega_3 h_{v3\,-}\,[m_{w2} - (\omega_1\text{-}\ \omega_3)]\ h_{f4}$

1.005(32-40)+(0.0224 x 2560)+(m_{w2} x 167.5) =(0.04605 x 2574.7)+[m_{w2}(0.04605-0.0224)] x 138.2

-8.04+57.6+ m_{w2}167.2 = 118.565+ m_{w2}138.2 - 3.268

=> m_{w2} = 1.59626 kg of water/kg of dry air

Hence total quantity of air flow required to cool 7000 m^3/Hr (Ambient air density is 1.102 kg/m^3)

=> m_a = [7000 x 1000] / [1.102 x1.59626 x 3600]
= 1105.4 m3/Sec

Required air flow per each cell = (1105.4 / 3)
= 368.5m^3/Sec

In the performance curves of fan, the fan blade angle was set at 12^0 to meet the above required air flow with optimum static pressure and efficiency.

From the curves at 12^0 of the blade angle:

Fan shaft power =43.5 kW

Static pressure =98 Pascal

$$\text{Static efficiency} = \frac{\text{Air flow in m}^3\text{/sec x static pressure}}{\text{1000 x Fan shaft powering kW}}$$

= 86.33%

Case 2:

Water flow Q = 7000 m^3/Hr
Water inlet temperature = 40^0C
Water outlet temperature = 30^0C
Ambient pressure = 1.0132 bar
Ambient air density = 1.102 kg/m3
Air inlet temperatures T_1 = DBT =28^0C

WBT $=25^0$C (i.e., RH is 78.7%)

Air outlet temperatures T_2 = DBT $=36^0$C

WBT $=35^0$C (i.e., RH is 94.6%)

Vapor pressure of the entering air pv1:

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

P_{vsat} at 25^0C WBT =0.03166 bar

$$= 0.03166 - \frac{[1.0132 - 0.03166]\,[28-25]}{2854 - 1.325[(1.8x28)+32]}$$

Pv_1 = 0.02973 bar

Absolute humidity or specific humidity of entering air ω_1:

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

$$= 0.622 \frac{0.02973}{1.0132 - 0.02973}$$

ω_1 = 0.0188 kg/kg of dry air

Vapor pressure of the leaving air P_{v3}:

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

P_{vsat} at 35^0C WBT =0.05622

$$=0.05622 - \frac{[1.0132 - 0.05622]\,[40-39]}{2854 - 1.325[(1.8x40)+32]}$$

Pv_3 = 0.0556 bar

Absolute humidity or specific humidity of leaving air ω_3:

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

$$= 0.622 \frac{0.0556}{1.0132 - 0.0556}$$

ω_3 = 0.0361 kg/kg of dry air

From steam tables:

h_{v1} at 28^0C = 2552.7 kJ/kg

h_{v3} at 36^0C = 2567.2 kJ/kg

h_{f3} = 150.7 kJ/kg

h_{f4} = 125.7 kJ/kg

Now from energy balance

$[h_{a1} + \omega_3 h_{v1}] + m_{w2} \times h_{f2} = [h_{a3} + \omega 3 h_{v3}] + m_{w4} \times h_{f4}$

$C_p(T_1-T_3) + \omega_1 h_{v1} + m_{w2} = \omega_3 h_{v3} - [m_{w2} - (\omega_3 - \omega_1)] h_{f4}$

$1.005(32-40)+(0.0188 \times 2552.7)+(mw_2 \times 150.7) = (0.0361 \times 2567.2)+[mw_2(0.0361-0.0188)] \times 138.2$

=> $-8.04+47.99+ m_{w2}150.7 = 92.676+ m_{w2}125.7 - 2.1746$

=> $m_{w2} = 2.022$ kg of water/kg of dry air

Hence total quantity of air flow required to cool 7000 m^3/Hr (Ambient air density is 1.102 kg/m^3)

=> $m_a = [7000 \times 1000] / [1.102 \times 2.022 \times 3600]$

$= 872.64 m_3/Sec$

Required air flow per each cell $= (872.64 / 3)$

$= 290.878 m_3/Sec$

In order to achieve optimum efficiency and static pressure, the fan blade angle is to be set at 11^0 for the above calculated air flow in the case 2.

From the curves at 11^0 of the blade angle:

Fan shaft power =27 kW

Static pressure =86 Pascal

$$\text{Static efficiency} = \frac{\text{Air flow in m}^3\text{/sec x static pressure}}{\text{1000 x Fan shaft powering kW}}$$

= (290.87 x 86) / (1000 x 27)

= 92.6%

Saving in terms of money = (43.5-27) x 2.00/-per unit power (Assumed power cost is Rs.2/- per unit)

= Rs. 33.00 per hour

For a month the saving is = 33 x 24 x 30

= **Rs. 23,760 per month.**

IV. DISCUSSION OF RESULTS

- Static efficiency of the cooling fan has been improved for the reduced flow of air.

V. CONCLUSION

- From the above theoretical calculations we can conclude that the efficiency of the fan can be improved by altering the blade angle according to the air flow requirement with respect to the climatic conditions.
- During the selection of the optimum blade angle, one should take care of static pressure, as it plays a vital role in velocity recovery in cooling tower stack. As we are aware that insufficient static pressure of the fan will lead to reverse rotation of the fan and recirculation of discharge hot air.
- Here for the case 2, the blade angle selected will prone the fan to develop sufficient static pressure to avoid above mentioned problems.
- It is obvious that during winter season the power consumption of the fan is decreased down to 27 kWh from 43.5 kWh, and in terms of money it is Rs. 23,760 per month.

It is obvious that as per the fan performance curves the static efficiency of the fan will be reduced for further decrement in air flow for the same blade angle. So we have to select optimum angle of the blade corresponding to the requirement of air flow.

REFERENCES

[1] Satoshi, Y. (1996). Experimental performance of the shower cooling tower in Japan. Renewable Energy, vol. 10, no. 2-3, p. 179-183.
[2] Kröger, D.G. (1998). Air-cooled Heat Exchangers and Cooling Towers, Befell House, New York, p. 50-56.
[3] Geoffrey, F.H. (2002). Heat Exchanger Design Handbook, Begell House, London, p. 137-151.

[4] Thirapong, M., Wanchai, A., Somchai, W. (2007). An energy analysis on the performance of a counter flow wet cooling tower. Applied Thermal Engineering, vol. 27, no. 5-6, p. 910-917, DOI:10.1016/j. applthermaleng.2006.08.012.
[5] Dvoršek, M., Hočevar, M. (2011). The influence of air flow inlet region modifications on the local efficiency of natural draft cooling tower operation. Strojniški vestnik Journal of Mechanical Engineering, vol. 57, no. 10, p. 750-759, DOI:10.5545/sv-jme.2010.208.