International Journal of Scientific & Engineering Research Volume 3, Issue 6, June-2012
ISSN 2229-5518

Deterministic Study on the optimum performance of counter flow Cooling Tower

Adel Alyan Fahmy and Loula A. Shouman
Atomic Energy Authority, Reactors Department, 13759 Cairo, Egypt
adelalyan@yahoo.com

Abstract: In the current theoretical study, a parametric analysis of counter flow cooling tower is presented by optimizing the thermo-hydraulic–performance. Simple mathematical formula is correlated for estimating the optimum performance of counter flow cooling towers at different operating conditions. Based on the number of the transfer unit (NTU) and the effectiveness. The effect of the wet bulb temperature on the performance of cooling tower is studied. It is induced that there are different optimal (L/G) values for the best performance of counter flow cooling towers for each mean water temperature. It is found that the wet bulb has a great influence on NTU and effectiveness.

Index Terms— Mathematical modeling, cooling tower performance, wet bulb influence, NTU method

1. INTRODUCTION

The cooling system is considered one of the most important parts in the power plant because it extracts as little heat energy as possible from the thermodynamic cycle to the environment to ensure high thermal efficiency of the power plant. Cooling towers are the most cooling systems that widely used to reject waste heat in industrial power generation plants, refrigeration and air conditioning plants, in addition to the chemical, petrochemical and petroleum industries. The cooling tower is steady flow device that uses a combination of mass and energy transfer to cool water by exposing it as extended surface to the atmospheric air. There are several types of cooling towers; where the mechanical draft tower is considered the probably most common model. In such cooling tower a hot water to be cooled is sprayed by nozzles and flowing downward in the pack. Ambient air is drawn into the tower using fan and flows in counter or cross current direction to the water stream. Many investigations were presented investigating the thermal performance of the cooling towers. V. Steforovic', et al [1] studied experimentally the heat and mass transfer in cooling tower. N. Makkinejad [2] presented the temperature profile in counter current/co-current spray towers. It was confirmed that the exit air temperature is strongly influenced by inlet liquid temperature. A mathematical model of control system for the mechanical draft cooling tower was developed by S. P. Fisenko, et al [3], the model discussed the heat and mass transfer processes between water film and turbulent dramp air flow at quasi-state approximation. Various regimes of cooling tower performance are compared and the optimization method is proposed too. J. Khan, et al [4] presented a comprehensive design and performance evaluation study of counter flow wet cooling towers. It was found that, the sensitivity of the effectiveness and water outlet temperature with respect to inlet air wet bulb temperature and water inlet temperature is investigated for different (L/G) mass flow rate ratios. J. C. Kloppers, et al [5] evaluated the cooling tower performance by employing the Merkel- Poppe and e-NTU method of analysis at different operating and ambient conditions. Analysis of optimal spray patterns for counter flow cooling towers with

structured packing was investigated by S. C. Karanc [6]. It was concluded that nozzles developing high efficiency in thermal performance do not necessarily apply liquid with high uniformity over the top layer of the packing. F. Charagheizi, et al [7] conducted an experimental and comparative study on terms of tower characteristics (KaV/L), water to air flow ratio (L/G) and efficiency of two film type packing. The tower performance showed a decrease with an increase in the (L/G) ratio. The effect of inlet relative humidity and inlet temperature on the performance of counter flow wet cooling tower based on exergy analysis was studied by T. Muangoni, et al [8]. M. S. Soylemez [9] presented a thermo-hydraulic performance and optimization analysis, yielding simple formula for estimating the optimal performance point of counter current mechanical draft wet cooling towers. It was proved that there is different optimal (L/G) for each local elevation, barometric pressure and water temperature.

It is evident from the previous review that many works studied the performance and characteristic analysis of cooling towers, but there were few studies on optimizing performance for counter flow cooling towers. Optimization of the cooling tower performance is one of the most important problems in the theory and engineering practice of cooling of water because it permits significantly to reduction of energy consumption.

The effect of the wet bulb temperature on the performance and optimizing condition of the cooling tower is studied theoretically.

2- Mathematical Formula

The mathematical relations for the counter current flow cooling tower shown in Fig. 1 can be given as:

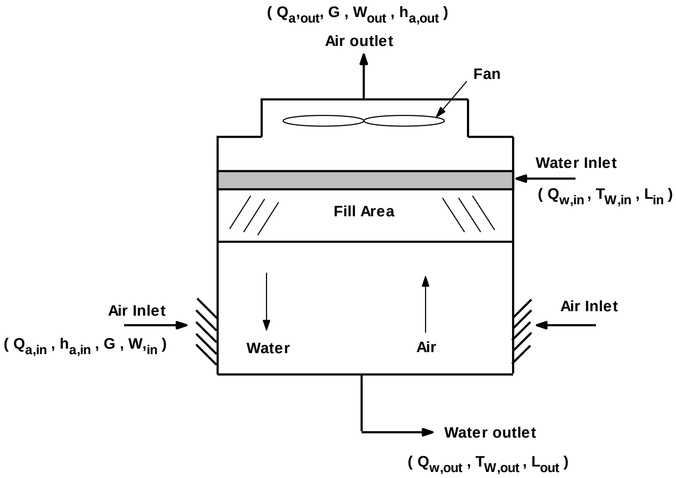

Fig. 1. Schematic figure of counter current flow cooling tower.

Heat balance for counter flow cooling tower:

$$Q_{w,in} + Q_{a,in} = Q_{w,out} + Q_{a,out} \quad (1)$$

$$L_{in} * C_w * T_{w,in} + G * h_{a,in} = L_{out} * C_w * T_{w,out} + G * h_{a,out} \quad (2)$$

Mass balance for the counter flow cooling tower:

$$L_{in} + G_{in} = L_{out} + G_{out} \quad (3)$$

The difference between entering water flow rate (Lin) and leaving water flow rate (Lout) is a loss of water due to the evaporation in the direct contact of water with air. This evaporation is transfer to the air as a difference in the water vapor content between the inlet air and exit air of cooling tower. The vapor transfer from water to air can be calculated from the following mass balance:

$$L_{in} - L_{out} = G * (W_{out} - W_{in}) \quad (4)$$

$$L_{out} = L_{in} - G * (W_{out} - W_{in}) \quad (5)$$

The number of transfer unit (NTU) can be calculated as [10]:

$$NTU = \frac{K * a * V}{L} = C_w \int_{T_{w,out}}^{T_{w,in}} \frac{dT_w}{h_w - h_a} \quad (6)$$

Figure 2 shows that the NTU is an area of multiplying the cooling range by the average of 1/ (hw - ha) at four points in the X- axis (temperature) [10].

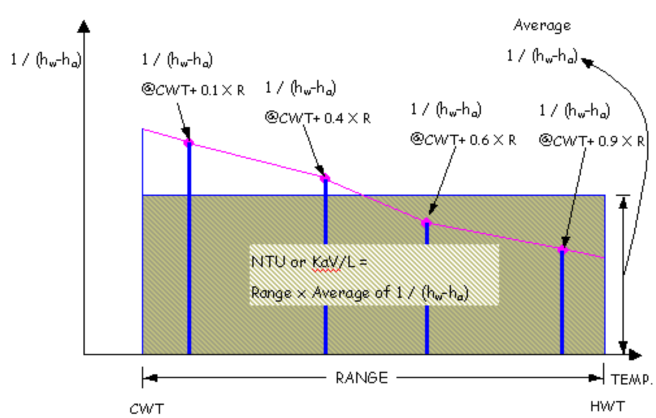

Fig.2. The average of 1/ (h w – h a) variation with the cooling range [10]

By solving the previous Eqn.(6) and the above figure, the following relation can be obtained:

$$NTU = C_w * (T_{w,in} - T_{w,out}) * \left[\left(\frac{1}{Dh_1}\right) + \left(\frac{1}{Dh_2}\right) + \left(\frac{1}{Dh_3}\right) + \left(\frac{1}{Dh_4}\right) \right] / 4 \quad (7)$$

$$Range = C_w (T_{w,in} - T_{w,out}) \quad (8)$$

Where, the value of Dh can be calculated as following [10]:
Dh1= value of (hw-ha) at temp. of CWT + (0.1 * Rang)
Dh2 = value of (hw-ha) at temp. of CWT + (0.4 * Rang)
Dh3 = value of (hw-ha) at temp. of CWT + (0.6* Rang)
Dh4 = value of (hw-ha) at temp. of CWT + (0.9* Rang)
CWT is the cold water temperature. From Eqn.(8) and the figure, the previous Eqn. (7) we can be simplified as:

$$NTU = Rang * \left[\sum \frac{1}{(h_w - h_a)} \right] / 4 \quad (9)$$

The effectiveness of the counter flow cooling tower can be calculated as follows [11]:

$$eff = \frac{1 - e^{NTU(R-1)}}{1 - R * e^{NTU(R-1)}} \quad (10)$$

The first derivative of the effectiveness with respect to R yields:

$$\frac{\partial eff}{\partial R} = 0 \rightarrow NTU_{opt} = \frac{1}{1 - R_{opt}} \quad (11)$$

And the derivative of the effectiveness with respect to NTU yields:

$$\frac{\partial eff}{\partial NTU} = 0 \rightarrow R_{opt} * [1 - e^{NTU_{opt} * (R_{opt} - 1)}] = 1 - R_{opt} * e^{NTU_{opt} * (R_{opt} - 1)} \quad (12)$$

$$R_{opt} = 1 \quad (13)$$

To determine R the following equations can be used:

$$R = \frac{m_{da} * C_{a,sat}}{m_w * C_w} = \frac{m_{da}}{m_w * C_w} * \frac{(h_{a,sat,i} - h_{a,sat,o})}{(T_{w,i} - T_{w,o})} \quad (14)$$

And,

$$R = \frac{\Delta h_{sat}}{M * C_w * \Delta T_w} = \frac{m_{da}}{m_w * C_w} * \frac{dh_{sat}}{dT_w} \quad (15)$$

The enthalpy of saturated air is formulated as follows [13]:

$$h_{sat} = C_a * T_w + h_{fg} * W_{sat} \quad (16)$$

The Ca and hfg can be treated as constant yields the following:

$$\frac{dh_{sat}}{dT_w} = C_a + h_{fg} * \frac{dW_{sat}}{dT_w} \quad (17)$$

The moisture content of the saturated air can be evaluated by the following relation:

$$W_{sat} = \frac{0.622 * P_g}{P - P_g} \quad (18)$$

The dew point temperature of water vapor has good approximation by the successive regression method as [14]:

$$P_g = e^{18.6 - \frac{5206.9}{T_w}} \quad (19)$$

By substituting Eqn.(19) into Eqn. (18), the value of Wsat can be determined by:

International Journal of Scientific & Engineering Research Volume 3, Issue 6, June-2012
ISSN 2229-5518

$$W_{sat} = \frac{0.622 * e^{18.6 - \frac{5206.9}{T_w}}}{P - e^{18.6 - \frac{5206.9}{T_w}}} \quad (20)$$

The first derivative of previous equation with respect to Tw is presented as following:

$$\frac{dW_{sat}}{dT_w} = \frac{0.622 * e^{18.6 - \frac{5206.9}{T_w}} * 5206.9 * P}{T_w^2 * \left[P - e^{18.6 - \frac{5206.9}{T_w}}\right]^2} \quad (21)$$

By substituting Eqn. (21) in Eqn.(17) the following equation can be derived:

$$\frac{dh_{sat}}{dT_w} = C_a + \frac{h_{fg} * 0.622 * e^{18.6 - \frac{5206.9}{T_w}} * 5206.9 * P}{T_w^2 * \left[P - e^{18.6 - \frac{5206.9}{T_w}}\right]^2} \quad (22)$$

From previous, it is obtained the Mopt by combining Eqns.(12), (13) and (22) as:

$$M_{opt} = \frac{1}{R_{opt} * C_w} * \frac{dh_{sat}}{dT_w} \Rightarrow R_{opt} = 1$$

$$M_{opt} = \frac{C_a}{C_w} + \frac{h_{fg} * 0.622 * e^{18.6 - \frac{5206.9}{T_w}} * 5206.9 * P}{C_w * T_w^2 * \left[P - e^{18.6 - \frac{5206.9}{T_w}}\right]^2} \quad (23)$$

3- RESULTS AND DISCUSSION

For a cooling tower operation, it is assumed that the operating pressure is the ambient pressure (101.325 kPa). The performance of cooling tower is evaluated at different L/G values. The wet bulb temperature is an important factor in performance of evaporative water cooling equipment. It is a controlling factor from the aspect of minimum cold water temperature to which water can be cooled by the evaporative method. Thus, the wet bulb temperature of the air entering the cooling tower determines operating temperature levels. So, the effect of the wet bulb temperature on the NTU is obtained at different (L/G) as shown from Fig. (3) to Fig. (9).

For different (L/G) conditions, Comparison of number of transfer unit (NTU) at different wet bulb temperature is plotted in the figures from (3) to (9). The figures show a deceasing trend of NTU as the wet bulb temperature increases. This due to the fact that the heat transfer decreases as the wet bulb temperature increases, resulting in a decrease in the difference of the inlet and outlet temperatures of the cooling tower.

Also, the effect of Decreasing of (L/G) values are studied from the design value (1.366) to (0.9562), it found that, the NTU decreases as the (L/G) value decreases due to decreasing of the heat transfer from water to air.

The effect of wet bulb temperature on the effectiveness of cooling tower is illustrated at Fig. (10). from the figure, the increasing of wet bulb temperature decreases the cooling tower effectiveness. And with increasing the mass flow rate ratio (L/G) the effectiveness of the tower increased.

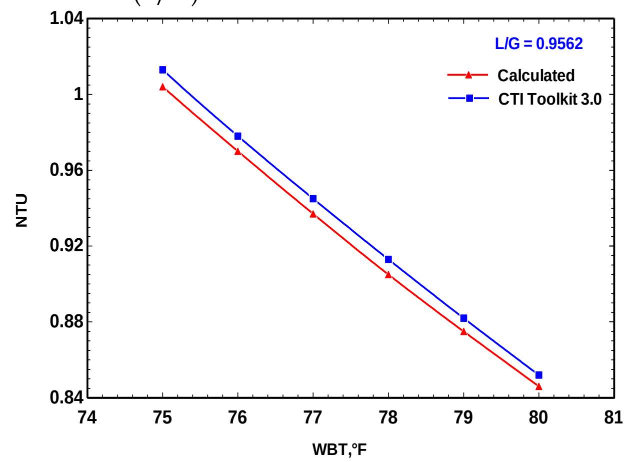

Fig.3. NTU at different wet bulb temperature and L/G=0.9562.

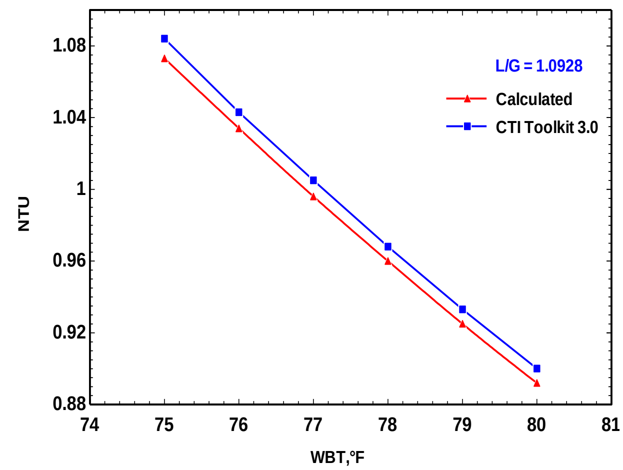

Fig. 4. NTU at different wet bulb temperature and L/G=1.0928.

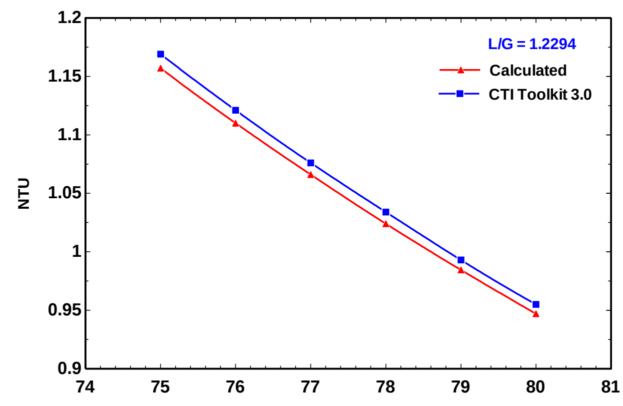

Fig. 5. NTU at different wet bulb temperature and L/G=1.2294.

International Journal of Scientific & Engineering Research Volume 3, Issue 6, June-2012
ISSN 2229-5518

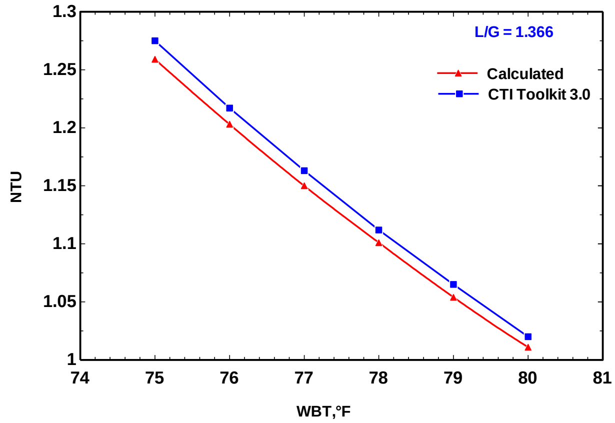

Fig.6. NTU at different wet bulb temperature and L/G=1.366.

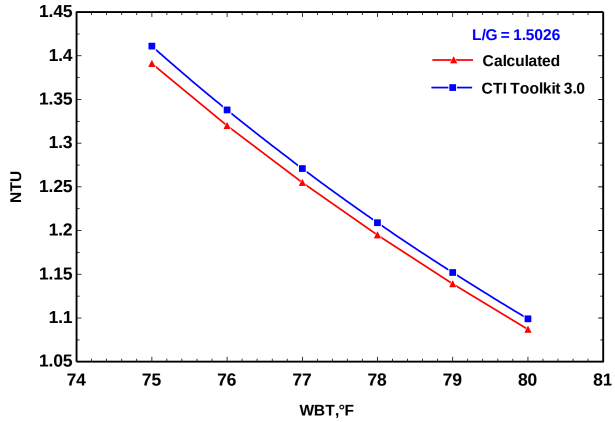

Fig. 7. NTU at different wet bulb temperature and L/G=1.5026.

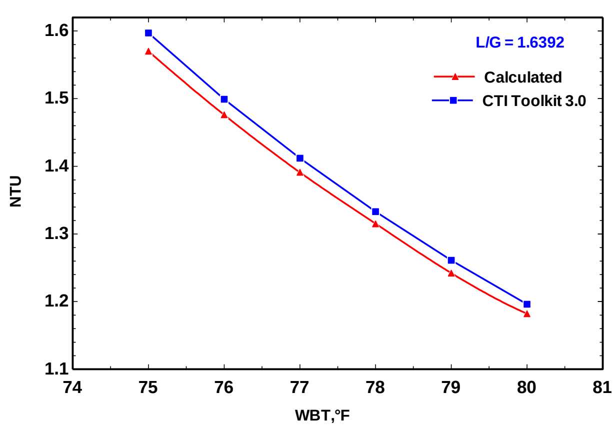

Fig. 8. NTU at different wet bulb temperature and L/G=1.6392.

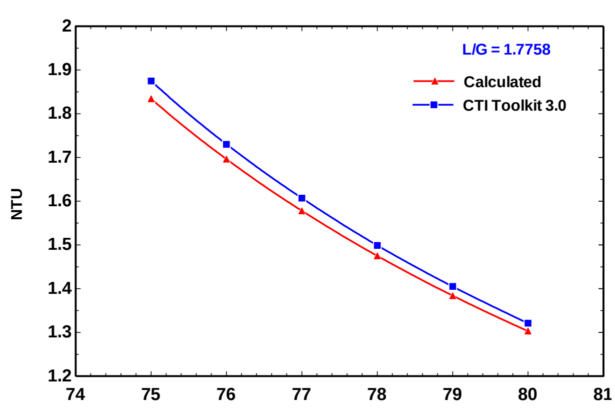

Fig. 9. NTU at different wet bulb temperature and L/G=1.7758.

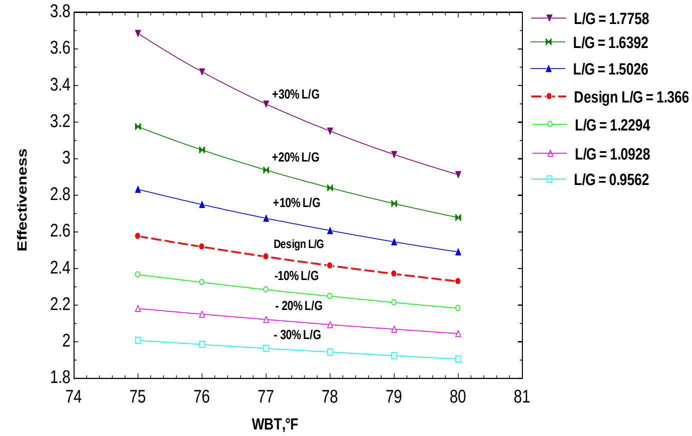

Fig. 10. Comparison Of the effectiveness at different L/G.

4-CONCLUSION

Parametric study of counter flow cooling tower is presented, by a thermo-hydraulic–performance optimization analysis. Simple mathematical formula is correlated for estimating the optimum performance of counter flow cooling towers. The effect of the wet bulb temperature on the optimum performance of cooling tower is studied. It is clear that, there are different (L/G) values for the best thermo-hydraulic performance of counter flow cooling towers for each mean temperature level. The wet bulb temperature has great influence on the NTU and effectiveness of cooling tower. The increase in the wet bulb temperature decreases both the effectiveness and NTU. To improve the performance of the cooling tower, the increase in water temperature must be combined with increase of water mass flow rate for the same air flow rate.

NOMENCLATURE:

a	Specific area per unit volume,(m2/m3)
C	Specific heat, [kJ/(kg*K)]
eff	Effectiveness of cooling tower.
G	Mass flow rate of dry air, (kg/s)
h	Enthalpy,(kJ/kg)
hfg	Latent heat of evaporation, (kJ/kg)
K	Mass transfer coefficient
L	Mass flow rate of water, (kg/s)
NTU	Number of transfer units.
m	Mass flow rate, (kg/s)
M	Mass ratio of flow rate of water to dry air, (kgw/kga),(L/G).
P	Pressure,(kPa)
R	Ratio of heat capacity rate of air to water
WBT	Wet Bulb Temperature, (F)
V	Volume of cooling tower, (m3)
W	Humidity ratio, (kgw/kga)

Subscript

a	air
da	dry air
g	dew point
in	inlet
opt	optimum
out	outlet
sat	Saturation
w	water

REFERENCE

[1] Velimir Stefonovic', Slobodan Lakovic', Nanad Rodojkovic' and Gradimir Ilic'. Experimental Study on Heat and Mass Transfer in Cooling Towers. Mechanical Engineering vol. 1, No 7, pp. 849-861, 2000.

[2] N. Makkinejad. Temperature Profile in Counter current / Co-current Spray Towers. Inter. Journal of Heat and Mass Transfer 44, 429 – 442, 2001.

[3] S. P. Fisenko, A. I. Petruchik. Toward to the Control System of Mechanical Draft Cooling Tower of Film Type. Inter. Journal of Heat and Mass Transfer 48 , 31-35, 2005.

[4] Jameel-Ur- Rehman Khan, Bilal Ahmed Qureshi, and Syed M. Zubair. A comprehensive Design and Performance Evaluation Study of Counter Flow Wet Cooling Towers. International of refrigeration 27, 914- 923, 2004.

[5] Johannes C. Kloppers and Detlevg. Kroger. Cooling Tower Performance Evaluation: Merkel, Poppe, and e - NTU Methods of Analysis. Journal of Engineering for Gas Turbines and Power. JANUARY. Vol. 127, 2005.

[6] S. C. Kranc. Optimal Spray Patterns for counter Flow Cooling Towers With Structured Packing. Applied Mathematical Modeling, 2005.

[7] Farhad Gharagheizi, Reza Hayati, and Shohreh Fatem. Experimental Study on the Performance of Mechanical Cooling Tower With Two type of Film Packing. Energy Conversion and Management, 2006.

[

[8] Thiropong Muangnei, Wanchai Asvapoo Sit Kul;, and Somchai Wongwises. Effects of Inlet Relative Humidity and Inlet Temperature on the performance of Counter Flow Wet Cooling Tower Based on Exergy Analysis. Energy Conversion and Management ,2006.

[9] M. S. Soylemez. On the Optimum Performance of Forced Draft Counter Flow Cooling Towers. Energy Conversion and Management 45, 2335- 2341, 2004.

[10] Cooling tower Technical site of Daeil Aqua Co., Ltd. For cooling tower engineers, Operators and Purchaser.

[11] Kreider JF, Rabl A. Heating and cooling of Buildings. McGraw- Hill; 1994.

[12] M. S. Soylemez. On the Optimum Sizing of Cooling Towers. Energy Conversion and Management 42, 783- 9, 2001.

[13] McQuiston FC, Parker JD.Heating Ventilating, and Air Conditioning. John Wiely&Sons Inc.; 1994

[14] Stoecker WF. Design of Thermal Systems. 3rd ed. New York; McGraw-Hill; 1989.