A Comprehensive Approach to an Optimum Design and Simulation Model of a Mechanical Draft Wet Cooling Tower

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ABSTRACT: *The present paper describes the designing of a thermally and economically optimum mechanical draft counter-flow wet cooling tower. The design model allows the use of a variety of packing materials in the cooling tower toward optimizing heat transfer. Once the optimum packing type is chosen, a compact cooling tower with low fan power consumption is modelled within the known design variables. Moreover, a simulation model of the cooling tower is developed for studying the tower's performance as the main component of a water cooling system. The model also allows the influence of the environmental conditions on the thermal efficiency of the cooling tower to be considered. The thermal performance of the cooling tower is simulated in terms of varying air and water temperatures, and of the ambient conditions. The model is tested against experimental data. The suggested design and simulation algorithms of cooling tower are computed using Visual Studio.Net 2003 (C++).*

KEY WORDS: *Cooling tower, Heat and mass transfer, Thermo-economic, Optimization, Modelling.*

INTRODUCTION

Cooling towers are commonly used for releasing the waste heat arising from industrial processes into the environment. In mechanical draft towers, which are the most commonly used of the several types of cooling towers, water enters at the top and flows downwards while air is forced upwards by a fan [1-4]. Heat ejection from the cooling tower occurs as convectional transfer between water droplets and the surrounding air, and also

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as the evaporation of a small portion of the water into the moving air. Therefore, the process involves both heat and mass transfer. The inside of the tower is packed with a material that provides large surface areas for this combination of heat and mass transfer.

Several projects have been undertaken throughout the last century toward investigating the performance of cooling towers. The basis of cooling tower operation was irst proposed by *Walker et al.* [5]. *Merkel* [6] subsequently developed a practical model by combining the differential equations of heat and mass transfer between water and air in a cooling tower. *Mohiuddin & Kant* [7,8] described a detailed procedure for the thermal design of the material for wet fill, counter and cross flow, and mechanical and natural draft cooling towers. *Braun* [9] modelled the thermal effectiveness and modified the definitions of a number of transfer units. The performance characteristics of counter flow wet cooling towers were presented by *Khan et al.* [10].

However, little attention has been focused on optimizing the design of cooling towers. In 2001, *Milosavljevic & Heikkila* [11] presented a comprehensive approach to cooling tower design. *Söylemez* [12] published a brief method for estimating cooling tower sizing based on an effectiveness model and the number of transfer units. All of these studies deal only with the heat and mass transfer in the packing zone, which was considered to be the main component of heat ejection in a cooling tower. However, *Kröger* [13] indicated that 15% of the cooling may occur in the spray zone of large cooling towers. Furthermore, 10-20% of the total heat ejection occurs in the rain zone of large-scale towers [14]. Therefore, more zones of the cooling tower must be included in investigating thermal performance and its effect on the design parameters.

The objective of the present article is to put forward a comprehensive approach to cooling tower design through thermo-economic optimization, which considers heat ejection throughout the entire tower. This design model describes the change in air temperature along the tower and the heat and mass transfer area, and allows different packing materials to be chosen for the cooling tower toward investigating heat transfer optimization.

Moreover, a simulation model of a cooling tower, as the main component of the water cooling system, is developed for predicting the properties of the water and air that exit the tower. To this end, a mathematical model is derived that accounts for the heat and mass transfer through energy and mass balance equations. The thermal behaviour of the cooling tower under various operating and environmental conditions is also studied. This allows the integration of a cooling tower to be investigated based on its performance.

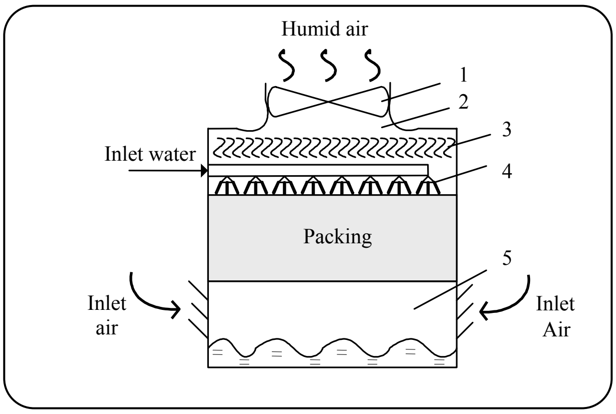

Fig. 1: Diagram of a mechanical draft wet cooling tower: (1) Fan, (2) Plenum chamber, (3) Drift eliminator, (4) Water distribution system with spraying nozzles (spray zone), (5) Rain zone.

OPTIMUM COOLING TOWER DESIGN

Heat ejection in cooling towers occurs in three zones known as the spray, packing, and rain zones. Fig. 1 shows a schematic diagram of a counter flow wet cooling tower. To optimize the design, a technique is developed through a series of iterations. The computations are conducted using the software Visual Studio.Net 2003 (C++) [15]. The water flow rate, water inlet and outlet temperatures, and the ambient air wet bulb and dry bulb temperatures are the known design parameters.

The effect of energy transfer in each region is considered on the basis of the cooling tower's characteristics. At the initial stage, heat ejection from the cooling tower is described by the following equation:

$$Q_{rej} = m_w C_{pw} (T_{w,in} - T_{w,out}) \quad (1)$$

The enthalpy and flow rate of the outlet air is then calculated with reference to tower height. The air flow rate is calculated by the following expression:

$$i_{a,out} = \frac{Q}{m_a} + i_{a,in} \quad (2)$$

The tower characteristic is as follows [13]:

$$Me = \int_{T_{w,out}}^{T_{w,in}} \frac{C_{pw} dT_w}{(i_{fw} - i)} \tag{3}$$

$$\int_{T_{w,out}}^{T_{w,in}} \frac{C_{pw} dT_w}{(i_{fw} - i)} \approx \frac{(T_{,w,in} - T_{w,out})}{4}\left[\frac{C_{Pw,1}}{\Delta i_1} + \frac{C_{Pw,2}}{\Delta i_2} + \frac{C_{Pw,3}}{\Delta i_3} + \frac{C_{Pw,4}}{\Delta i_4}\right] \approx \frac{C_{Pwm}(T_{w,in} - T_{w,out})}{4}\left[\frac{1}{\Delta i_1} + \frac{1}{\Delta i_2} + \frac{1}{\Delta i_3} + \frac{1}{\Delta i_4}\right] \tag{4}$$

where the enthalpy difference is given by:

$$\Delta i_{(i)} = i_{fw(i)} - i_{(i)} \tag{5}$$

The height of fill zone is computed via the expression:

$$\frac{h_{dfi} a_{fi}}{G_w} = a'_f L_{fi}^{b'_f} \left(\frac{G_a}{G_w}\right)^{c'_f} \tag{6}$$

where a'_f, b'_f and c'_f are the packing constants that are specific for different types of packing material [13]. The design incorporates a selection of packing materials with high transfer coefficients. In other words, an optimum fill type can be selected toward achieving a compact cooling tower design with low fan power consumption.

The computation is then continued toward determining the ideal frontal area, fan power and fan casing area, and the number of packing decks. It is assumed that the cooling tower frontal area and cross-sectional area will be approximately equal. If the design is for a rectangular cooling tower, the frontal area is given by [13]:

$$A_{fr} = L_i \times W_i \tag{7}$$

The power of the air fan is a function of the air flow rate, which is determined by multiplying the pressure drop with the air flow rate [12]:

$$P_f = \frac{m_a^3 \left(6.5 + K_{el} + 2\left(\frac{A_{fr}}{A_{fan}}\right)^2\right)}{2\rho_a A_{fr}^2 \eta_{fan} \eta_{motor}} \tag{8}$$

The height of the fan diffuser is given by [13]:

$$L_{Dif} = 0.4 d_{fan} \tag{9}$$

The relationship between the cooling tower height and width is given by [14]:

$$\frac{H}{W_i} = 8 \frac{L_{rz} W_i}{A_{fan}} \tag{10}$$

The rain zone is required in conventional cooling towers so as to permit uniform air flow into the fill. However, this zone is a thermally inefficient portion of the cooling tower. The droplets in the rain zone are formed from water dripping from the sheets of packing material. Therefore, the radii of the droplets are quite large compared to those of the spray zone [16]. The heights of the spray and rain zones in a cooling tower are expressed as:

$$\frac{h_{dsp} a_{sp} L_{sp}}{G_w} = a'_s L_{sp} \left(\frac{G_a}{G_w}\right)^{b'_s} \tag{11}$$

$$\frac{h_{drz} a_{rz} L_{rz}}{G_w} = 3.6\left(\frac{\frac{p_{atm}}{R_v T_a}}{\rho_w}\right)\left(\frac{D}{\upsilon_{a,in} d}\right)\left(\frac{L_{rz}}{d}\right) Sc^{0.33} \times \left[\frac{\ln\left(\frac{w_s + 0.622}{w + 0.622}\right)}{w_s - w}\right] \times \{5.01334 a_1 \rho_a - 192121.7 a_2 \mu_a - 2.57724 + 23.61842 \times [0.2539(a_3 \upsilon_{a,in})^{1.67} + 0.18] \times [0.83666(a_4 L_{rz})^{-0.5299} + 0.42] \times [43.0696(a_4 d)^{0.7947} + 0.52]\} \tag{12}$$

where the a_i coefficients represent combinations of g, ρ_w, σ_w. These values are given by Eqs. (13) to (16) [17].

$$a_1 = 998/\rho_w \tag{13}$$

$$a_2 = 3.06 \times 10^{-6} \left[\rho_w^4 g^9 / \sigma_w\right]^{0.25} \tag{14}$$

$$a_3 = 73.298 \left[\sigma_w^3 g^5 / \rho_w^3\right]^{0.25} \tag{15}$$

$$a_4 = 6.122 \left[\sigma_w g / \rho_w^3\right]^{0.25} \tag{16}$$

Since the heat and mass transfer occurs throughout the entire tower, the relation of cooling tower characteristic is applied to the entire region between the inlet of the rain zone and the outlet of the spray zone [17].

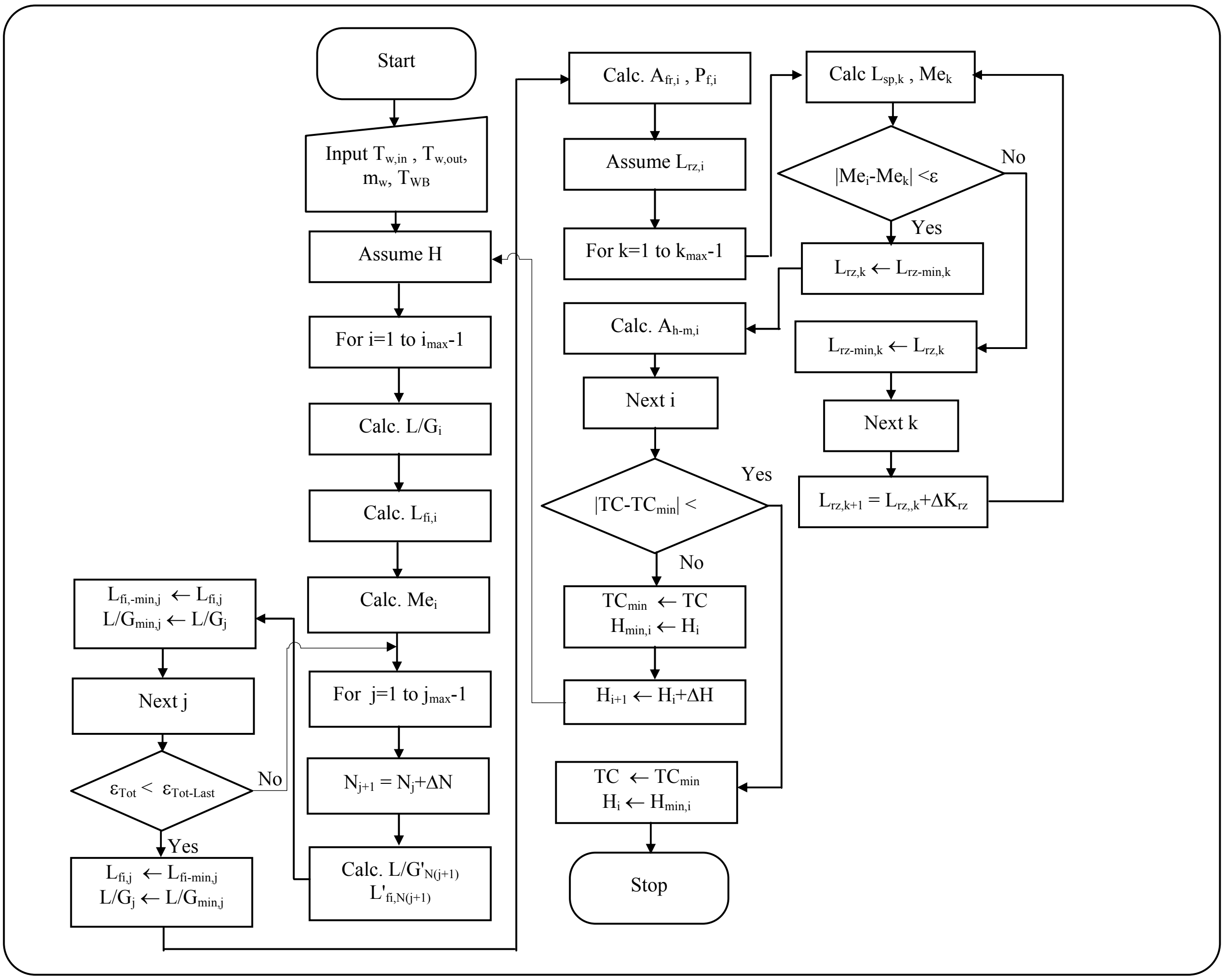

Fig. 2: Optimum cooling tower design flowchart.

$$\frac{h_{dfi}a_{fi}L_{fi}}{G_w}+\frac{h_{drz}a_{rz}L_{rz}}{G_w}+\frac{h_{dsp}a_{sp}L_{sp}}{G_w}=\int_{T_{w,out}}^{T_{w,in}}\frac{C_{pw}dT_w}{(i_{fw}-i)} \qquad (17)$$

The h_d terms in the above equations are the heat transfer coefficients [14]. The total height of the cooling tower is [17]:

$$H = L_{rz} + L_{fi} + L_{sp} + L_{Dif} + L_{pl} \qquad (18)$$

where the L_{pl} is the plenum chamber height. The plenum chamber is the enclosed space between the drift eliminator and the fan.

The heat and mass transfer area of the entire tower is given by [13]:

$$A_{h-m} = A_{fr}RyH \qquad (19)$$

The operating cost and the capital cost of the cooling tower have different effects on the overall cost of cooling. Therefore, the problem becomes one of designing an optimal cooling tower. The total cost of a cooling tower as an objective function is expressed by [12]:

$$TC = C_i\left(\frac{A_{h-m}}{Ry}\right)+\frac{\left[C_{elec}E_f m_a^3 Ry^2 H^2 S\left(6.5+K_{el}+2\left(\frac{A_{h-m}}{RyHA_{fan}}\right)^2\right)\right]}{2\rho_a A_{h-m}^2\eta_{fan}\eta_{motor}}+A_{ic} \qquad (20)$$

The iteration ends when the optimum heat and mass transfer area is achieved at an optimum cooling tower height and minimum cost. The computational procedure is outlined in Fig. 2.

The results obtained from the optimum cooling tower design are compared to a sample tower built to the actual size of the designed tower. The following specifications are considered for the cooling tower design:

Inlet water temperature is 45°C; outlet water temperature is 33 °C; inlet water flow rate is 2.57 kg/s; air temperature is 30 °C; wet bulb temperature is 25 °C; electricity cost is 0.1 $/kWh; operating time period is 8600 h/yr; fan efficiency is 70%; motor efficiency is 80%; eliminator characteristic is 115 m^{-1}; effective droplet diameters at rain zone are 6.2 mm.

The cooling tower design specifications are presented in Table 1.

A comparison between the cooling tower design and the sample tower illustrates that the optimum cooling tower area, achieved through specific design parameters, is 146.46 m^2 whereas the actual available area is about 236.67 m^2. This indicates that the sample cooling tower contains approximately 38% extra heat and mass transfer area. The height of each zone of the optimum tower is presented in Table 2.

Three type of filling material were tested in the sample cooling tower: Ecodyne-shaped material, Toschi asbestos-free fibre cement, and corrugated fill. The performance of the cooling tower is influenced by the heat and mass transfer area, as illustrated in Fig. 3. This figure demonstrates that the cooling tower's performance increases as the heat and mass transfer area is increased at constant tower height. It has been noted that the heat and mass transfer area can be increased by using rougher packing cells.

Fig. 4 shows the effects of increasing the heat and mass transfer area on the air outlet conditions. The results show that the outlet air temperature, and therefore the outlet humidity ratio, is increased by increasing the heat and mass transfer area of the tower.

Fig. 5 shows the water temperature profile through cooling tower at different fill materials. It reveals that to achieve the desired cooling water temperature of 33°C, different heat and mass transfer areas are available. Therefore, for accomplishing the optimum cooling tower design, economical consideration is necessary to avoid extra unnecessary costs.

Economical considerations reveal that by increasing the heat and mass transfer areas, the capital cost of the cooling tower increases whereas the energy cost decreases.

Table 1: Comparison of the areas of the actual and optimized cooling towers.

Design Parameter	A_{fr} (m^2)	A_{h-m} (m^2)
Actual cooling Tower	0.98	236.67
Optimum Design	0.75	146.46

Table 2: Optimum cooling tower height.

	L_{sp} (m)	L_{fi} (m)	L_{rz} (m)	L_{Dif} (m)
Optimum Design	0.38	0.58	0.49	0.2

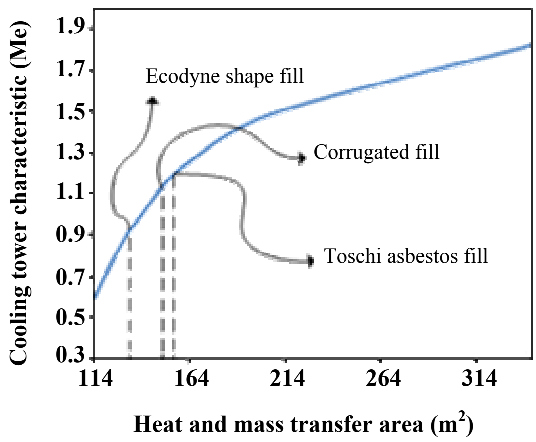

Fig. 3: The influence of heat and mass transfer area on cooling tower performance.

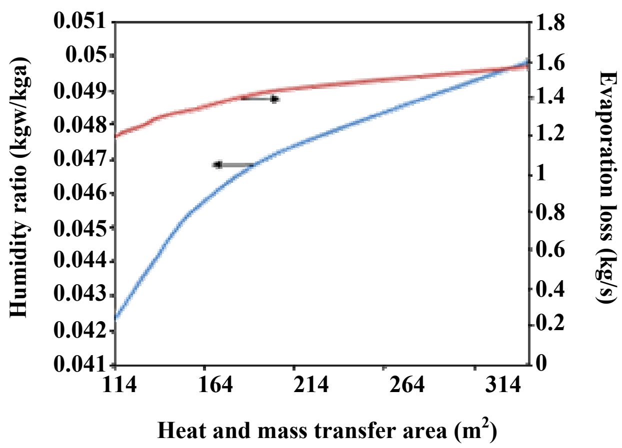

Fig. 4: Influence of heat and mass transfer area on outlet humidity and evaporation loss.

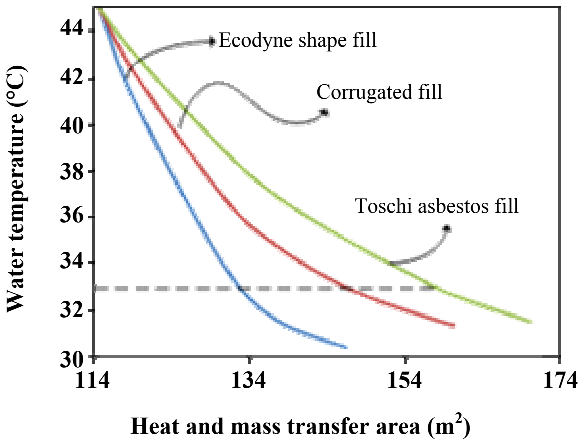

Fig. 5: Water temperature profile at different fill materials.

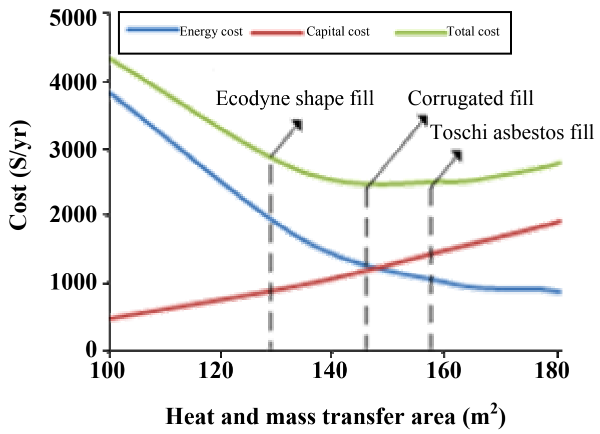

Fig. 6: Variations in total cost relative to heat and mass transfer area.

This introduces a trade-off between capital and energy costs which leads to minimizing the total annual cost. The experiments with the sample cooling tower demonstrate that the different packing materials entail different costs. Therefore, the optimum heat and mass transfer area that is achieved through the minimum cost will always optimize a cooling tower's efficiency. Fig. 6 illustrates the variations in the total cost of a cooling tower relative to various heat and mass transfer areas. The results show that the cost of the designed cooling tower is 2.74 k$/yr, whereas that of the sample cooling tower is 3.52 k$/yr. This reveals a 22% cost reduction compared with the existing cooling tower design.

The cooling tower is the main component of a water cooling system. Thus, the performance of a water cooling system is significantly influenced by the tower's performance. The cooling tower performance is also a function of environmental conditions that vary throughout the year. It is therefore important to accurately predict variations in the performance of cooling towers for periods in which the ambient conditions will change [18]. A cooling tower simulation model is therefore crucial for studying thermal performance in terms of varying air and water temperatures and environmental conditions.

MATHEMATICAL MODEL OF COOLING TOWER

The total enthalpy transfer at the air-water interface consists of an enthalpy transfer associated with the mass transfer due to the difference in vapour concentration, and a heat transfer due to the difference in temperature [13].

The heat and mass transfer between the air and water within the cooling tower's packing material is illustrated in Fig. 7. The following mathematical model entails the following assumptions:

1- Heat and mass transfer through the tower wall to the environment is negligible.

2- The flow rates of dry air and water are constant.

3- Temperatures of water and air are uniform at any cross section.

4- Temperature has no influence on the transfer coefficients.

5- Water loss by drift is negligible.

6- Interface areas for heat and mass transfer are equal.

The total heat transfer is expressed as [19]:

$$dQ = dQ_e + dQ_C \tag{21}$$

The evaporative enthalpy transfer is:

$$dQ_e = i_v \frac{dm_w}{dH} dH = i_v h_d (w_{sw} - w) dA \tag{22}$$

The convective transfer of sensible heat at the interface is given by:

$$dQ_C = h_c (T_w - T_a) dA \tag{23}$$

At steady state conditions, the energy balance between air and water, including evaporation, is given by the following relation:

$$m_a \frac{di}{dH} = m_w C_{pw} \frac{dT_w}{dH} + T_w C_{pw} \frac{dm_w}{dH} \tag{24}$$

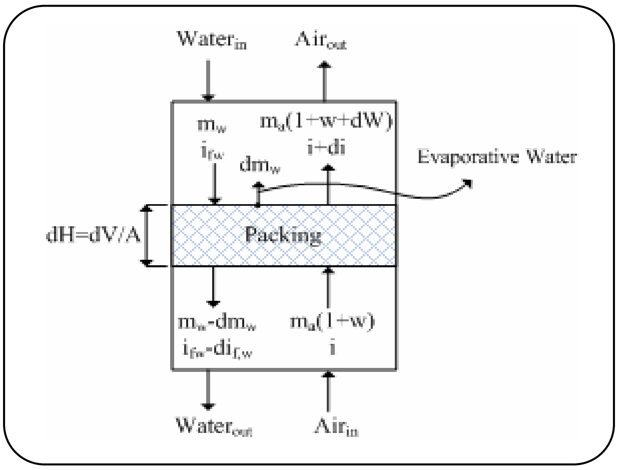

Fig. 7: Control volume of counter flow tower.

The mass balance of the control volume is written as:

$$m_a(1+w)-m_w = \quad (25)$$
$$m_a\left(1+w+\frac{dw}{dH}dH\right)-\left(m_w+\frac{dm_w}{dH}dH\right)$$

An amount of water dm_{evap} is evaporated in the control volume. At the water surface, evaporation can be expressed as [20]:

$$\frac{dm_{evap}}{dH}=\frac{m_a dw}{dH} \quad (26)$$

The temperature difference between water and air is:

$$T_w-T_a=[(i_{fw}-i)-(w_{sw}-w)i_v]/C_{pm} \quad (27)$$

By combining Eqs. (22)-(27) and substituting the Lewis factor, $Le_f=h_c/(C_{pm}h_d)$ [21], the enthalpy change along the tower can be written as:

$$\frac{di}{dH}=\frac{h_d dA}{m_a dH}[Le_f(i_{fw}-i)+(1-Le_f)i_v(w_{sw}-w)] \quad (28)$$

Therefore, the corresponding change of water temperature with tower height, taking into consideration the tower characteristic ($Me=h_dA/m_w$), is defined as:

$$\frac{dT_w}{dH}=\frac{Me}{C_{pw}H}[Le_f(i_{fw}-i)+(1-Le_f)i_v(w_{sw}-w)] \quad (29)$$

The humidity change along the cooling tower is expressed as [22]:

$$\frac{dw}{dH}=\frac{Me}{C_{pw}H}(w_{sw}-w) \quad (30)$$

An iterative calculation is used to achieve the outlet properties of air and water from the cooling tower. The computational procedure is outlined in Fig. 8.

A variety of packing materials can be used in the cooling tower simulation toward investigating their influence on a tower's performance. The use of different packing materials affects the heat and mass transfer area, the related coefficients, and hence the cooling tower performance [23]. This correlation is given by Eq. (4). The cooling tower model also includes the heat transfer in the rain and spray zones (Eqs. (8)-(13)). The cooling tower liquid to gas ratio is given by [13]:

$$\frac{m_w}{m_a}=\frac{m_{w,in}}{m_a}\left[1-\frac{m_a}{m_{w,in}}(w_{out}-w)\right] \quad (31)$$

The cooling tower's effectiveness (ε), which is defined as the ratio of actual energy to the maximum possible energy transfer, is given by:

$$\varepsilon=\frac{i_{out}-i_{in}}{i_{fw,in}-i_{in}} \quad (32)$$

PERFORMANCE SIMULATION

We tested our method by applying the experimental data of *Simpson & Sherwood* [24] to the cooling tower model. The most comparable results are those of the exit water temperatures and the exit wet-bulb temperatures. The simulation results are presented in Table 3.

These results suggest that the proposed model is accurate based on the limited amount of available experimental data. Therefore, the model can be used to predict the properties of the exit water and air from the tower for a given design and operating conditions.

It is usually important to supply cooling water at a specific temperature. However, the performance of a cooling tower will vary with changes in environmental conditions. This will affect the cooling water outlet temperature. Investigating the thermal behaviour of the cooling tower at different environmental conditions allows the prediction of a tower's performance at different atmospheric conditions. Fig. 8 shows the effect of wet bulb temperature on water outlet temperature and evaporation loss for different liquid to gas ratios. The plots are drawn using the following set of input data: P_{atm}= 101325 Pa; $T_{w,in}$= 41 °C; m_a= 32.44 kg/s, H= 2.51 m.

Table 3: Comparison of the cooling tower model results.

Experimental Data	1	2	3	4	5
Water Inlet Temperature (ºC)	41.44	38.78	38.78	34.5	28.72
Water Outlet Temperature (ºC)	26	29.33	29.33	26.22	24.22
Air Inlet Dry Bulb Temperature (ºC)	34.11	35	35	30.5	29
Exit Wet Bulb Temperature (ºC)	30.72	32.89	32.89	29.94	26.17
Mass flow rate of water to air	0.65	0.79	0.80	1.06	1.06
Heat Rejection (kW)	48.54	39.72	39.72	43.47	23.62
Model Output Result					
Water Outlet Temperature (ºC)	26.03	29.3	29.29	26.26	24.24
Exit Wet Bulb Temperature (ºC)	30.69	32.84	32.93	29.90	26.13
Heat Rejection (kW)	48.45	39.84	39.88	43.26	23.52
Result Error					
Water Outlet Temperature Error (%)	0.11	-0.10	-0.13	0.15	0.08
Exit Wet Bulb Temperature (%)	-0.09	-0.15	0.12	-0.13	0.15
Heat Rejection (%)	-0.18	0.30	0.40	-0.48	-0.42

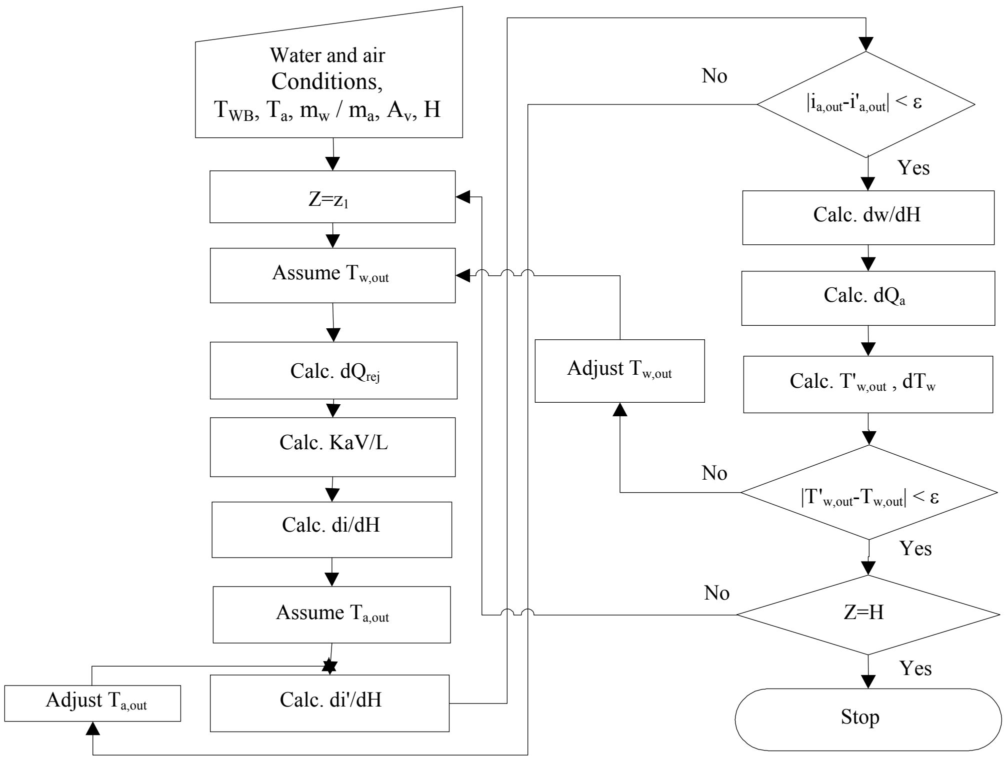

Fig. 8: Computational procedure of cooling tower simulation model.

As shown in Fig. 9, the water outlet temperature increases when the environment wet bulb temperature is increased. The outlet conditions, flow rate, and temperature of the water are affected by evaporation. Fig. 9 also demonstrates that reducing the wet bulb temperature results in increasing evaporation loss. When the wet bulb temperature is 16 ºC and the liquid to gas ratio of tower is 1.5, the tower can supply cooling water at a temperature of 32.8 ºC. However, with an increase of 2.3 ºC in the environment wet bulb temperature (18.3 ºC), the temperature of the cooling water from the tower increases to 34.25 ºC. This affects the performance of the cooling system. Therefore, to provide cooling water at a temperature of 32.8 ºC under the new environmental conditions, the liquid to gas ratio needs to be decreased to 1.1.

The cooling tower approach is defined as the difference between the water outlet temperature and the wet bulb temperature [25]. Fig. 10 shows the isothermal cooling line of the cooling system outlet temperature. The graphs are drawn for different approach values of 5 ºC, 8 ºC, and 11 ºC. It is shown that if the temperature of the cooling water outlet remains constant, the water inlet temperature needs to be reduced when the water flow rate increases. Moreover, decreasing water flow rate and increasing water inlet temperature simultaneously result in reducing the water outlet temperature.

The cooling tower's heat ejection versus water inlet flow rate at different inlet temperatures is shown in Fig. 11. It demonstrates that when the water flow rate is decreased by 4 kg/s, the heat removal accomplished by the tower increases by 74 kW for a water inlet temperature of 45 ºC. The rate of heat ejection continues to increase at higher water inlet temperatures. In other words, when the inlet cooling water has a high temperature and low flow rate, the tower ejects more heat from the water.

Fig. 12 shows the variation of evaporation rate versus heat removal. The water flow rate is set at 16.58 kg/s. It can be seen from the figure that the evaporation rate increases as heat removal increases, and that a constant heat ejection value does not necessarily ensure a fixed evaporation rate. The amount of evaporation depends on the air flow rate, the humidity of the inlet air, and the humidity of the cooling tower outlet air. The exit air humidity is interconnected with the water temperature and the transfer area of the packing material

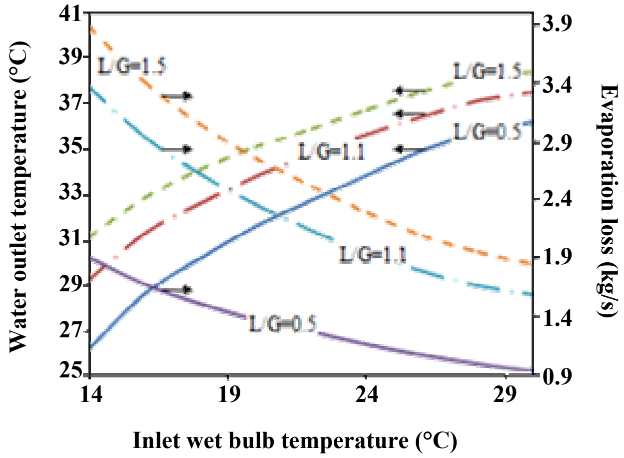

Fig. 9: Water outlet temperature and evaporation rate profile versus wet bulb temperature.

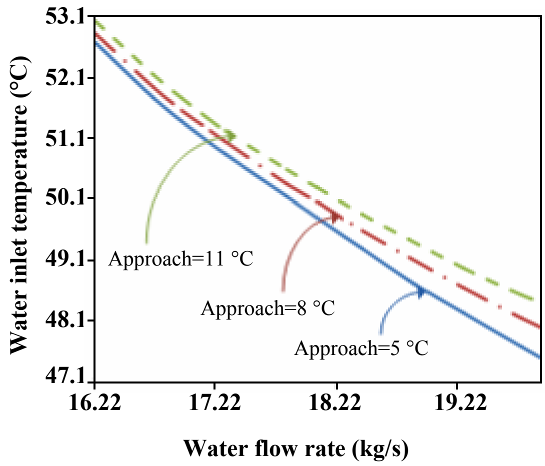

Fig. 10: Isothermal cooling water supply at different approaches.

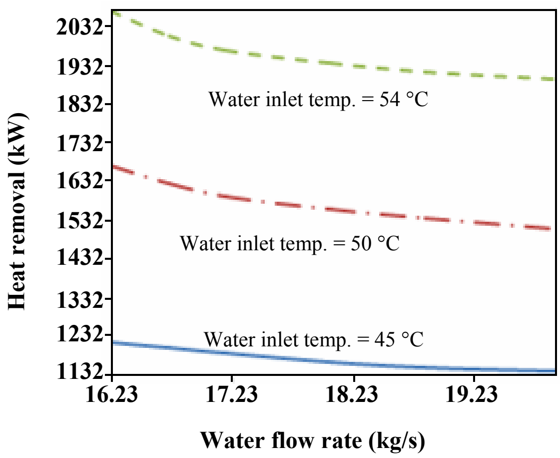

Fig. 11: Heat removal versus inlet water flow rate at different inlet temperatures.

Fig. 13 shows the variation of the tower characteristic Me, with the inlet water temperature for liquid to gas ratios of 0.5, 1.1, and 1.5. The figure demonstrates that this tower characteristic decreases with an increase of L/G. In other words, the tower Me is higher for the lowest L/G values, corresponding to the lower water flow rate, which results the best cooling.

Fig. 14 shows cooling tower performance in terms of effectiveness. A high degree of tower effectiveness corresponds to better cooling performance and higher heat removal. It can be seen in Figure 14 that when the inlet cooling water has a high temperature and low flow rate, the effectiveness of the cooling tower increases. This confirms the experimental results of Bedekar et al. [26].

CONCLUSIONS

A comprehensive approach to the designing and simulating of optimum models for wet cooling towers is developed. The procedure allows the systematic exploration of thermo-economical design optimization. The relations between the tower's characteristics and the design parameters are studied. This approach considers heat ejection throughout the entire cooling tower, including the spray, fill, and rain zones. The design presented here accommodates a variety of packing materials for investigating the optimization of heat transfer. The validity of the optimization formulation is confirmed by a sample problem.

A tower simulation model is developed for testing the designed tower's cooling system performance. The cooling tower is simulated through a theoretical analysis and a computational model based on conservation equations. Our simulation model can be used for predicting the physical properties of the moving air and water inside the cooling tower, and considers the cooling tower's packing material. Moreover, the factors that affect the performance of the counter-flow wet cooling tower are studied. These factors are the diameter of the water droplets, the liquid to gas ratio, the inlet water temperature, the wet bulb temperature of the surrounding air, the air velocity inside the tower, and the height and frontal area of the cooling tower. The model is tested against experimental data, the results of which suggest that the simulation is quite accurate. Furthermore, the influence of atmospheric conditions on thermal behaviour of the cooling tower is also studied. The thermal performance and efficiency

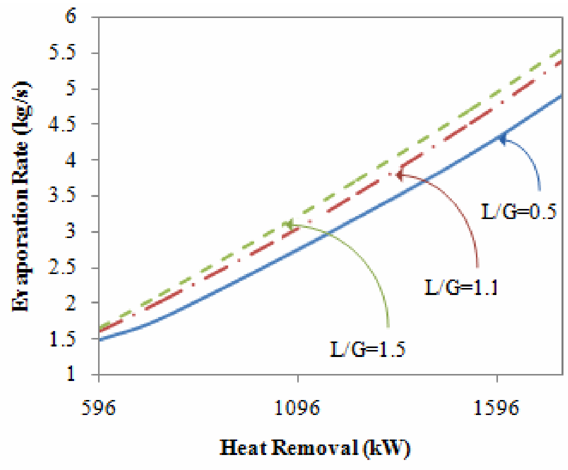

Fig. 12: Variation of evaporation rate with heat removal at different liquid to gas ratios.

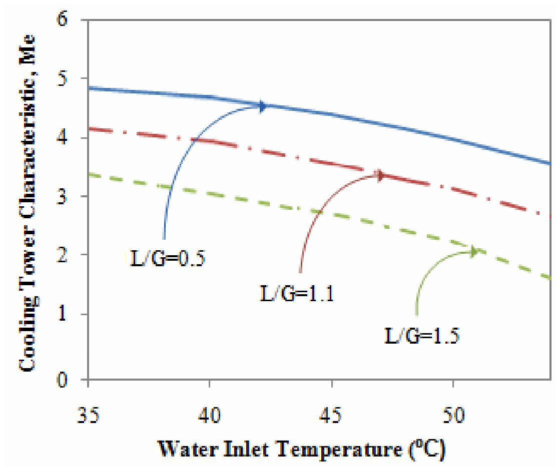

Fig. 13: Variation of tower characteristic Me with water inlet temperature.

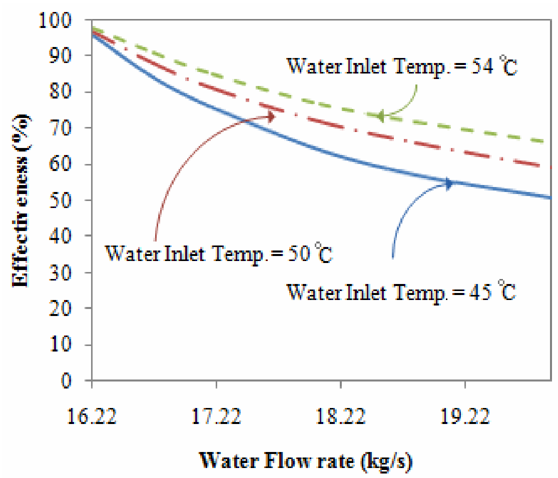

Fig. 14: Effects of inlet water flow rate and temperature on tower effectiveness.

of the cooling tower are investigated in terms of varying operational conditions in the presented simulation model. Programming in Visual Studio.Net 2003 (C++) is developed toward obtaining computational results for the optimum design and simulation models.

Nomenclature

a	Air-water interface area per unit volume of tower, (m^2/m^3)
a'_f, b'_f, c'_f	Fill zone constants
a'_s, b'_s	Spray zone constants
a_i	Rain zone constants
A_{fan}	fan casing area, (m^2)
A_{fr}	Tower frontal area, (m^2)
A_{h-m}	Heat and mass transfer area, (m^2)
A_{ic}	Area independent initial cost, ($)
C_{elec}	Electricity cost, ($/kWh)
C_i	Initial cost of tower per unit volume, ($/$m^3$)
C_{pw}	Specific heat of water at constant pressure, (kJ/kg K)
D	Diffusion coefficient, (m^2/s)
d	Droplet diameter, (m)
d_{fan}	Fan diameter, (m)
E_f	Economic factor
G	Mass velocity, (kg/sm^2)
g	Gravitational acceleration, (m/s^2)
i	Enthalpy, (kJ/kg)
i_{fw}	Enthalpy of saturated water evaluated as T_w, (kJ/kg)
i_v	Enthalpy of air-water vapour mixture, (kJ/kg)
h_c	Convection heat transfer coefficient of , (W/m^{2o}C)
h_d	Mass transfer coefficient of , (kg/m^2s)
H	Cooling tower height, (m)
K_{el}	Eliminator coefficient
L	Length, (m)
L_i	Cooling tower length, (m)
Le_f	Lewis factor
Me	Cooling tower characteristic
m	Flow rate, (kg/s)
N_{deck}	Number of decks
P_{atm}	Atmospheric pressure, (Pa)
P_f	Fan power, (hp)
Q	Heat ejection rate, (kW)
R_v	Gas constant, (J/kg ºC)
Ry	Eliminator characteristic, (m^{-1})
S	Annual total operation time, (h)
Sc	Schmidt number
T	Temperature, (ºC)
TC	Total cost, ($/yr)
V	Tower volume, (m^3)
w	Humidity ratio, (kgw/kga)
w_s	Saturated humidity ratio, (kgw/kga)
W_i	Cooling tower width, (m)

Subscripts

a	Air
c	Conductive
Dif	Diffuser
e	Evaporative
evap	Evaporation
env	Environment
fi	Fill zone
in	Inlet
out	Outlet
pl	Plenum chamber
rej	Rejection
rz	Rain zone
sp	Spray zone
v	Vapour
w	Water
WB	Wet bulb

Greek Letters

∇	Velocity, (m/s)
ρ	Density, (kg/m^3)
σ	Surface tension, (N/m)
η	Efficiency
μ	Viscosity, (kg/ms)
ε	Effectiveness

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