Journal of Applied Sciences

ISSN 1812-5654

Journal of Applied Sciences 9 (3): 561-566, 2009
ISSN 1812-5654

A New Algorithm for Optimum Design of Mechanical Draft Wet Cooling Towers

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Abstract: The present study 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. The design model incorporated the cooling tower factors to achieve the optimum design. The main factors included: the diameter of the water droplets, the liquid to gas mass ratio, the height of rain zone, packing zone and spray zone, the air and water velocity inside the tower and the frontal area. Once the optimum packing type is chosen, a compact cooling tower with low fan power consumption is modelled within the known design variables. The optimization model is validated against a sample problem. The suggested design algorithms of cooling tower are computed using Visual Studio.Net 2003 (C++).

Key words: Packing materials, optimization, heat transfer

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 (Lemouari *et al*., 2007; Jin *et al*., 2007; Panjeshahi and Ataei, 2008; Smith, 2005). Heat ejection from the cooling tower occurs as convectional transfer between water droplets and the surrounding air and also 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 first proposed by Walker *et al*. (1923). Mesarovic (1973) subsequently presented a computer program for thermal and hydraulic design of cooling towers. Mohiuddin and Kant (1996a, b) 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. Ludwig (1999) performed process design for chemical and petrochemical plants. Khan and Zubair (2001) presented an improved design and rating analysis of counter flow wet cooling towers. The performance characteristics of counter flow wet cooling towers were presented by Khan *et al*. (2003).

However, little attention has been focused on optimizing the design of cooling towers. In 2001, Milosavljevic and Heikkila (2001) presented a comprehensive approach to cooling tower design. Söylemez (2001) 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 (2004) 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 (Qureshi and Zubair, 2006). 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 study 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.

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MATERIALS AND METHODS

Heat ejection in cooling towers occurs in three zones known as the spray, packing and rain zones. Figure 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++). 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 Eq. 1:

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

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:

$$i_{a,out} = \frac{Q}{m_a} + i_{a,in} \tag{2}$$

The tower characteristic (Me number) and fill height are then computed as follows (Kröger, 2004):

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

$$\frac{h_{dfi} a_{fi}}{G_w} = a'_f L_{fi}^{b'_f} \left(\frac{G_a}{G_w}\right)^{c'_f} \tag{4}$$

where, a'_f, b'_f and c'_f are the packing constants that are specific for different types of packing material (Kröger, 2004). The design incorporates a selection of packing

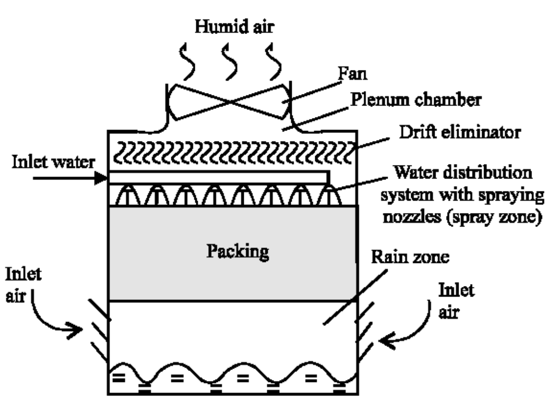

Fig. 1: Diagram of a mechanical draft wet cooling tower

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 No. 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 Kröger (2004):

$$A_{fr} = L_i \times W_i \tag{5}$$

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 (Söylemez, 2001):

$$P_f = \frac{m_a^3 \left(6.5 + K_{el} + 2\left(\frac{A_{fr}}{A_{fan}}\right)^2\right)}{2\rho_a A_{fr}^2 \eta_{fan} \eta_{motor}} \tag{6}$$

The height of the fan diffuser is given by Kröger (2004):

$$L_{dif} = 0.4\, d_{fan} \tag{7}$$

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 (Qureshi and Zubair, 2006). The heights of the spray and rain zones in a cooling tower are expressed as:

$$\frac{h_{dsp} a_{sp} L_{sp}}{G_w} = a'_s L_{sp} \left(\frac{G_a}{G_w}\right)^{b'_s} \tag{8}$$

$$\begin{aligned} \frac{h_{drz} a_{rz} L_{rz}}{G_w} = {} & 3.6 \left(\frac{\frac{P_{atm}}{R_v T_a}}{\rho_w}\right) \left(\frac{D}{\vartheta_{a,in} d}\right) \left(\frac{L_{rz}}{d}\right) Sc^{0.33} \left[\frac{\ln\left(\frac{w_s + 0.622}{w + 0.622}\right)}{w_s - w}\right] \\ & \times (5.01334 a_1 \rho_a - 192121.7 a_2 \mu_a - 2.57724 + 23.61842 \times [0.2539(a_3 \vartheta_{a,in})^{1.67} \\ & + 0.18] \times [0.83666(a_4 L_{rz})^{-0.5299} + 0.42] \times [43.0696(a_4 d)^{0.7947} + 0.52]) \end{aligned} \tag{9}$$

where, the a_i coefficients represent combinations of g, ρ_w, σ_w. These values are given by Eq. 10-13.

$$a_1 = 998/\rho_w \tag{10}$$

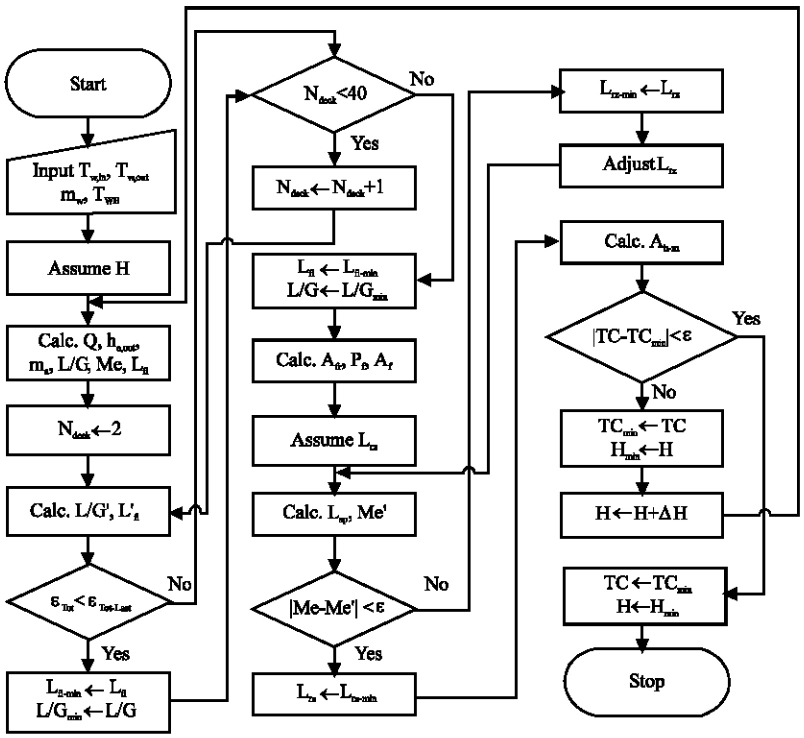

Fig. 2: Optimum cooling tower design flowchart

$$a_2 = 3.06\times10^{-6}\left[\rho_w^4 g^9 / \sigma_w\right]^{0.25} \tag{11}$$

$$a_3 = 73.298\left[\sigma_w^3 g^5 / \rho_w^3\right]^{0.25} \tag{12}$$

$$a_4 = 6.122\left[\sigma_w g / \rho_w\right]^{0.25} \tag{13}$$

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.

$$\frac{h_{dfi}a_{fi}L_{fi}}{G_w} + \frac{h_{drz}a_{rz}L_{rz}}{G_w} + \frac{h_{dsp}a_{sp}L_{sp}}{G_w} = \int_{T_{w,out}}^{T_{w,in}} \frac{C_{pw}dT_w}{(i_{fw} - i)} \tag{14}$$

The h_d terms in the above equations are the heat transfer coefficients (Deng and Tan, 2003). The total height of the cooling tower is:

$$H = L_{rz} + L_{fi} + L_{sp} + L_{Dif} + L_{pl} \tag{15}$$

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 Kröger (2004):

$$A_{h-m} = A_{fr}R_yH \tag{16}$$

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 Söylemez (2001):

$$TC = C_i\left(\frac{A_{h-m}}{Ry}\right) + \frac{\left[C_{elec}E_f m_a^3 Ry^2 H^2 S(6.5 + K_{el} + 2(\frac{A_{h-m}}{RyHA_{fan}})^2)\right]}{2\rho_a A_{h-m}^2 \eta_{fan}\eta_{motor}} + A_{ic} \tag{17}$$

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.

RESULTS AND DISCUSSION

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.

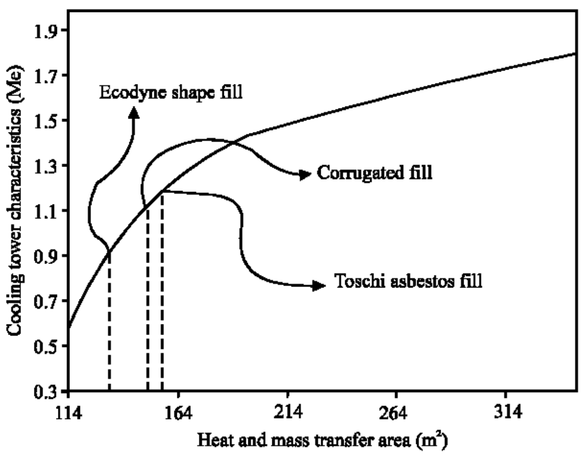

Fig. 3: The influence of heat and mass transfer area on cooling's tower Me number

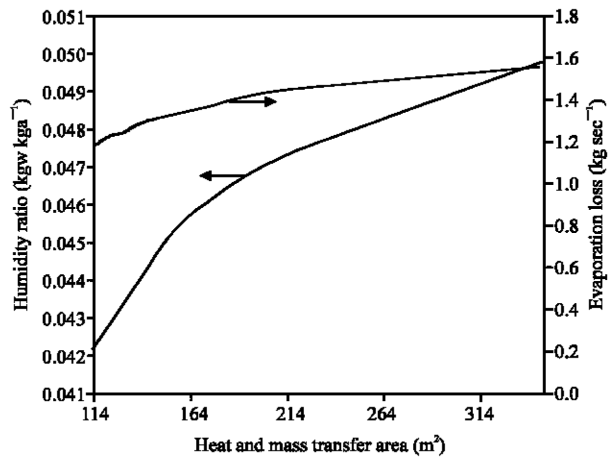

Fig. 4: Influence of heat and mass transfer area on outlet humidity and evaporation loss

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

Design parameter	A_{fr} (m^2)	$A_{h\text{-}m}$ (m^2)
Actual cooling tower	0.98	236.67
Optimum design	0.75	146.46

Table 2: Optimum cooling tower height

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

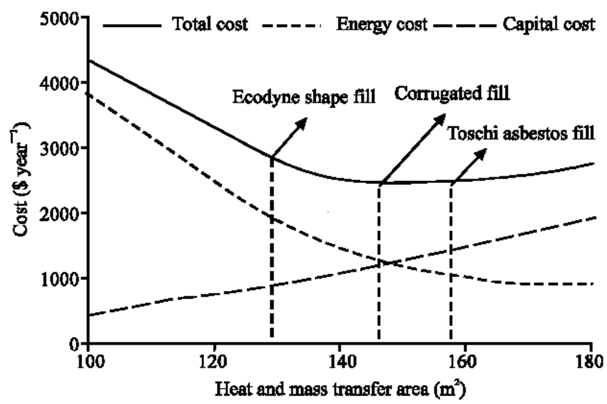

Fig. 5: Variations in total cost relative to heat and mass transfer area

Inlet water temperature is 45°C; outlet water temperature is 33°C; inlet water flow rate is 2.57 kg sec^{-1}; air temperature is 30°C; wet bulb temperature is 25°C; electricity cost is 0.1 \$ kW^{-1} h^{-1}; operating time period is 8600 h year^{-1}; 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 shown 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 shown 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 cooling tower characteristic (Me number) is influenced by the heat and mass transfer area, as shown in Fig. 3. Figure 3 shows that the cooling tower's Me number 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.

Figure 4 shows that the outlet air temperature and therefore the outlet humidity ratio, is increased by increasing the heat and mass transfer area of the tower.

Economical considerations reveal that by increasing the heat and mass transfer area, the capital cost of the cooling tower increases whereas the energy cost decreases. 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. Figure 5 shows the variations in the total cost of a cooling tower relative to various heat and mass transfer areas.

The results show that the corrugated fill is the optimum packing type for the designed cooling tower. Also, the cost of the designed cooling tower is

2.74 k\$ year^{-1}, whereas that of the sample cooling tower is 3.52 k\$ year^{-1}. This reveals a 22% cost reduction compared with the existing cooling tower design.

CONCLUSION

A comprehensive approach to optimum design model of wet cooling tower was developed. The design procedure allowed thermo-economical optimization to be explored systematically. The relation between the tower characteristic and the design parameters were studied. In this approach, the opportunity of heat rejection in the whole cooling tower, spray zone, fill zone and rain zone, was considered. The presented design model allowed different packing types to be chosen for the cooling tower to investigate the transfer opportunity for optimization purpose. The validity of optimization formulation was confirmed with a sample problem. Related coding in Visual Studio.Net 2003 (C++) was developed to get computation results of the optimum design model.

NOMENCLATURE

a	:	Air-water interfacial 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 (\$/kW/h)
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 ($m^2\ sec^{-1}$)
d	:	Droplet diameter (m)
d_{fan}	:	Fan diameter (m)
E_f	:	Economic factor
G	:	Mass velocity (kg/sec/m^2)
g	:	Gravitational acceleration (m sec^{-2})
i	:	Enthalpy (kJ kg^{-1})
i_{fw}	:	Enthalpy of saturated water evaluated as T_w (kJ kg^{-1})
h_d	:	Mass transfer coefficient of (kg/m^2/sec)
H	:	Cooling tower height (m)
K_{el}	:	Eliminator coefficient
L	:	Length (m)
L_i	:	Cooling tower length (m)
Me	:	Cooling tower characteristic
m	:	Flow rate (kg sec^{-1})
N_{deck}	:	No. of decks
P_{atm}	:	Atmospheric pressure (Pa)
P_f	:	Fan power (hp)
Q	:	Heat rate (kW)
R_v	:	Gas constant (J/kg/°C)
Ry	:	Eliminator characteristic (m^{-1})
S	:	Annual total operation time (h)
Sc	:	Schmidt No.
T	:	Temperature (°C)
TC	:	Total Cost, (\$ $year^{-1}$)
w	:	Humidity ratio (kgw kga^{-1})
w_s	:	Saturated humidity ratio (kgw kga^{-1})
W_i	:	Cooling tower width (m)

Subscripts

a	:	Air
Dif	:	Diffuser
fi	:	Fill zone
in	:	Inlet
out	:	Outlet
pl	:	Plenum chamber
rej	:	Rejection
rz	:	Rain zone
sp	:	Spray zone
w	:	Water

Greek letters

υ	:	Velocity (m sec^{-1})
ρ	:	Density (kg m^{-3})
σ	:	Surface tension (N m^{-1})
η	:	Efficiency
μ	:	Viscosity (kg/m/sec)

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