Chapter 1
INTRODUCTION

1.1 Function and Mechanism of a Cooling Tower

The function of a cooling tower is to reduce the temperature of circulating water so that it may be reused in condensers and other heat exchange equipment. Direct contact or counter flow cooling towers are widely used in Power Plants, Chemicals & Petrochemicals, Refrigeration and Air-Conditioning Industries etc. for cooling water or other working medium to near the ambient temperature.

In a counter flow cooling tower, hot water sprayed from the spray nozzles comes in direct contact with the ambient air. In this process, a small portion of cooling water (1-2%) evaporates. This evaporation of water causes an increase in temperature and humidity of air and decrease of temperature of cooling water. Although there is some sensible heat transfer from the water to air, cooling effect in a cooling tower results almost entirely from the evaporation of a portion of the water as the water falls through the cooling tower. The heat to vaporize the portion of water that evaporates is drawn from the remaining mass of water so that the temperature of the mass is reduced. The vapor resulting from the evaporating process is carried away by the air circulating through the cooling tower. Since both the temperature and the moisture content of the air are increased as the air passes through the cooling tower, it is evident that the effectiveness of the cooling tower depends to a large degree on the wet bulb temperature of inlet air. The lower is the wet bulb temperature of the inlet air; the more effective is the cooling tower. Other factors that influence the performance of an induced draft cooling tower are:

(i) Amount of exposed water surface and the length (time) of exposure,

(ii) Velocity of air passing through the cooling tower,

The exposed water surface includes surface of water in the cooling tower basin, all wetted surfaces in the cooling tower and combined surface of the water droplets falling through the cooling tower (Spray and rain zone).

Theoretically, the lowest temperature to which the water can be cooled in a cooling tower is the wet bulb temperature of the inlet air, in which case the water vapor in the outlet air will be saturated. In practice, it is not possible to cool the water to the wet bulb temperature of the air. In most cases, the temperature of the outlet water leaving the cooling tower will be 4-8℃ above the wet bulb temperature of the inlet air. Also, the outlet air leaving the cooling tower will always be somewhat less than saturated.

1.2 Description of Induced Draft Cooling Tower

In an induced draft cooling tower, a fan at the discharge pulls air through the cooling tower as shown in Figure 1.1.The fan induces hot moist air out of the discharge. This produces low entering and high exiting air velocities, reducing the possibility of recirculation in which discharged air flows back into the air intake. This fan/fill arrangement is also known as draw - through.

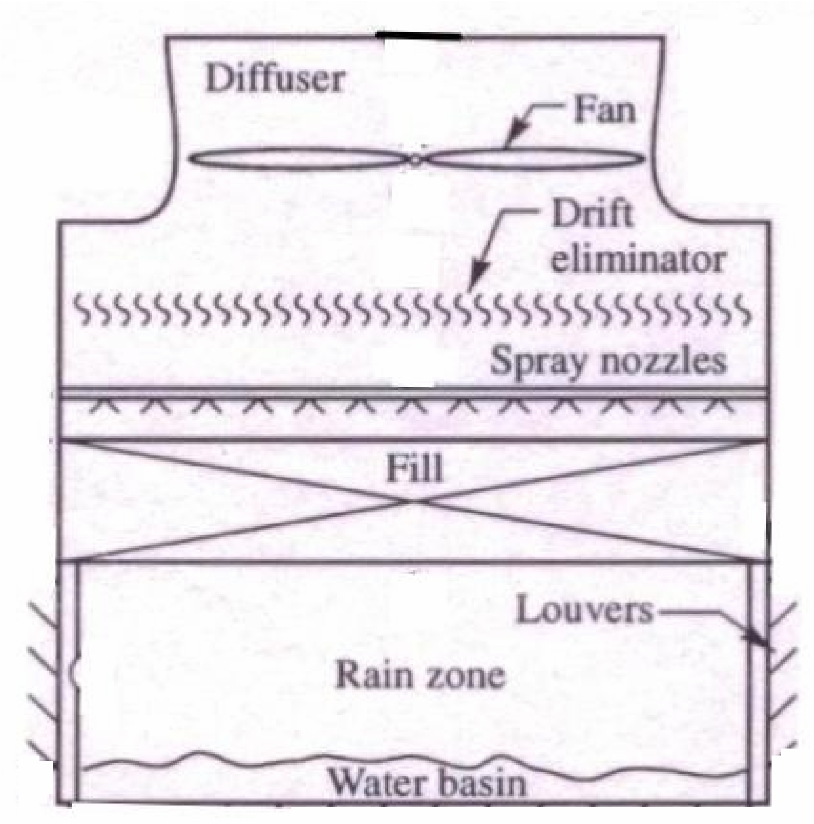

Figure 1.1. Induced draft counter flow cooling tower.

1.3 Motivation

In any power generating or refrigeration cycle, heat has to be discharged. This is also true in many chemical and process plant cycle, internal combustion engines, computers and electronic systems. Because of the restrictions on thermal discharge to natural bodies of water, most new generating capacity (including small to large industrial units) require the mandatory use of closed cycle cooling system. Induced draft cooling towers generally are the most economical choice for closed cycle cooling where an adequate supply of suitable water is available at a reasonable cost to meet the makeup water requirement of these systems. Therefore, an appropriate and well designed induced draft cooling tower system can have a very significant and positive impact on plant performance and profitability.

1.4 Objectives of the Project

The aim of the present study is to evaluate the thermal flow performance or design of the induced draft cooling towers by using modern analytical and empirical tools. To achieve this aim, following objectives are laid down for this study:

(i) Development of a simple and efficient mathematical model for estimating heat and mass transfer between hot water and air stream, to enable an accurate prediction of cooling tower performance and fan power simultaneously with available empirical relations for pressure drop.

(ii) Parametric study of governing variables.

Chapter 2
LITERATURE REVIEW

Large number of literature is available for induced draft cooling towers. A brief summary of the same is given below.

2.1 Summary of Literature Review

The basic theory of cooling tower operation was first proposed by Walker et al. in 1923. Merkel developed the theory for the thermal evaluation of cooling towers in 1925. This work was largely neglected until 1941 when the paper was translated into English. Since then, the model has been widely applied by the researchers.

Merkel combined the equations for heat and water vapor transfer and used enthalpy as the driving force to allow for both sensible and latent heat transfer. Heat is removed from the water by transfer of the sensible heat due to the difference in temperature levels and by the latent heat equivalent of the mass transfer resulting from the evaporation of a portion of the circulating water. Merkel combined these into a single process based on the enthalpy difference as the driving force. Merkel theory relies on several critical assumptions to reduce the solution to a simple hand calculation. However, the Merkel method does not accurately represent the physics of heat and mass transfer process in the cooling tower fill.

Jaber and Webb [1] developed the equations necessary to apply the e-NTU method directly to counter flow or cross flow cooling towers. The e-NTU method is based on the same simplifying assumptions as the Merkel method. Kloppers et al. [2] investigated the critical differences in the heat and mass transfer analyses and solution techniques of the Merkel and Poppe methods using enthalpy diagrams and psychrometric charts. They made a detailed derivation of the heat and mass transfer equations of evaporative cooling in wet-cooling towers and derived the governing equations of Poppe method from first principles. Poppe method is well suited for the analysis of hybrid cooling towers as the state of the outlet air is accurately predicted. The governing equations of the Merkel method were subsequently derived after some simplifying assumptions. The equations of e-NTU method applied to wet-cooling towers were also presented. The governing equations of Poppe method were extended to give a more detailed representation of the Merkel number. Again, Kloppers et al. [3] investigated the performance evaluation of the Merkel, e-NTU and Poppe methods for a certain fill material at different operating and ambient conditions. Kloppers et al. [4] investigated the effect of the Lewis factor (which relates the relative rates of heat and mass transfer in wet cooling towers) on the performance prediction of natural draft and mechanical draft wet-cooling towers and also investigated the relation of the Lewis factor to the Lewis number. Kloppers et al. [5] proposed a new form of empirical equation that correlates fill loss coefficient data more effectively when compared to other forms of empirical equation commonly found in literature. Naphon [6] investigated both experimental and theoretical results of the heat transfer characteristics of the cooling tower and found that there is a reasonable agreement from the comparison between the measured data and predicted results in his model. Kachhwaha et al. [7] carried out heat and mass transfer analysis of a counter flow wet cooling tower. Sutherland [8] compared accurate analysis of mechanical draft counter flow cooling tower, including water loss by evaporation, with the approximate common method based on enthalpy driving force (Merkel method) for wide range of inlet water and air conditions. A few theoretical and experimental works have been reported on the heat transfer characteristics.

Fisenko et al. [9] developed a new mathematical model of a mechanical draft cooling tower performance which represents a boundary-value problem for a system of ordinary differential equations, describing a change in the droplets velocity, its radii and temperature, and also a change in the temperature and density of the water vapor in a mist air in a cooling tower. The model describes available experimental data with an accuracy of about 3%. The model takes into account the radii distribution function of water droplets and simulation based on the model allows one to calculate contributions of various physical parameters on the processes of heat and mass transfer between water droplets and damp air, to take into account the cooling tower design parameters and the influence of atmospheric conditions on the thermal efficiency of the tower. The explanation of the influence of atmospheric pressure on the cooling tower performance has been obtained for the first time. It was shown that the average cube of the droplet radius practically determines thermal efficiency. The relative accuracy of well-defined mono disperse approximation is about several percent of heat efficiency of the cooling tower. A mathematical model of a control system of the mechanical draft cooling tower is suggested and numerically investigated. This control system permits one to optimize the performance of the mechanical draft cooling tower under changing atmospheric conditions. Soylemez [10] carried out a thermo-hydraulic performance optimization analysis yielding simple algebraic formula for estimating the optimum performance point of counter current mechanical draft wet cooling towers using e-NTU method along with derivation of psychrometric properties of moist air based on a numerical approximation method for thermal performance analysis of counter flow wet cooling towers.

Guang Yu Jin et al. [11] developed a new, simple and accurate mechanical cooling tower model for the purpose of energy conservation and management based on Merkel's theory and e-NTU method by energy balance and heat, mass transfer analysis. Compared with the existing models, this model has simple characteristic parameters to be determined and without requiring iterative computation when the operating point changes and the model is validated by real operating data from the cooling towers of a HVAC system of a commercial hotel. The test results show that the performance of the cooling tower varies from time to time due to different operating conditions and this model reflects these changes by tuning its parameters and thus it can accurately predict the performance of the real-time operating cooling tower.

Bilal A. Qureshi et al. [12] modeled three zones of the cooling tower; namely, spray zone, packing and rain zones and the developed models of these zones were validated against experimental data. For a case study, error in calculation of tower volume is 6.5% when spray and rain zones are neglected. This error is reduced to 3.15% and 2.65% as spray and rain zones are incorporated in the model, respectively. Furthermore, fouling in cooling tower fills as well as its modeling strategy is explained and incorporated in the cooling tower model to study performance evaluation problems. The fouling model is presented in terms of normalized fill performance index as a function of weight gain due to fouling.

Again, Bilal A. Qureshi et al. [13] presented thermodynamic analysis of counter flow wet cooling towers and evaporative heat exchangers using both the first and second laws of thermodynamics. A parametric study is carried out to determine the variation of second-law efficiency as well as exergy destruction as a function of various input parameters such as inlet wet bulb temperature. Irreversible losses are determined by applying an exergy balance on each of the systems investigated using engineering equation solver (EES) program. The concept of total exergy as the sum of thermo-mechanical and chemical parts is employed in calculating the flow exergies for air and water vapor mixtures. For the different input variables investigated, efficiencies were, almost always, seen to increase or decrease monotonically. We notice that an increase in the inlet wet bulb temperature invariably increases the second-law efficiency of all heat exchangers. Furthermore, it was also investigated that the variation in the dead state does not significantly affect the overall efficiency of the system.Muangnoi et al. [14] carried out an exergy analysis to indicate exergy and exergy destruction of water and air flowing through the cooling tower. The model was validated against experimental data and it was noted from the results that the amount of exergy supplied by water is larger than that absorbed by air, because the system produces entropy. To depict the utilizable exergy between water and air, exergy of each working fluid along the tower were presented. The results showed that water exergy decreases continuously from top to bottom. On the other hand, air exergy is expressed in terms of convective and evaporative heat transfer. Exergy of air via convective heat transfer initially loses at inlet and slightly recovers along the flow before leaving the cooling tower. However, exergy of air via evaporative heat transfer is generally high and able to consume exergy supplied by water. Exergy destruction is defined as the difference between water exergy change and air exergy change. It reveals that the cooling processes due to thermodynamics irreversibility perform poorly at bottom and gradually improve along the height of the tower. The results show that the lowest exergy destruction is located at the top of the tower.

2.2 Conclusion of Literature Review

From the above literature survey, it can be concluded that Merkel, e-NTU and Poppe methods are most common and widely used by the researchers. The formulations of Merkel method are described in Chapter 3. Limited information is available in the literature regarding simultaneous solution of energy equation and draft equation. Information regarding various pressure losses is available through empirical relations.

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Chapter 3
FORMULATION OF MATHEMATICAL MODEL

This chapter describes various formulations used for design and analysis of induced draft cooling towers. The details are given below.

3.1 Merkel Method

Merkel method relies on the critical simplifying assumptions as given below:

(i) The value of Lewis factor (Le_f) relating heat and mass transfer for air-water vapor system is equal to 1.

(ii) The air leaving the cooling tower is saturated with water vapor and it is characterized only by its enthalpy.

(iii) The reduction of water flow rate by evaporation is neglected.

According to Merkel method,

$$\frac{di_{ma}}{dz} = \frac{h_d a_{fi} A_{fr}}{m_a}(i_{masw} - i_{ma}) \tag{3.1}$$

$$\frac{dt_w}{dz} = \frac{m_a}{m_w}\frac{1}{c_{pw}}\frac{di_{ma}}{dz} \tag{3.2}$$

Equations (3.1) and (3.2) describe change in enthalpy of air-vapor mixture and change in water temperature, respectively, as air travel distance changes. Equations (3.1) and (3.2) can be combined to yield upon integration, the Merkel equation,

$$Me_M = \frac{h_d a_{fi} A_{fr} L_{fi}}{m_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{t_{wo}}^{t_{wi}} \frac{c_{pw} dt_w}{(i_{masw} - i_{ma})} \tag{3.3}$$

where Me_M is the transfer coefficient or Merkel number according to the Merkel method, a_{fi} is the surface area of the fill per unit volume of the fill, h_d is the mass transfer coefficient and L_{fi} is the height of the fill or air travel distance. The above equation is commonly known as Merkel equation. The terms i_{masw} and i_{ma} denotes enthalpy of saturated air-vapor and air-vapor respectively.

In the literature, the notation frequently used for the Merkel number is KaV/L,

where $K = h_d$ $a = a_{fi}$ $V = (A_{fr}.L_{fi})$ $L = m_w$

Different numerical integration methods may be considered to approximate the Merkel integral equation (3.3). These methods vary both in accuracy and computational effort. Chebyshev's method uses values of the integrand at predetermined values within the integration interval selected so that the sum of these values multiplied by the interval times a constant gives the approximate integral. In its four-point form, the approximate formula is

$$Me_M = \frac{h_d a_{fi} A_{fr} L_{fi}}{m_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{t_{wo}}^{t_{wi}} \frac{c_{pw} dt_w}{(i_{masw} - i_{ma})} = \left(\frac{t_{wi} - t_{wo}}{4}\right)\left[\frac{c_{pw1}}{\Delta i_{(1)}} + \frac{c_{pw2}}{\Delta i_{(2)}} + \frac{c_{pw3}}{\Delta i_{(3)}} + \frac{c_{pw4}}{\Delta i_{(4)}}\right]$$

$$= \frac{c_{pwm}(t_{wi} - t_{wo})}{4}\left[\frac{1}{\Delta i_{(1)}} + \frac{1}{\Delta i_{(2)}} + \frac{1}{\Delta i_{(3)}} + \frac{1}{\Delta i_{(4)}}\right] \tag{3.4}$$

Enthalpy differentials (Δi), are dependent on following intermediate temperatures:

$$t_{w(1)} = t_{wo} + 0.1\,(t_{wi} - t_{wo}) \tag{3.4a}$$

$$t_{w(2)} = t_{wo} + 0.4\,(t_{wi} - t_{wo}) \tag{3.4b}$$

$$t_{w(3)} = t_{wo} + 0.6\,(t_{wi} - t_{wo}) \tag{3.4c}$$

$$t_{w(4)} = t_{wo} + 0.9\,(t_{wi} - t_{wo}) \tag{3.4d}$$

The subscripts 1,2,3,4 used in equation (3.4) refer to the intervals in the Chebyshev's integral [16].

3.2 System Description and Model Formulation

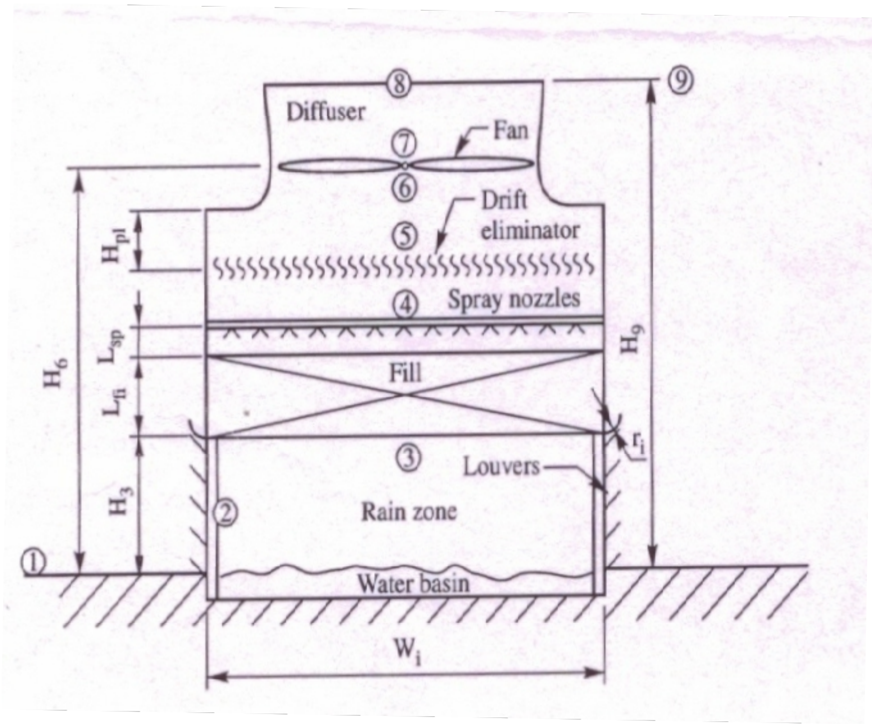


Figure 3.1. Induced draft counter flow cooling tower with geometry.

Figure 3.1 represent the schematic of an induced draft counter flow cooling tower with geometrical information. As shown in Figure 3.1, point 1 refers to the ground level, point 2 refers to the conditions to the inlet of rain zone of the cooling tower, point 3 refers to the conditions to the outlet of rain zone or inlet of the fill zone of the cooling tower, point 4 refers to the conditions to the outlet of spray zone, point 5 refers to the conditions to the outlet of drift eliminator, point 6 refers to the conditions upstream of fan, point 7 refers to the conditions downstream of fan and point 8 refers to the conditions in the atmosphere. Various geometries required in the model are shown in Figure 3.1.The fill characteristics used in the present formulation is given below.

The transfer coefficient of the fill (Me_{fi}) of cooling tower is given by

$$Me_{fi} = a_d.\ L_{fi}.G_w^{bd}\ G_a^{\ cd}$$

The values of coefficients are

a_d = 0.2692; b_d = – 0.094 and c_d = 0.6023 for the selected fill.

The loss coefficient of the fill due to frictional and drag effects (K_{fdm}) at mean conditions of cooling tower is given by

$$K_{fdm} = a_{dl}.\ L_{fi}.G_w^{bdl}\ G_a^{\ cdl}$$

The values of coefficients are

a_{dl} = 1.9277; b_{dl} = 1.2752; c_{dl} = –1.0356 for the selected fill.

The mathematical model for the induced draft cooling tower consists of three main equations namely; energy equation, draft equation and pressure equation. These equations are described below.

3.2.1 Energy equation

The amount of heat transferred (J/s) to the air stream from the circulating water is expressed by the energy equation as

$$q = m_w\ .\ c_{pwm}\ .\ (t_{wi} - t_{wo}) = m_a\ (i_{mas5} - i_{ma1}) \tag{3.5}$$

where i_{masw5} is the enthalpy of saturated air-vapor at 5 and i_{ma1} is the enthalpy of air-vapor at the inlet of the cooling tower.

The amount of water lost due to evaporation [$m_{w(evap)}$, kg/s] is given by

$$m_{w(evap)} = (m_{av5} - m_{av1}) \tag{3.6}$$

3.2.2 Draft equation

For an induced draft counter flow cooling tower shown in Figure 2, ignoring pressure differences due to gravity field, the draft equation obtained by matching fan performance curve and the flow characteristics is expressed as

$$(K_{ilfi} + K_{rzfi} + K_{fsfi} + K_{fi} + K_{spfi} + K_{wdfi} + K_{defi} + K_{ctfi} + K_{upfi}) \times \quad (3.7)$$

$$(m_{av15}/A_{fr})^2/(2 \ \rho_{av15}) - (K_{Fs}(m_{av5}/A_c)^2/ \ (2 \ \rho_{av6}) = 0$$

where K denotes the loss coefficient and

m_{av15} = average air-vapor mass flow rate between 1 and 5

A_{fr} = frontal area of the fill

ρ_{av15} = harmonic mean density of air-vapor between 1 and 5

= $2 / (1/\rho_{av1} + 1/\rho_{av5})$

m_{av5} = air-vapor mass flow rate at 5

A_c = area of the fan casing

ρ_{av6} = density of air-vapor at 6.

The various loss coefficients shown in equation (6) are calculated by using following empirical equations.

The specified loss coefficient due to inlet louvers (K_{ilfi}) referred to the mean conditions through the fill is

$$K_{ilfi} = K_{il} \ (\rho_{av15}/\rho_{av1})\{(W_i.B_i) / (2H_3.W_i)\} \ (m_{av1}/m_{av15})^2 \quad (3.7a)$$

where K_{il} denotes loss coefficient for inlet louvers and

ρ_{av1} = density of air-vapor at 1

W_i = tower inlet width

B_i = tower breadth or length

H_3 = tower inlet height

m_{av1}= air-vapor mass flow rate upstream of fill

The rain zone loss coefficient (K_{rzfi}) referred to the mean conditions through the fill is given by

$$K_{rzfi} = K_{rz}.\ (\rho_{av15}/\rho_{av1}).\ (m_{av1}/m_{av15})^2 \qquad (3.7b)$$

where K_{rz} = loss coefficient for the rain zone

The specified loss coefficient of the support structure of the fill (K_{fsfi}) referred to the mean conditions through the fill is given by

$$K_{fsfi} = K_{fs}.\ (\rho_{av15}/\rho_{av1}).\ (m_{av1}/m_{av15})^2 \qquad (3.7c)$$

where K_{fs} = loss coefficient for fill support

For the specified fill, loss coefficient (K_{fdm}) is given by

$$K_{fdm} = a_{dl}.\ L_{fi}.G_w^{\ bdl}\ G_a^{\ cdl} \qquad (3.7d)$$

The values of the coefficients for the fill selected are given in section 3.2.

The actual fill loss coefficient (K_{fi}) applicable to cooling tower is given by

$$K_{fi} = K_{fdm} + [(G_{av5}^2/\rho_{av5}) - (G_{av1}^2/\rho_{av1})] / (G_{av5}^2/\rho_{av15}) \qquad (3.7e)$$

where G_{av1} = mass velocity of air-vapor at 1 [$G = m / A_{fr}$]

G_{av5} = mass velocity of air-vapor at 5

The loss coefficient through the spray zone (K_{spfi}) above the fill referred to the mean conditions through the fill is given by

$$K_{spfi} = L_{sp}[0.4(G_w/G_a) + 1].(\rho_{av15}/\rho_{av5}).\ (m_{av5}/m_{av15})^2 \qquad (3.7f)$$

where L_{sp} = height of the spray zone

G_w = mass velocity of water based on frontal area of the fill

G_a = mass velocity of dry air based on frontal area of the fill

The specified loss coefficient due to water distribution system(K_{wdfi}) referred to the mean conditions through the fill is given by

$$K_{wdfi} = K_{wd} (\rho_{av15}/\rho_{av5}). (m_{av5}/m_{av15})^2 \tag{3.7g}$$

where K_{wd} = loss coefficient for water distribution system

The loss coefficient for drift eliminator (K_{defi}) based on the fill conditions is given by

$$K_{defi} = a_{de}. R_y^{bde}. (\rho_{av15}/\rho_{av5}). (m_{av5}/m_{av15})^2 \tag{3.7h}$$

where R_y = characteristic flow parameter = $m / (\mu. A_{fr})$

In the present case, the commercially available "type c" drift eliminator has been selected for which

$a_{de} = 27.4829$, $b_{de} = -0.14247$

For "type a" and "type b" drift eliminators,graphical information is available[16].

The inlet loss coefficient ($K_{ct(norz)}$)for an induced draft, isotropically packed, rectangular cooling tower is given by

$$\begin{aligned} K_{ct(norz)} = {} & 0.2339 + (3.919 \times 10^{-3} K_{fie}^2 - 6.84 \times 10^{-2} K_{fie} + 2.5267) \\ & \times \exp[W_i\{0.5143 - 0.1803 \exp(0.0163 K_{fi})\}/H_3] \\ & -\sinh^{-1}[2.77 \exp(0.958 W_i/H_3) \\ & \exp\{K_{fie}(2.457-1.015 W_i/H_3) \times 10^{-2}\}(r_i/W_i - 0.013028)] \end{aligned} \tag{3.7i}$$

Where the effective loss coefficient in the vicinity of the fill (K_{fie}) is given by

$$K_{fie} = K_{fsfi} + K_{fi} + K_{spfi} + K_{wdfi} + K_{defi} \tag{3.7j}$$

According to [17], it becomes acceptable to ignore the inlet loss correction factor in cases $W_i/H_3 \leq 3$. In this case, $W_i/H_3 = 3$, which means that

$K_{ct} = K_{ct(norz)}$

Referred to mean conditions through the fill, the inlet loss coefficient (K_{ctfi}) becomes

$$K_{ctfi} = K_{ct} . (\rho_{av15}/\rho_{av5}). (m_{av5}/m_{av15})^2 \qquad (3.7k)$$

The specified fan upstream loss coefficient (K_{upfi}) referred to mean conditions through the fill is given by

$$K_{upfi} = K_{up}. (\rho_{av15}/\rho_{av5}). (m_{av5}/m_{av15})^2.(A_{fr}/A_c)^2 \qquad (3.7l)$$

where K_{up} = fan upstream losses

3.2.3 Pressure equation

The pressure of air upstream of fan (p_{a5}) is expressed as

$$p_{a5} = p_{a1}[1-(0.009754(H_3 + L_{fi}/2)/t_{a1}]^{3.5(1+w1)(1-w1/(w1+0.622))} - (K_{ilfi}+K_{rzfi} + K_{fsfi} + K_{fi} + K_{spfi} + K_{wdfi} + K_{defi} + K_{ctfi}) \times (m_{av15}/A_{fr})^2 / (2\, \rho_{av15}) \qquad (3.8)$$

Here, it is assumed that the air-vapor leaving the cooling tower is saturated and condition of the air at the inlet of the fan is equal to the condition at the outlet of the fill, so the properties of air-vapor at section 5 and 6 are taken as same.

3.2.4 Fan Power Equations

The actual air volume flow rate (V_F, m^3/s) through the fan is given by

$$V_F = m_{av5} / \rho_{av5} \qquad (3.9)$$

As actual air density and rotational speed of the fan are not the same as the reference conditions for which fan performance characteristics were specified, the relevant fan laws [17] are employed.

Accordingly, air volume flow rate ($V_{F/dif}$, m^3/s) is given by

$$V_{F/dif} = V_F.(N_{Fr}/ N_F) . (d_{Fr}/d_F)^3 \qquad (3.10)$$

where N_{Fr} = reference fan rotational speed (r/min)

N_F = fan rotational speed (r/min)

d_{Fr} = test fan diameter (m)

d_F = fan diameter (m)

The reference fan static pressure difference ($\Delta p_{F/dif}$, N/m^2) is given by

$$\Delta p_{F/dif} = 320.85 - 6.9604\ V_{F/dif} + 0.31373\ V_{F/dif}^2 - 0.021393\ V_{F/dif}^3 \qquad (3.11)$$

The actual fan static pressure difference (Δp_{Fs}, N/m^2) is given by

$$\Delta p_{Fs} = \Delta p_{F/dif}.(N_F/\ N_{Fr})^2\ .\ (d_F/d_{Fr})^2.(\rho_{av6}/\rho_r) \qquad (3.12)$$

The fan shaft power at reference conditions ($P_{F/dif}$, W) is given by

$$P_{F/dif} = 4245.1 - 64.134\ V_{F/dif} + 17.586\ V_{F/dif}^2 - 0.71079\ V_{F/dif}^3 \qquad (3.13)$$

The actual fan shaft power (P_F,W) is given by

$$P_F = P_{F/dif}.(N_F/\ N_{Fr})^3\ .\ (d_F/d_{Fr})^5.(\rho_{av6}/\rho_r) \qquad (3.14)$$

The static pressure rise coefficient of the fan ($K_{F/difs}$) is represented as

$$K_{F/difs} = 2.\Delta p_{Fs}.\rho_{av6}\ /\ [m_{av5}/A_c]^2 \qquad (3.15)$$

3.2.5 Formulations for three zones of the cooling tower

(i) Transfer coefficient in rain zone (Me_{rz}) of the cooling tower from [12] is given by

$$\begin{aligned} Me_{rz} = {} & 3.6(p_a/R_v.t_a.\rho_w).(D/v_{a,in}.d_d).(H_{rz}/d_d).Sc^{0.33} \qquad (3.16) \\ & \times \ln[(w_s+0.622)/(w+0.622)]\ /(w_s-w) \\ & \times \{5.01334.b_1.\rho_a - 192121.7.\ b_2.\mu_a - 2.57724 + 23.61842 \\ & \times [0.2539\ (b_3.v_{a,in})^{1.67} + 0.18] \times [0.83666\ (b_4.\ H_{rz})^{-0.5299} + 0.42] \\ & \times [43.0696\ (b_4.\ d_d)^{0.7947} + 0.52] \end{aligned}$$

where humidity ratio (w_{s1}) of saturated air at t_{wo} is calculated from equation A.3.5 given in appendix A.

Diffusion coefficient (D_1, m^2/s) at inlet conditions is given by

$$D_1 = 0.04357\, T^{1.5} (1/M_a + 1/M_v)^{0.5} / [p.(V_a^{0.333} + V_v^{0.333})^2] \quad (3.16a)$$

Schmidt number at 1 is given by

$$Sc_1 = \mu_{av1} / (\rho_{av1} / D_1) \quad (3.16b)$$

Air-vapor velocity at 3 before the fill is given by

$$v_{av3} = m_{av1} / (\rho_{av1} / A_{fr}) \quad (3.16c)$$

The 'b' coefficients appearing in equation of rain zone are given by

$b_1 = 998/\rho_{wo}$ $\qquad$ $b_2 = 3.061 \times 10^{-6}(\rho_{wo}^4\, g^9/\sigma_{wo})^{0.25}$

$b_3 = 73.298\, (g^5\, \sigma_{wo}^3/\rho_{wo}^3)^{0.25}$ $\qquad$ $b_4 = 6.122(g\, \sigma_{wo}/\rho_{wo})^{0.25}$

The above formulation [equation (16)] for transfer coefficient in rain zone (Me_{rz}) of the cooling tower is applicable with following restrictions:

$\rho_a = 0.927 – 1.289$ kg/m^3 $\qquad$ $v_{a,in} = 1 – 5$ m/s

$d_d = 0.002 – 0.008$ m $\qquad$ $\mu_a = 1.717 – 1.92 \times 10^{-5}$ kg/ms

where d_d = droplet diameter in rain zone (m)

μ_a = dynamic viscosity of air (kg/ms)

(ii) Transfer coefficient in fill zone (Me_{fi}) of cooling tower for any fill is given by

$$Me_{fi} = a_d.\, L_{fi}.G_w^{bd}\, G_a^{cd} \quad (3.17)$$

The coefficients a_d, b_d and c_d are taken from the fill data [16].

(iii) Transfer coefficient in spray zone (Me_{sp}) of the cooling tower is given by

$$Me_{sp} = 0.2\, L_{sp}\, .(G_a/G_w)^{0.5} \quad (3.18)$$

(iv) Total transfer characteristic of cooling tower (Me_T) is given by

$$Me_T = Me_{rz} + Me_{fi} + Me_{sp} \tag{3.19}$$

3.2.6 Formulation for Exergy Analysis

Energy exists in different forms. It is a measure of quantity but an energy source cannot be evaluated on its quantity alone. A measure of the quality of energy is defined as exergy, which is the work potential of energy in a given environment. Exergy analysis is defined as a method of performing system analysis according to the conservation of mass, conservation of momentum and second law of thermodynamics. It consists of using the first and second law together, for the purpose of analyzing performance in the reversible limit, and estimating the departure from this limit [13]. We note that it is exergy, not energy that represents the true potential of a system to perform an optimal work with respect to a dead state or surrounding. The greater the difference between the energy source and its surroundings, the capacity to extract work from the system increases. It is important to understand that before analyses can be applied with confidence to engineering systems, the significance of the sensitivities of exergy analysis results to reasonable variations in or selection of dead state properties should be evaluated.

In a counter flow cooling tower, water and air are the only two kinds of working fluids revealed in operation. So it is important to write the exergy equations for both water and used in the exergy analysis.

Exergy of Water

According to [14], the exergy (W) of water is given by

$$X_w = m_w.\ [\ (h_{fw} - h_{fwr}) - \ t_r.(s_{fw} - s_{fwr}) - R_v.\ t_r\ .\ln\ (\theta_r)] \tag{3.20}$$

where $\theta_r = p_a.w/(0.622 + w).p_{vs}$

and h and s represent enthalpy and entropy of water respectively.

Exergy of Air-Vapor

The exergy of air-vapor is sum of exergy of dry air and exergy of vapor.

The specific exergy of dry air (J/kg) is given by [14]

$$\Psi_a = [x_a.(c_{pa}/M_a).\{\ t - tr - tr.\ \ln(t/t_r)\ \} + (R/M_a).t_r.(p/p_r) + (R/M_a).t_r.\ \ x_a.\ \ \ln\ \ (x_a/x_{ar})] \tag{3.21}$$

The specific exergy of vapor is given by [14]

$$\Psi_v = [x_v.(c_{pv}/M_v).\{\ t - tr - tr.\ \ln(t/t_r)\ \} + (R/M_v).t_r.(p/p_r) + (R/M_v).t_r.\ \ xv.\ \ \ln\ \ (x_v/x_{vr})] \tag{3.22}$$

With the above equations, the exergy of air-vapor mixture becomes

$$X_{av} = m_a\ [\Psi_a + \Psi_v] \tag{3.23}$$

Exergy Destruction (X_d)

The exergy destruction is given by

$$X_d = (X_{wi} + X_{avi} + X_{wimakeup}) - (X_{wo} + X_{avo}) \tag{3.24}$$

Second Law Efficiency (η_{II})

The second law efficiency is given by

$$\eta_{II} = 1 - \ [(X_d\ \ / (X_{wi} + X_{avi} + X_{wimakeup})] \tag{3.25}$$

Thermal Efficiency (η_{th})

The thermal efficiency of a cooling tower or the efficiency of evaporative cooling is given by [15]

$$\eta_{th} = (t_{wi} - t_{wo}) / (t_{wi} - t_{wb1}) \tag{3.26}$$

Chapter 4
RESULTS & DISCUSSIONS

The present chapter describes the solution procedure of the mathematical formulation given in Chapter 3 followed by results and discussions.

4.1 Solution Procedure

The formulation described in Chapter 3 consist of energy equation, draft equation and pressure equation represented by equation (3.5),(3.7) and (3.8) respectively. These equations are highly non-linear in nature and therefore, an iterative procedure has been developed to solve these equations. The unknown in these three equations are; air-vapor outlet temperature (t_{a5}), average mass flow rate of air-vapor (m_{av15}) through the cooling tower and the pressure (p_{a5}) at Section 5. Initially, guess values of m_{av15} and p_{a5} are supplied to the energy equation. The initial guess value of m_{av15} is chosen nearly equal to mass flow rate of inlet water (m_w) and the value of p_{a5} is chosen slightly less than the atmospheric pressure (p_{a1}) at ground level at Section 1. By using these guess values; the energy equation is solved for air-vapor outlet temperature (t_{a5}). Now, the calculated value t_{a5} and initial guess value of p_{a5} is supplied to the draft equation which is further solved for new value of m_{av15}. Now, the calculated values of m_{av15} and t_{a5} are supplied to the pressure equation to determine the calculated value of p_{a5}. At this juncture, the calculated values of m_{av15} and p_{a5} are compared with the initial guess values and modified accordingly. Using these modified values, the procedure stated above is repeated. The computer program advances when the difference between the calculated values and initial guess values satisfy the prescribed tolerance limit. The above procedure is explained in information flow diagram shown in Figure 4.1.

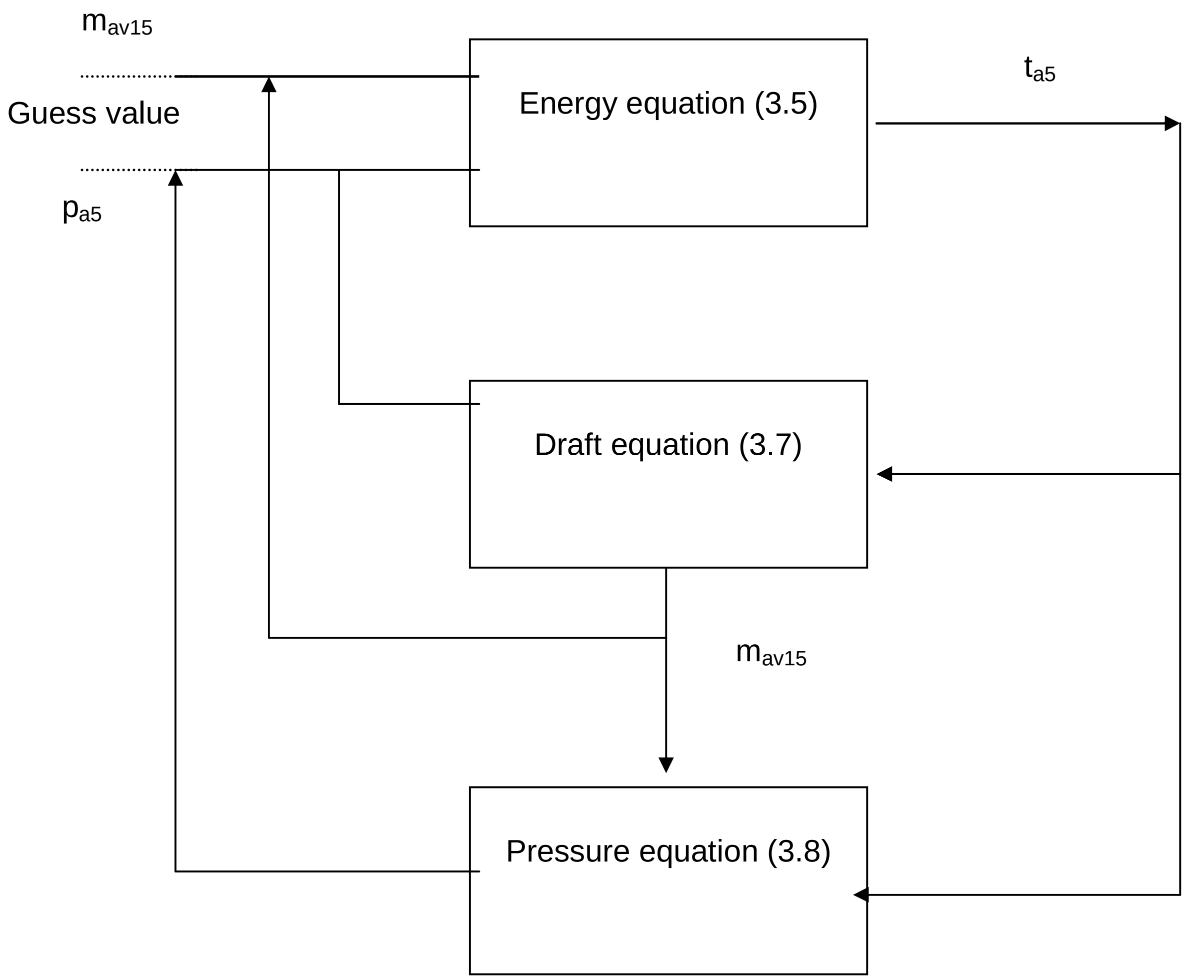
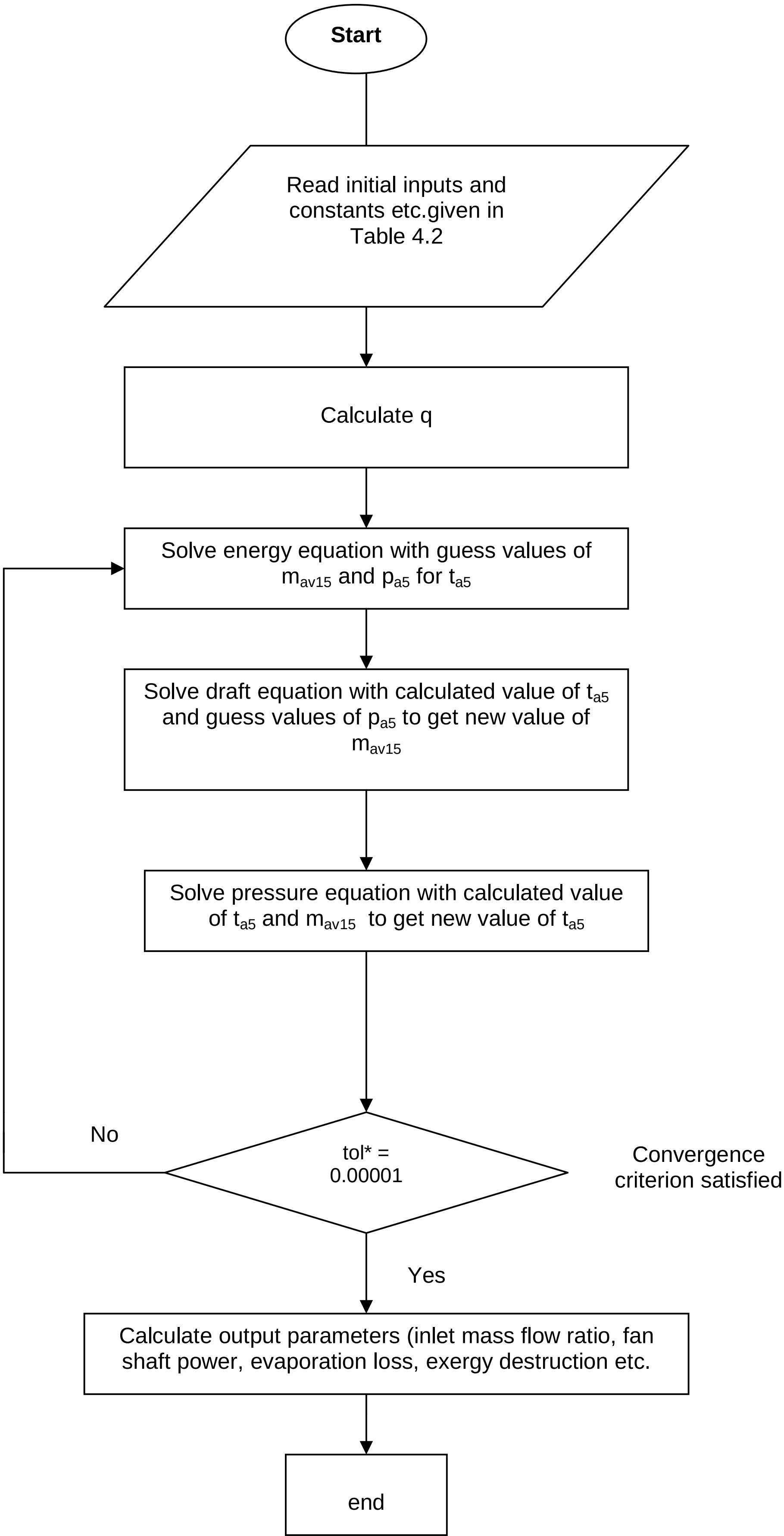


Figure 4.1. Information flow diagram

Based upon above calculated values of m_{av15}, t_{a5} and p_{a5} and the supplied input values, total transfer characteristics of the cooling tower are calculated by Merkel method using Chebyshev's four-point formula [equation(3.4)] given in Chapter 3. Further, the transfer characteristics of the three zones of the cooling tower namely; spray zone, fill zone and rain zones are calculated individually using empirical relations represented by equations (3.16) to (3.19) of Chapter 3.The sum of the transfer characteristics of the three zones were verified with the total transfer characteristics obtained from Merkel method using Chebyshev's four-point formula. The magnitude of various loss coefficients of the different zones of the cooling tower (Chapter 3) has been calculated. The fan shaft power calculations were performed using equations (3.8) to (3.13). The evaporation loss is calculated using equation (3.15). Besides this, exergy destruction (X_d), second law efficiency (η_{II}) and thermal efficiency (η_{th}) of the cooling tower is calculated using equations (3.24) to (3.26) respectively. A detailed flowchart showing the various steps is shown in Figure 4.2.


Start
Read initial inputs and constants etc.given in Table 4.2
Calculate q
Solve energy equation with guess values of m_{av15} and p_{a5} for t_{a5}
Solve draft equation with calculated value of t_{a5} and guess values of p_{a5} to get new value of m_{av15}
Solve pressure equation with calculated value of t_{a5} and m_{av15} to get new value of t_{a5}
No
tol* = 0.00001
Convergence criterion satisfied
Yes
Calculate output parameters (inlet mass flow ratio, fan shaft power, evaporation loss, exergy destruction etc.
end

4.2 Initial Conditions

The initial conditions required for the computer program are shown in Table 4.1.These inputs include air/water conditions at inlet of the cooling tower, water outlet temperature, geometrical parameters of the fill and cooling tower, fan parameters, droplet diameter in the rain zone, loss coefficients and guess values of m_{av15} and p_{a5}.

Table 4.1. Initial conditions supplied to the computer program.

Input Parameters	
Air/water conditions	
Atmospheric pressure at ground level 1(Pa),p_{a1}	101325.00
Water inlet temperature (K),t_{wi}	314.6500
Water outlet temperature (K),t_{wo}	303.4677
Inlet water mass flow rate(kg/s),m_w	412.0000
Inlet air dry bulb temperature(K),t_{a1}	306.6500
Inlet air wet bulb temperature(K),t_{wb1}	298.1500
Geometric parameters	
Tower height,H_9 (m)	12.5
Fan height,H_6 (m)	9.5
Tower inlet height,H_3 (m)	4.0
Tower inlet width,W_i (m)	12.0
Tower breadth or length, B_i (m)	12.0
Fill height (m),L_{fi}	1.878
Height of the spray zone(m),L_{sp}	0.5
Inlet rounding (m),r_i	0.025 W_i
Plenum chamber height (m),H_{pl}	2.4

Input Parameters (continued…………..)	
<u>Fan parameters</u>	
Fan diameter(m),d_F	8.0
Fan rotational speed (r/min),N_F	120
Test fan diameter(m),d_{Fr}	1.536
Reference rotational speed (r/min),N_{Fr}	750
Reference air density (kg/m^3), ρ_r	1.2
<u>Other specifications</u>	
Mean droplet diameter in rain zone, d_d (m)	0.0035
Loss coefficient for inlet louvers, K_{il}	3.5
Loss coefficient for fill support ,K_{fs}	0.5
Loss coefficient for water distribution system, K_{wd}	0.5
Fan upstream losses, K_{up}	0.52
<u>Guess Values</u>	
Average mass flow rate of air-vapor through the cooling tower, m_{av15} (kg/s)	$\cong m_w$
Pressure at 5, p_{a5}, (N/m^2)	$\leq p_{a1}$

The characteristics of the fill used are given in Section 3.2 of Chapter 3.

4.3 Results and discussions

The results obtained by using the values of input parameters of Table 4.1 are shown in Table 4.2.

Table 4.2. Output results of the induced draft cooling tower.

S.No.	Calculated values (Output)	
1.	Average mass flow rate of air-vapor(kg/s),m_{av15}	441.7592 (442.1426)
2.	Pressure of air at 5 upstream of fan(Pa),p_{a5}	101170.321 (101170.6)
3.	Air dry/wet bulb temperature at 5(K),t_{a5}	306.7647 (306.7645)
4.	Transfer coefficient for the rain zone, Me_{rz}	0.264781 (0.2851664)
5.	Transfer coefficient for the fill zone, Me_{fi}	0.886219 (0.8866959)
6.	Transfer coefficient for the spray zone, Me_{sp}	0.102264 (0.10231)
7.	Total transfer coefficient / Merkel number for the cooling tower, Me_T	1.253264 (1.274172)
8.	Merkel number by Chebyshev's formula, Me_C	1.27580 (1.274150)
9.	Actual fan shaft power (W),P_F	69242.37 (69222.04)
10.	Water lost due to evaporation (kg/s),m_{wevap}	7.4834 (7.4834)
11.	Mass flow rate ratio at inlet (m_{av1}/m_w)	1.0631
12.	Evaporation loss of water (kg/s), m_{wevap}	1.8164
13.	Exergy destruction (W), X_d	2260169.503
14.	Second law efficiency, η_{II}	0.9204
15.	Thermal efficiency of the cooling tower, η_{th}	0.6777

The output values are compared with the values (given in bracket) reported in literature. The results match reasonably well.

4.4 Parametric Study

In the parametric study, the variables selected are; wet bulb temperature of inlet air, droplet diameter in rain zone. The effect of variation in these parameters on cooling tower performance is discussed below.

Effect of variation in wet bulb temperature of inlet air

The effect of variations of wet bulb temperature of inlet air is on various performance parameters is shown in Table 4.3.The variation in the wet bulb temperature of inlet air is done in reference to the base case (Run 4) described in Section 4.3.

Table 4.3 Effect of variation in wet bulb temperature of inlet air.

	t_{wb1} (K)	t_{a5} (K)	m_{av1}/m_w	m_{wevap} (%)	P_F (W)	η_{th}	X_d (W)	η_{II}
Run 1	292.15	302.5997	1.0809	2.0004	70486.71	0.4970	4119781	0.8552
Run 2	294.15	303.9592	1.0752	1.9399	70082.76	0.5455	3498385	0.877
Run 3	296.15	305.3476	1.0693	1.8785	69668.17	0.6044	2877996	0.8987
Run 4	**298.15**	**306.7647**	**1.0631**	**1.8164**	**69242.37**	**0.6777**	**2260170**	**0.9204**
Run 5	300.15	308.2106	1.0568	1.7535	68804.93	0.7712	1646612	0.9419
Run 6	302.15	309.6848	1.0503	1.6898	68355.37	0.8946	1039240	0.9633
Run 7	303.4677	310.6714	1.0459	1.6476	68052.33	1	643454.6	0.9773

(a) Air outlet temperature v/s wet bulb temperature of inlet air

The effect of variation of wet bulb temperature of inlet air on air outlet temperature is shown in Figure 4.3.

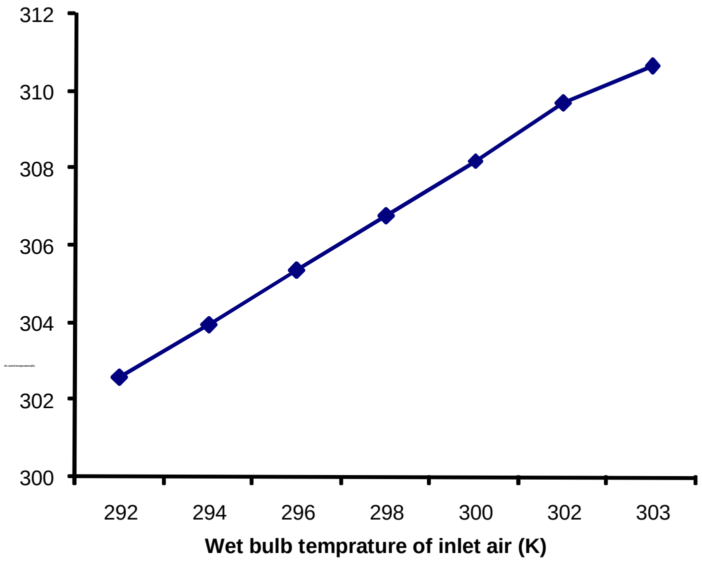


Figure 4.3. Air outlet temperature v/s wet bulb temperature of inlet air.

It is observed that from Run 1 to Run 3, air outlet temperature is less than the air inlet dry bulb temperature whereas from Run 4 to Run 7, air outlet temperature increases with increase in wet bulb temperature of inlet air. Thus, from Run 1 to Run 3, both air and water are cooled. This is possible in very hot and extreme dry conditions, because of the latent heat transfer from water to air ($w_{sw} > w$) and sensible heat transfer from air to water ($t_a > t_w$). The net enthalpy transfer is from water to air since $i_{masw} > i_{ma}$.

(b) Variation in inlet mass flow rate ratio with wet bulb temperature of inlet air

The effect of variation of wet bulb temperature of inlet air on inlet mass flow rate ratio is shown in Figure 4.4.

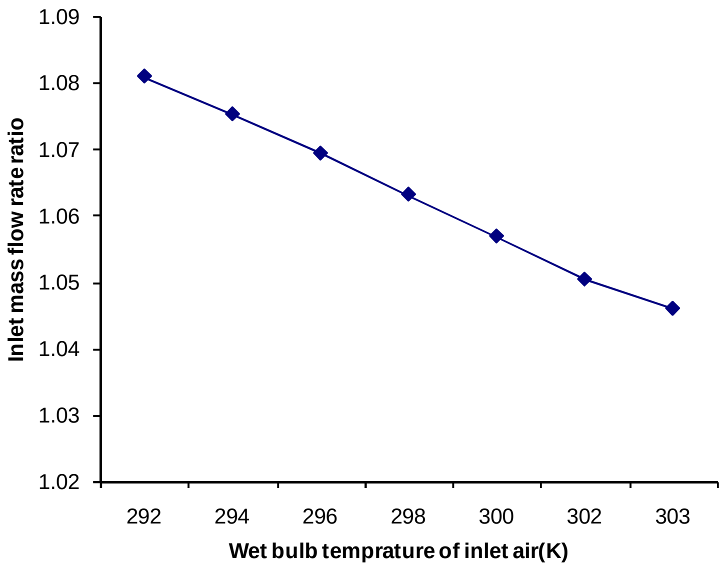


Figure 4.4. Inlet mass flow rate ratio v/s wet bulb temperature of inlet air

It is observed that inlet mass flow rate ratio decreases with increase in wet bulb temperature of inlet air. It is because of decrease in density of air-vapor mixture at inlet.

(c) Evaporation loss(%) v/s wet bulb temperature of inlet air

The effect of variation of wet bulb temperature of inlet air on evaporation loss is shown in Figure 4.5.

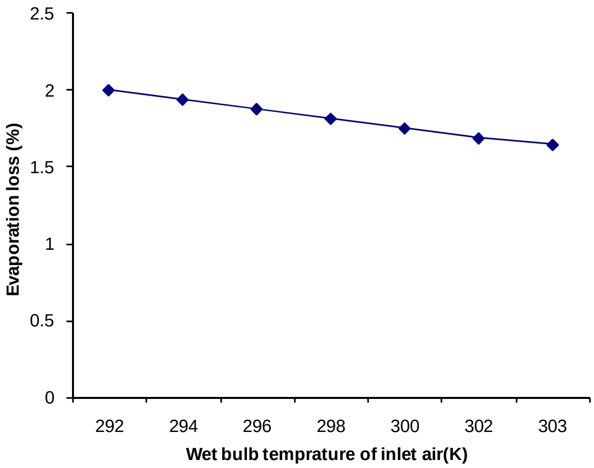


Figure 4.5. Evaporation loss v/s wet bulb temperature of inlet air

It is observed that evaporation loss (%) decreases with increase in wet bulb temperature of inlet air. It is because of increase in air outlet temperature due to which sensible heat component increases and latent heat component decreases resulting in reduction in evaporation loss.

(d) Actual fan shaft power v/s wet bulb temperature of inlet air

The effect of variation of wet bulb temperature of inlet air on actual fan shaft power is shown in Figure 4.6.

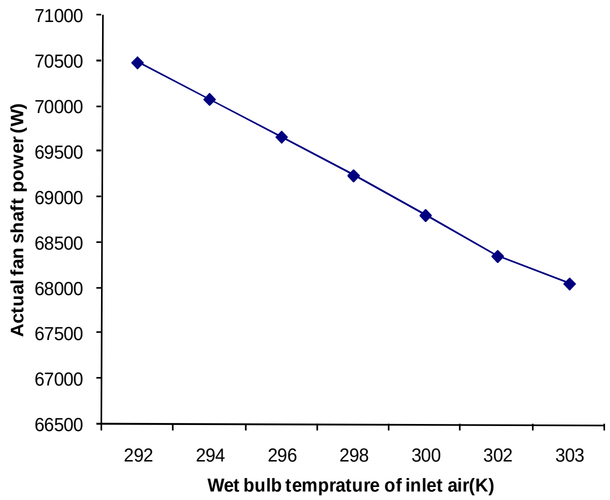


Figure 4.6. Actual fan shaft power v/s wet bulb temperature of inlet air

It is observed that fan shaft power decreases with increase in wet bulb temperature of inlet air. The increase in wet bulb temperature of inlet air causes decrease in mass flow rate and density of air-vapor; the net relative effect is reduction of volume flow rate of air. Since fan shaft power is a function of volume flow rate of air, so fan shaft power decreases with decrease in volume flow rate of air.

(e) <u>Thermal efficiency v/s wet bulb temperature of inlet air</u>

The effect of variation of wet bulb temperature of inlet air on thermal efficiency of the cooling tower is shown in Figure 4.7.

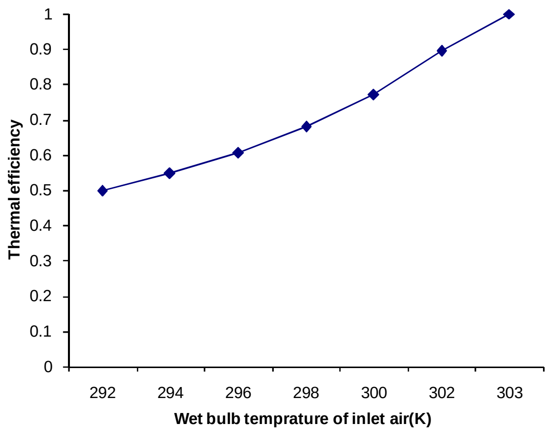


Figure 4.7. Thermal efficiency v/s wet bulb temperature of inlet air

It is observed that thermal efficiency of the cooling tower increases continuously with increase in wet bulb temperature of inlet air.

(f) <u>Exergy destruction v/s wet bulb temperature of inlet air</u>

The effect of variation of wet bulb temperature of inlet air on exergy destruction in the cooling tower is shown in Figure 4.8.

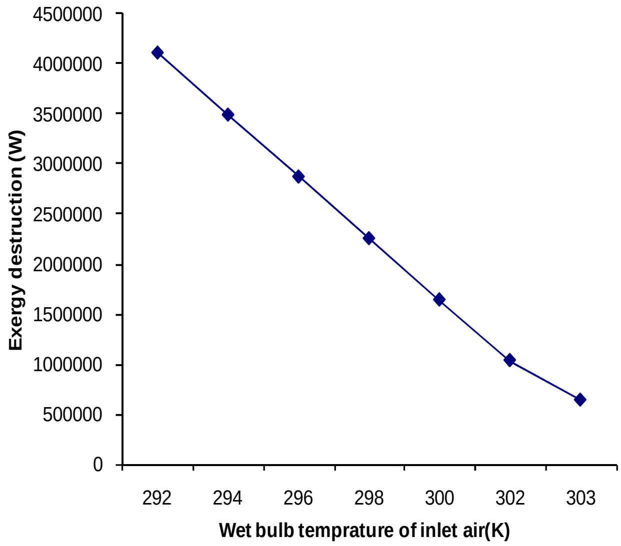


Figure 4.8. Exergy destruction v/s wet bulb temperature of inlet air

It is observed that exergy destruction in the cooling tower decreases continuously with increase in wet bulb temperature of inlet air. It is because that the exergy of outlet air stream increases continuously due to higher air outlet temperature. Also, since the evaporation loss decreases with the increase in wet bulb temperature of inlet air, exergy of makeup water also decreases. Since, the exergy of inlet and outlet water remains constant, the net result is decrease in the exergy destruction with increase in wet bulb temperature of inlet air.

(g) <u>Second law efficiency v/s wet bulb temperature of inlet air</u>

The effect of variation of wet bulb temperature of inlet air on second law efficiency is shown in Figure 4.9.

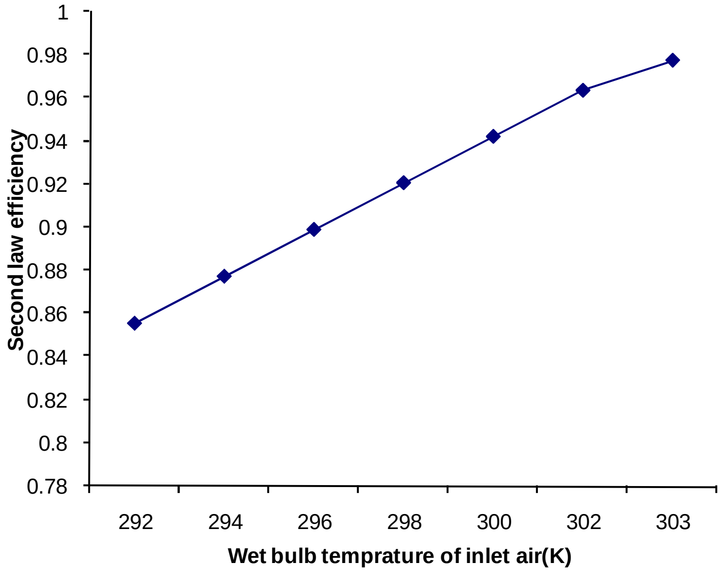


Figure 4.9. Second law efficiency v/s wet bulb temperature of inlet air

It is observed that second law efficiency increases continuously with increase in wet bulb temperature of inlet air. It is because of decrease in exergy destruction as discussed earlier.

Effect of variation in droplet diameter in rain zone

The effect of variations of droplet diameter is shown in Table 4.4. The variation in the droplet diameter is done in reference to the base case (Run 4) described in Section 4.3.The variation in droplet diameter is 3.5±0.5 mm.

Table 4.4. Effect of variation in droplet diameter.

	d_d (m)	t_{a5} (K)	m_{av1}/m_w	m_{wevap} (%)	P_F (W)	η_{th}	X_d (W)	η_{II}
Run 1	0.0030	306.7936	1.0589	1.8151	69331.78	0.6777	2248025	0.9208
Run 2	**0.0035**	**306.7647**	**1.0631**	**1.8164**	**69242.37**	**0.6777**	**2260170**	**0.9204**
Run 3	0.0040	306.7438	1.0663	1.8173	69175.62	0.6777	2269019	0.9201

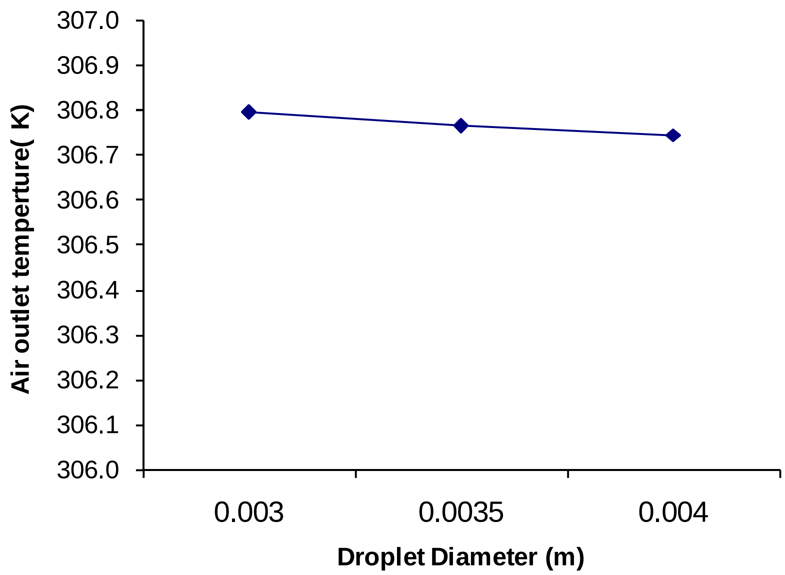


Figure 4.10. Air outlet temperature v/s droplet diameter.

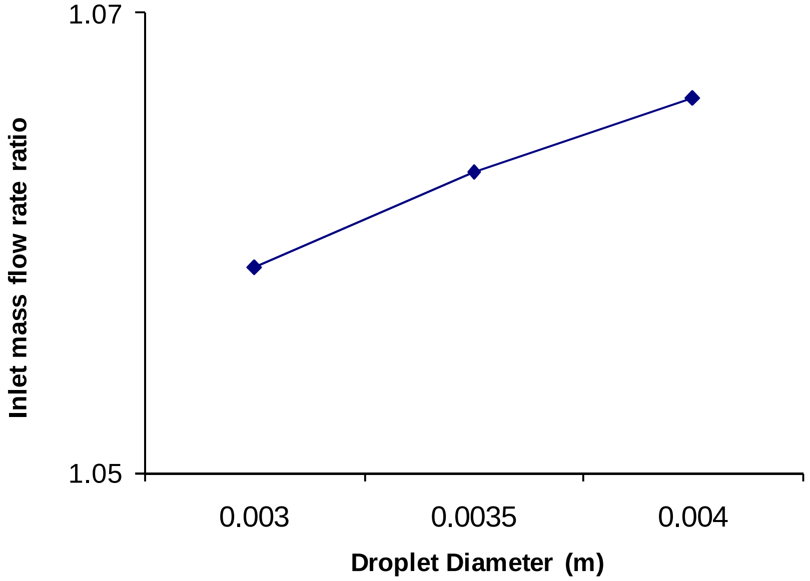


Figure 4.11. Inlet mass flow rate ratio v/s droplet diameter

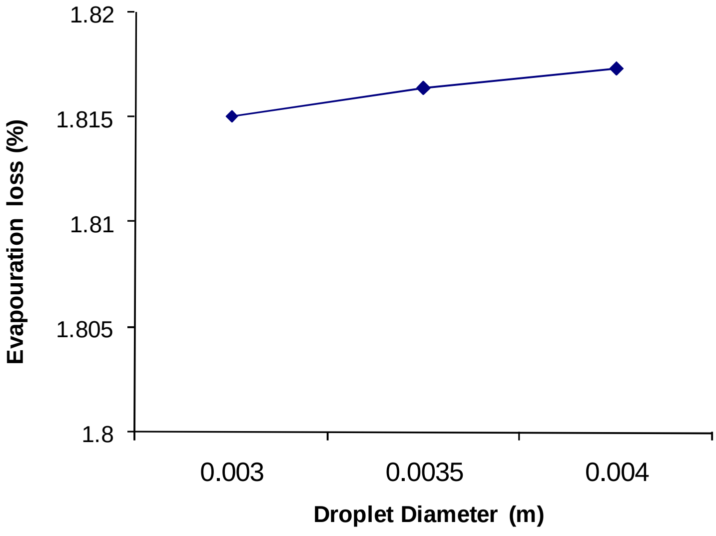


Figure 4.12. Evaporation loss v/s droplet diameter.

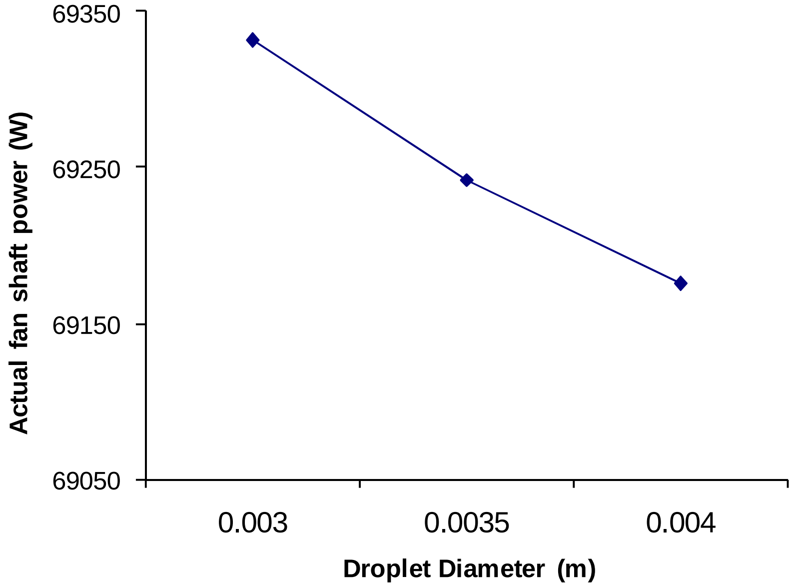


Figure 4.13. Actual fan shaft power v/s droplet diameter

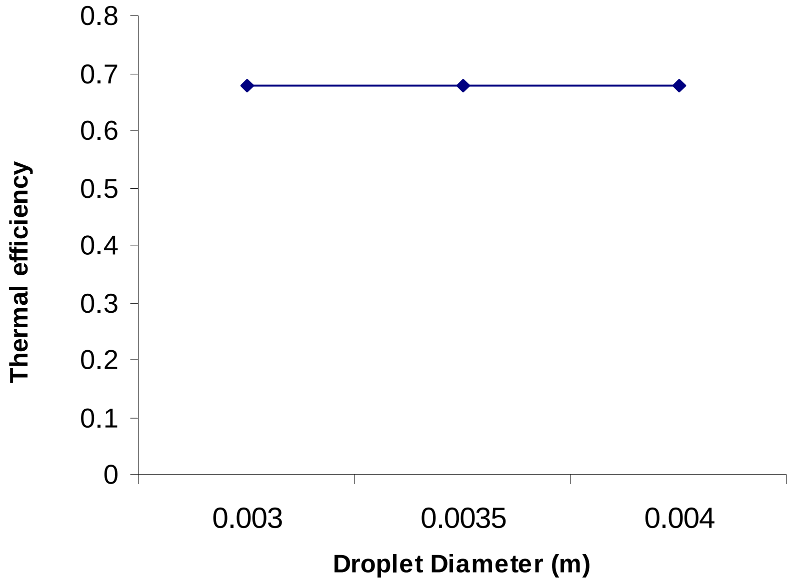


Figure 4.14. Thermal efficiency v/s droplet diameter.

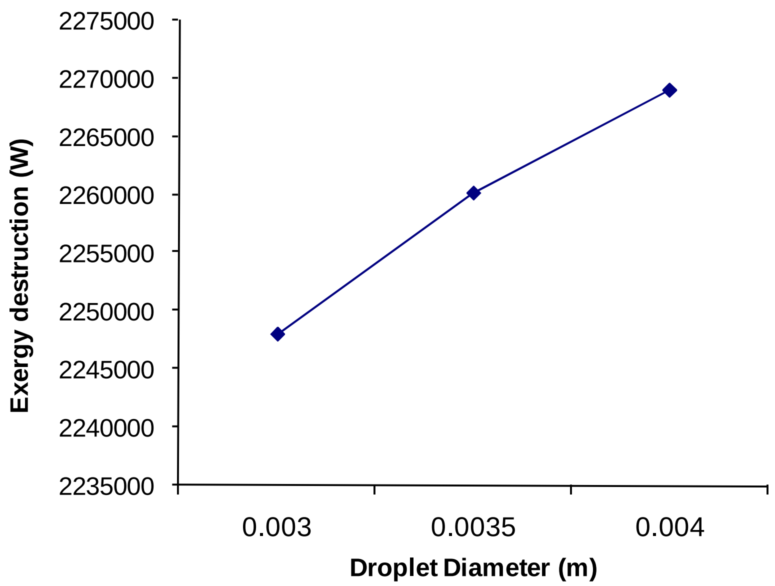


Figure 4.15. Exergy destruction v/s droplet diameter

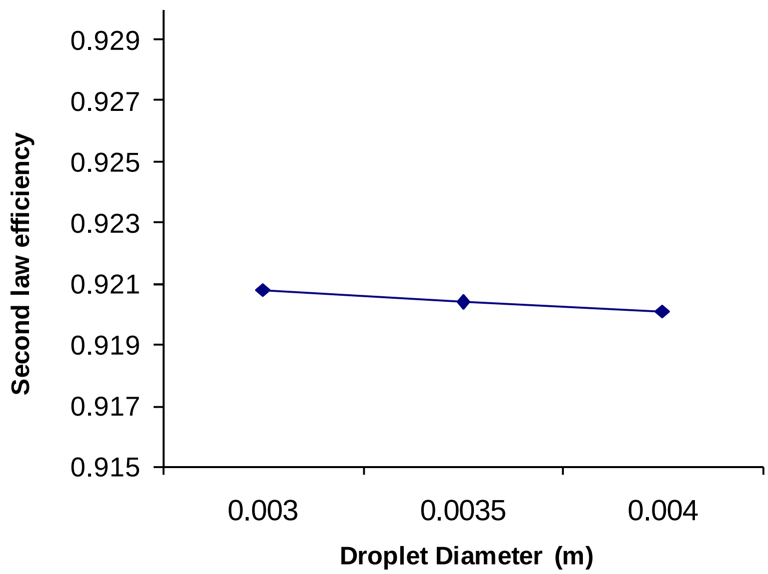


Figure 4.16. Second law efficiency v/s droplet diameter.

The effect of variation of droplet diameter in rain zone of the cooling tower on various parameters is shown in Figures 4.11 to 4.16.The variation in parameters shown in these figures is negligible.

Chapter 5

CONCLUSION & RECOMMENDATIONS FOR FUTURE WORK

This chapter describes the conclusion drawn from the results and discussions given in Chapter 4 followed by recommendations for the future work.

5.1 Conclusion

Following conclusion can be drawn from the present work:

1. For a given cooling tower load (mass flow rate of water and cooling range), the model successfully predicts the air outlet conditions, fan power requirements, make up water requirement and various evaluation parameters such as mass flow rate ratio, thermal efficiency, exergy destruction and second law efficiency.
2. The wet bulb temperature of inlet air plays a significant role on overall performance of the induced draft cooling tower.
3. From parametric study, it may be concluded that increase in wet bulb temperature of inlet air causes increase in air outlet temperature, thermal efficiency and second law efficiency and decrease in inlet mass flow rate ratio,evaporation loss, fan power and exergy destruction.
4. Droplet diameter in the rain zone has no significant role in the performance of cooling tower.
5. The present model can be successfully applied for air conditioning and power plant applications for wide range of parameters.

5.2 Recommendations for future work

Following recommendations are made for the future work related to the present model.

1. In the present model, Merkel method using Chebyshev formula has been used. The main limitation of this model is that it predicts air outlet temperature in saturated condition.This assumption can be relaxed by using sophasticated models like Poppe method etc., which also predicts the variation in temperature with respect to height of the cooling tower.
2. Empirical relations have been used in rain zone and spray zone to calculate the transfer coefficient. The empirical relations can be substituted by a detailed spray and rain zone model [12].
3. The effect of pressure and loss coefficients should be studied in detail to optimize the performance and enhance the applicability of the present model.The loss coefficients for wide geometries should be incorporated in the model.

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Thermo-Physical Properties

The thermo-physical properties summarized here are presented in Kroger [16].

All the temperature is expressed in Kelvin.

Thermo-physical properties of Dry air from 220 K to 380 K at standard atmospheric pressure of 101325 Pa.

Density, (kg/m^3)

$$\rho_a = p_a/(287.08\,T) \tag{A.1.1}$$

Specific heat, (J/kg K)

$$c_{pa} = 1.045356 \times 10^{3} - 3.161783 \times 10^{-1}\,T + 7.083814 \times 10^{-4}\,T^{2} - 2.705209 \times 10^{-7}\,T^{3} \tag{A.1.2}$$

Dynamic viscosity, (kg/sm)

$$\mu_a = 2.287973 \times 10^{-6} + 6.259793 \times 10^{-8}\,T - 3.131956 \times 10^{-11}\,T^{2} + 8.15038 \times 10^{-15}\,T^{3} \tag{A.1.3}$$

Thermo-physical properties of Saturated Water Vapor from 273.15 K to 380 K

Vapor pressure, (N/m^2)

$$p_v = 10^{z} \tag{A.2.1}$$

$$z = 10.79586\,(1-273.16/T) + 5.02808\,\log_{10}(273.16/T) + 1.50474 \times 10^{-4}\,[(1-10^{-8.29692\,\{(T/273.16)-1\}}] + 4.2873 \times 10^{-4} \times [10^{(4.76955(1-273.16/T)} - 1] + 2.786118312$$

Specific heat, (J/kg K)

$$c_{pv} = 1.3605 \times 10^{3} + 2.31334\,T - 2.46784 \times 10^{-10}\,T^{5} + 5.91332 \times 10^{-13}\,T^{6} \tag{A.2.2}$$

Dynamic viscosity, (kg/sm)

$$\mu_v = 2.562435 \times 10^{-6} + 1.816683 \times 10^{-8}\,T + 2.579066 \times 10^{-11}\,T^{2} - 1.067299 \times 10^{-14}\,T^{3} \tag{A.2.3}$$

Thermo-physical properties of mixture of Air and Water Vapor

Density , (kg air-vapor/m^3) $\rho_{av} = (1+w) [(1 – w/(w+0.62198)] p_{abs}) / (287.08\ T)$ (A.3.1)

Specific heat, (J/K kg dry air) $c_{pav} = (c_{pa} + w \cdot c_{pv})$ (A.3.2b)

Dynamic viscosity, (kg/sm) $\mu_{av} = (X_a \mu_a M_a^{0.5} + X_v \mu_v M_v^{0.5}) / (X_a M_a^{0.5} + X_v M_v^{0.5})$ (A.3.3)

Where $M_a = 28.97$ kg/mole ; $M_v = 18.016$ kg/mole

$X_a = 1/(1+1.608w)$; $X_v = w/(w+0.622)$

Humidity ratio, (kg/kg of dry air) $w = [(2501.6 – 2.3263 (t_{wb} –273.15) / (2501.6 + 1.8577(T –273.15)- 4.184\ x(twb –273.15)] [(0.62509\ p_{vwb})/(p_{abs} – (1.005\ p_{vwb})] – [1.00416(T – t_{wb})/(2501.6 +1.8577(T – 273.15) – 4.184\ x\ (t_{wb} –273.15))]$ (A.3.5)

Enthalpy, (J/kg of dry air) $i_{ma} = c_{pa} (T – 273.15) + w [i_{fgw0} + c_{pv} (T – 273.15)]$ (A.3.6b)

where the specific heats are calculated at (T+273.15)/2 and latent heat, i_{fgw0} is evaluated at 273.15 K.

Thermo-physical properties of Saturated Water Liquid from 273.15 K to 380 K

Density, (kg/m^3) $\rho_w = 1 /(1.49343 \times 10^{-3} – 3.7164 \times 10^{-6} T + 7.09782 \times 10^{-9} T^2 – 1.90321 \times 10^{-20} T^6)$ (A.4.1)

Specific heat, (J/kg K) $c_{pw} = 8.15599 \times 10^3 – 2.80627 \times 10T + 5.11283 \times 10^{-2} T^2 –2.17582 \times 10^{-13} T^6$ (A.4.2)

Latent heat, (J/kg) $i_{fgw} = 3.4831814 \times 10^6 – 5.8627703 \times 10^3 T + 12.139568 T^2 – 1.40290431 \times 10^{-2} T^3$ (A.4.5)

Surface tension, (N/m) $\sigma_w = 5.148103 \times 10^{-2} + 3.998714 \times 10^{-4} T – 1.4721869 \times 10^{-6} T^2 + 1.21405335 \times 10^{-9} T^3$ (A.4.7)