Heat and Mass Transfer Analysis of Mechanical Draught Counter flow Wet Cooling Towers

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In partial fulfillment of the requirements for the award of the degree of

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In
Thermal Engineering

By
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CERTIFICATE

This is to certify that the project entitled ***"Heat and Mass Transfer Analysis of Mechanical Draught Counter Flow Wet Cooling Tower",*** which is being submitted by Mr. Sunil Soni, is a bonafide record of student's own work carried by him under our guidance and supervision in partial fulfillment of requirement for the award of the Degree of Master of Engineering in Thermal Engineering, Department of Mechanical Engineering, Delhi College of Engineering, University of Delhi.

The matter embodied in this project has not been submitted for the award of any other degree.

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Abstract

Counter flow wet cooling towers are widely used in power plant and air conditioning application. The function of the cooling tower is to reduce the temperature of circulating water so that it may be reused in condensers and other heat transfer equipments. The hot water in the cooling tower is cooled by evaporation cooling mechanism.

In the present work, heat and mass transfer equations used in Merkel, e-NTU and Poppe methods are discussed. A computer programs based on these methods are developed to solve sizing and performance analysis problem of cooling towers. The model is capable for calculating air temperature, humidity ratio, cooling tower outlet temperature, and evaporation loss and tower volume. This model also considers variation in Lewis number and evaporation loss.

The differences in the heat and mass transfer analyses and solution techniques of the Merkel, e- NTU and Poppe methods are discussed for different input atmospheric conditions with the help of enthalpy diagrams, percentage error variation graphs for Merkel number and for tower outlet temperature.

The comparison of Merkel and e-NTU shows almost similar results because these methods are based on similar simplifying assumptions. The comparison of Merkel, e-NTU and Poppe methods for different initial conditions shows that variation in Merkel number calculated by Poppe method varies from 0.2 % to 1.8 %. Percentage average error variation in tower volume for Poppe and Merkel methods is 16.3 %. For Poppe method percentage errors in outlet water temperature varies 2.53 % to 5.87 % and percentage evaporation varies from 1.26 % to 4.62 %. Outlet water temperature remains smaller for Merkel Method than calculated by Poppe method.

Key words: Cooling tower, Merkel number, Tower volume, Evaporation loss, Lewis number, outlet water temperature, Percentage error in tower volume.

Nomenclature

A	Heat transfer surface area, m^2 or Approach, °C
a	Surface area per unit volume, m^2
C	fluid capacity ratio = mC_p, kJ/s-ºC
C_R	Capacity ratio = C_{mim}/C_{max}
c_p	Specific heat at constant pressure, kJ/kg K
d	Differential element
E	Error
e	Thermal effectiveness = q_{act}/q_{max}
f	Enthalpy correction factor, kJ/kg (Also represented by δ)
f’	Slope of saturated air enthalpy versus temperature curve (di_{masw}/dT), kJ/kg-ºC
G	Mass velocity, kg/m^2 s (Also represented by m_a)
h	Heat-transfer coefficient, W/m^2 K
h_d	Mass transfer coefficient, kg/m^2 s (Also represented by K_m)
i	Enthalpy, kJ/kg
i_{ma}	Enthalpy of dry air at wet bulb temperature of air, kJ/kg
i_{masw}	Enthalpy of saturated air at the local bulk water temperature, kJ/kg
i_{fg}	Latent heat, kJ/ kg
Δi	Enthalpy difference between air at interface and local bulk air, kJ/kg
L	Length (m) and Water mass flow rate kg/sec (Also represented by m_w)
L_{ef}	Dimensionless Lewis factor
m	Mass flow rate, kg/s
Me	Merkel number
M_w^+	Water side capacity ratio, kg/s
n	Number of increments (Also represented by N)
NTU	Number of transfer units, unit less (UA/C_{min})
P_b	Atmospheric pressure, (N/m^2)
P_v	Vapor pressure, N/m^2
Q	Heat transfer rate, W (Also represented by q)
R	Water to air mass flow rate or Cooling range, °C
T	Temperature, °C or K

U Overall heat-transfer coefficient, W/m^2 K

V Cooling tower packing volume, m^3

W Humidity ratio, kg water vapor/kg dry air

w_{sa} Humidity ratio of saturated air at T_a ,unit less

w_{sw} Saturation humidity ratio of air evaluated at the local bulk water temperature,

z Elevation, m

ad Fill characteristic coefficients

bda Fill characteristic coefficients

bdb Fill characteristic coefficients

ap Fill characteristic coefficients

bpa Fill characteristic coefficients

bpb Fill characteristic coefficients

bpc Fill characteristic coefficients

ATD Height of fill

M_{er} Merkel number required

M_{ea} Merkel number available

α Heat transfer coefficient, kW/m^2-s

β Water film thickness, m

Subscripts

a Air

act Actual heat transfer

av Average

c Cold or convective heat transfer coefficient

e e-NTU approach

fi Fill

fr Frontal

h Hot

i Inlet

M Merkel approach

m Mean

max	Maximum
min	Minimum
o	Outlet
P	Poppe approach
s	Saturation
ss	Supersaturated
v	Vapor
w	Water
wb	Wet bulb
1	Air or water inlet conditions
2	Air or water outlet conditions

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Chapter 1
Introduction

1.1 Function and Applications

The function of the cooling tower is to reduce the temperature of circulating water so that it may be reduce in condensers and other heat exchanger equipment. Direct contact or wet counter flow cooling towers are found principally in thermal power plants and air conditioning applications.

The water distribution in the tower by spray nozzles, splash bars, or film fill in a manner that exposes a very large water surface to atmosphere air. The movement of the air is accomplished by fans (Mechanical draft), Natural draft, or the induction effect from water sprays. In a cooling tower, water is put in direct contact with surrounding air. A small part of the cooling water, i.e. 1-2%, evaporates. This evaporation causes an increase in temperature and humidity of the air and a decrease of the temperature of the water. By using the evaporation, it is possible to cool below the normal air temperature. The minimal realizable temperature is the wet bulb temperature.

1.2 Types of Cooling Tower

In direct contact evaporative cooling tower hot water from the heat source is sprayed into the top of a large chamber containing packing (fill).The water flow and air stream enter as vertically opposite directions. The warmed air leaving the fill, maximizing evaporation and reducing energy costs. Wet cooling towers may be classified as natural draught and mechanical draught cooling tower. These are described below:

(a). Natural Draught Counter flow cooling Tower

The schematic of natural draught cooling tower is shown in Fig 1.1. These cooling towers are dependent upon atmospheric conditions as they do not have a mechanical device to create a flow of air through the tower. They are built high to ensure that discharged plume or drift does not recycle into a tower's air inlets. These are used in the power generation industry where the heat loads are high

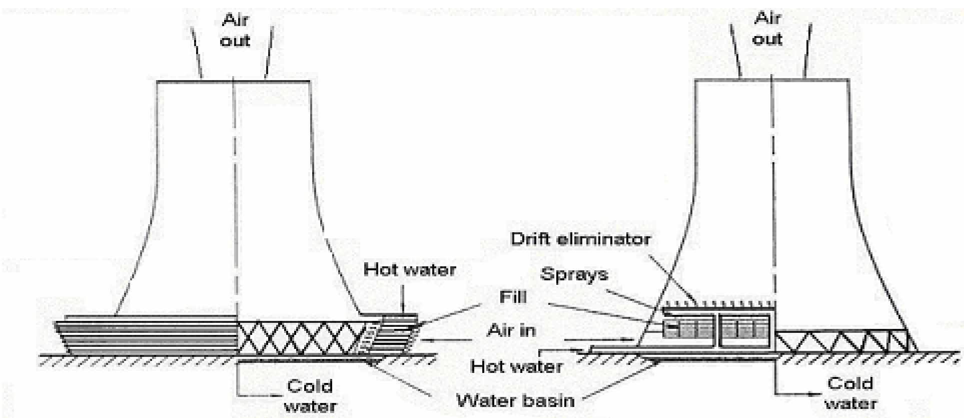


Fig. 1.1 Natural Draught Counter Flow Cooling Tower

(b). Mechanical Draught Counter Flow Cooling Tower

This type of cooling tower utilizes a motor-driven fan to move air through the tower, the fan being an integral part of the tower making the thermal performance more stable than the natural draught tower. Mechanical draft cooling towers are classified as either forced draft or induced draught.

Forced draught towers are designed to have high air entry velocities and low air exit velocities making them susceptible to recirculation. The fan is located in the air inlet to the tower and air is blown or forced through the tower. A schematic of forced draught cooling tower with air water temperature distribution is shown in Figure 1.2.

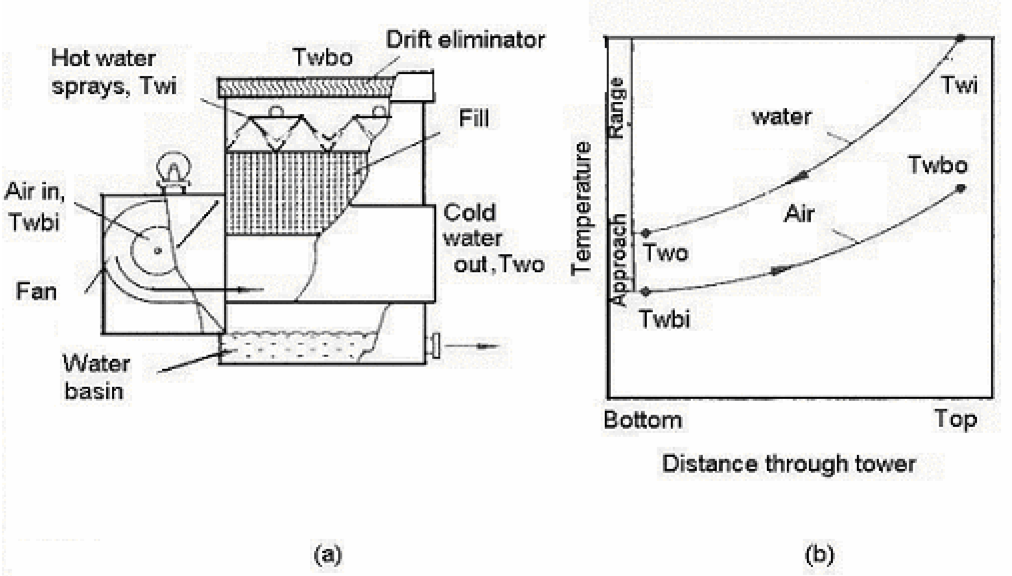


Fig. 1.2 Forced Draft Cooling Tower (a) Cooling Tower (b) Temperature Distribution

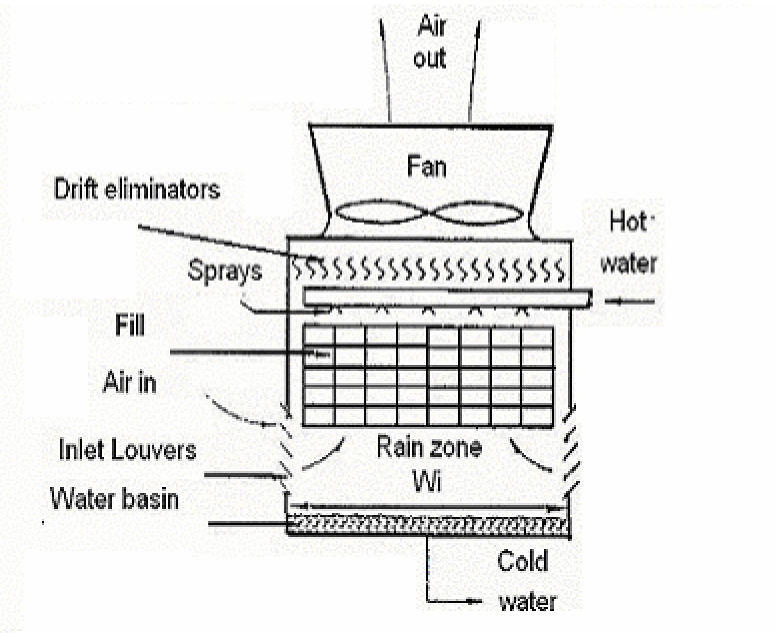


Fig. 1.4 Induced Draught Counter Flow Cooling Tower

Induced draft cooling tower Fig. 1.4 has the fan located in the exit air stream, usually at the top of the tower. The discharge air velocities are three to four times higher than the air entering velocity due to the large intake area, making these towers less prone to recirculation than forced draft towers.

1. 3 Important Terms of Cooling Tower

Various terms applicable to cooling towers are graphically shown in Fig. 1.5 are described below:

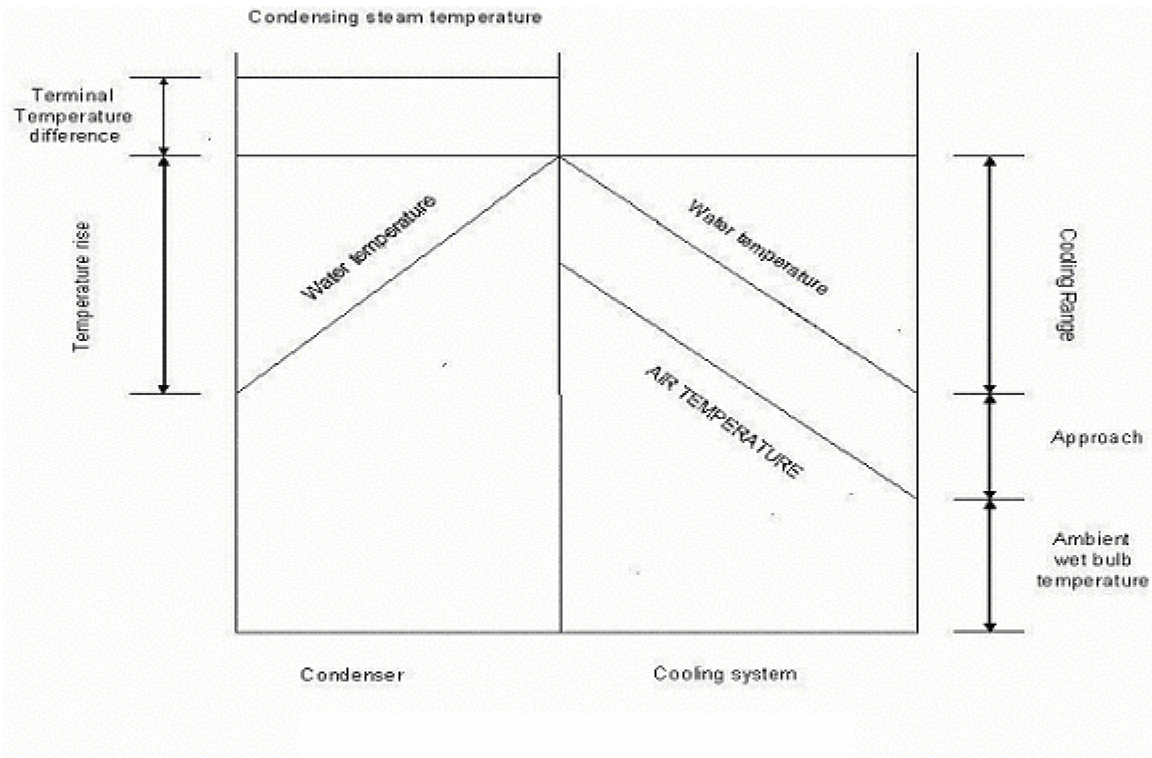


Fig. 1.5 Cooling Tower Temperature Variation

(a). Range (R): It is the difference of hot water inlet to cold water outlet temperature.

(b). Approach (A): It is the difference of the cold water temperature to wet bulb temperature.

(c). Cooling efficiency (E): It is defined as the ratio of the range to difference of hot water inlet temperature and wet bulb temperature.

(d). Humidity ratio (w): It is defined as the ratio of mass of water vapor to the mass of the dry air in a given volume of the air-water mixture.

(e).Wet bulb temperature (t_{wb}): It is defined as the temperature at which the water by evaporating in to moist air at a given dry bulb temperature and humidity ratio can bring air to saturation adiabatically at same wet bulb temperature while the total pressure is maintained constant.

(f). Relative humidity (φ): It is defined as the ratio of actual mass of water vapor in a given volume of moist air to the mass of water vapor in the same volume of saturated air at the same temperature and pressure.

(g). Dry bulb temperature (t_a): It is the temperature of air recorded by a thermometer, when it is not affected by the moisture present in the air.

(h). Lewis number (L_e): It is the ratio of thermal diffusivity (α) to mass diffusivity (D). Temperature and concentration profiles will be same if Lewis number is equal to unity. For air water system it is approximately equal to one.

1.4 Scope of Present Research Work

Objective of this project is to develop numerical model for design and performance analysis of counter flow wet cooling tower. The model is capable for calculating air temperature, humidity ratio, cooling tower outlet temperature, and evaporation loss and tower volume. This model also considers variation in Lewis number and evaporation loss. This model gives accurate results for different ambient condition, for different inlet water temperature, wet bulb temperature, approach, range and dry bulb temperature.

This numerical model also considers different fill design by using fill characteristic coefficient. This study gives a detailed derivation of the heat and mass transfer equation of the evaporative cooling in wet-cooling towers. The governing equations of the rigorous Poppe method of analysis are derived from first principles. The governing equations of the Merkel method of analysis are subsequently derived after some simplifying assumptions are made. The equations of the effectiveness – NTU method applied to wet cooling towers are also presented.

The differences in the heat and mass transfer analyses and solution techniques of the Merkel, e- NTU and Poppe methods are derived for different input atmospheric conditions with the help of enthalpy diagrams, percentage error variation graphs for Merkel number (represent tower volume) and for tower outlet temperature and psychrometric charts. This study is very helpful for cooling tower designer and analyzer for deciding which method gives good results in different input atmospheric condition with time and cost constraint keeping in mind. This study gives optimize design for different conditions. This project work has been validated with available data of Sutherland [1], Li and Priddy's handbook [2] and Jaber and Webb [3] data, gives quite précis results.

Chapter 2
Literature Review

Heat and mass transfer analysis of mechanical draught cooling tower has been extensively studied by several researchers. A summary of these studies is described below:

2.1 Summary of Literature Review

The art of evaporative cooling is quit ancient, although it is only relative recently that it has been studied recently in a scientific manner. Merkel developed the theory for the thermal evaluation of cooling towers in 1925, who developed the basic equations for total mass and energy transfer and considered each process separately.

The Merkel theory relies on several critical assumptions to reduce the solution to a simple hand calculation .Because of these assumptions; however, the Merkel method does not accurately represent the physics of heat and mass process in the cooling tower fill.
The critical simplifying assumptions of the Merkel theory are given below:

(a). The Lewis factor, Le_f, relating heat and mass transfer is equal to 1.
(b). The air exiting the tower is saturated with water vapor and it is characterized only by its enthalpy.
(c). The reduction of water flow rate by evaporation is neglected in the energy balance.

Jaber and Webb [3] developed the equations necessary to apply the e-NTU method directly applicable to counter flow or cross flow cooling towers. This approach is particularly useful in the latter case and simples the method of solution when compared to more conventional numerical method. The e-NTU method is based the same simplifying assumptions as described the Merkel method.

The Poppe model was developed by Poppe and Rogener [4]. The method of Poppe does not make the simplifying assumptions made by Merkel. The critical differences between the Merkel and Poppe methods are investigated by Kloppers and Kroger [5]. The objective of this investigation is to include the e-NTU method in the investigation.

J.W. Sutherland [1] compared accurate analysis of mechanical draught counter flow cooling tower, including water loss by evaporation, with the approximate common method based on enthalpy driving force developed by Merkel in 1925 for wide range of inlet water and air conditions for substantial underestimation of tower volume of around from 5 to 15 % are obtained when Merkel method is used, indicating the possibility of quite serious consequences as far as cooling tower design is concerned

J.C. Kroger and D.G. Klopper [5] has described that the heat rejected and water evaporated in mechanical and natural draft cooling towers are critically evaluated by employing the Merkel, Poppe, and e-NTU methods of analysis, respectively, at different operating and ambient conditions. The importance of using a particular method of analysis when evaluating the performance characteristics of a certain fill material and subsequently employing the same analytical approach to predict cooling tower performance is stressed. The effects of ambient humidity and temperature on the performance of cooling towers are evaluated by employing the Merkel, e-NTU, and Poppe methods.

J.C. Kroger and D.G. Klopper [6] gives study of a detailed derivation of the heat and mass transfer equation of evaporative cooling in wet cooling towers. The governing equations of the rigorous Poppe method of analysis are derived from first principles. The governing equations of the Merkel method of analysis are subsequently derived after some simplifying assumptions are made. The equations of the effective- NTU method applied to wet cooling towers are also presented. The differences in the heat and mass transfer analyses and solution techniques of the Merkel and Poppe methods are described with the aid of the enthalpy diagrams and psychrometric charts. The psychrometric chart is extended to accommodate air in the supersaturated state.

Kroger [7] has described thermal design of "Air cooled heat exchanger and cooling tower volume I and volume II". Analysis of natural draft cooling towers has been presented in an elaborate way in this book. A detailed derivation of the heat and mass transfer equation of evaporative cooling in wet cooling towers is described.

Mohiuddin and Kant [8] have laid a detailed selection and design procedure for wet counter flow and cross flow cooling towers. They have studied different models available in the literature like ESC code, FACTS, VERA2D, STAR, Sutherland's model, model by Fujita and Tezuka, Webb's model and model by Jaber and Webb. They found that each model makes use of a somewhat different set of assumptions. Consequetly, the results of calculations of heat – mass transfer coefficients from each one of the models also differ. Most of the codes are proprietary in nature hence their details are not available.

The ESC code is a one dimensional model, although for cross flow configurations it uses a two-dimensional matrix of the air and water flow, but treats the flow as one dimensional. FACTS code is more sophisticated than a one dimensional model, yet it contains simplifications that prevent it from being classified as a true two-dimensional code. An integral formulation of the conservation equations (conservation of mass and energy for both air and water) is applied , in conjunction with the Bernoulli equation (with head losses included).FACTS has the capability to model towers containing hybrid fills or fills that have voids or obstructions. To a limited extent, it can account for flow non-uniformities, for which FACTS offers the option of specifying a flow distribution of water at the tower inlet. It allows for the input of separate heat-mass transfer and pressure drop correlation for spray and rain regions in counter flow towers. The FACTS code package calculates volume of the cooling tower using the operating parameters m_a and m_w and known (or assumed values of mass transfer coefficient).

VERA2D code treats the flow of water in the cooling tower as one dimensional and the flow of the air as two dimensional and steady. Two dimensional, partial differential equations are solved for the conservation of mass and energy for both air and water and the conservation of momentum for moist air. It also calculates the distribution of airflow throughout the tower. The VERA2D code, because of the two-dimensional flow rates may be specified. The variation of air density through the tower is included as a function of the temperature and pressure. Evaporation of the water (which leads to non-uniform water distribution) is modeled. Heat transfer is related to both water temperature and ambient pressure. A local equilibrium model simulates turbulence.

Code STAR is applicable to counter flow and cross flow natural and mechanical draught cooling towers. It solves two dimensional differential equations of the fluid dynamics and the thermodynamics by applying a method of finite differences to a grid of rectangular mesh using a fractional step algorithm.

Model by Fujita and Tezuka [9] calculates the thermal performance of counter flow and cross flow mechanical draft cooling tower using the enthalpy potential theory. The method recommends the calculation of number of transfer units NTU = KaV/L, for the counter flow cooling towers by the CTI (Cooling Tower Institute) method (CTI code ATC 105). Then the NTU for cross flow tower can be calculated using a correction factor. Webb's model [10] outlines a design procedure for cooling towers. It is one dimensional model, which considers water loss by evaporation. The Lewis number is taken is equal to 0.87.

Mohiuddin and Kant [8] have calculated total packed height and the number of decks for the both type of packing splash and film type for different geometry and arrangement at different hot water inlet temperatures.

For splash type packing

KaV/L=0.07+ constant (A)*number of decks (ND)*$(L/G)^{-P}$.

Coefficients A and P related to different geometry.

K = mass coefficient per unit volume; L = Water flow rate

a = Surface area per unit volume; G = Air flow rate

As secondary effect causes some changes in the tower performance with a different water inlet temperature, a corrected value of KaV is used depending upon the inlet water temperature. This corrected value of KaV/L is calculated as follows:

KaV/L = 0.9x KaV/L For inlet water temperature $\leq$ 45°C

KaV/L = 1xKaV/L For inlet water temperature 45 - 55°C

KaV/L= KaV/L For inlet water temperature $\geq$ 55°C

Knowing the number of decks and the vertical deck spacing the total packed is involved in this set; the packed height of the tower is calculated directly using the relation of Lowe and Cristie (Ka/L) [9] by the equation of the form

$$Ka/L=k\,(L/G)^{-m}$$

Where k and m define the transfer characteristics of fill packing. The values of k and m are available in literature [9]. So the packing height can be calculated.

Milosavljevic Nenad at al. [11] have derived a mathematical model and a computer simulation program for performance prediction of as counter flow wet cooling tower based on one dimensional heat and mass transfer using the measured heat transfer coefficient. These equations have been solved numerically to predict the temperature and humidity of air at different heights of the packing. They have performed experiments on a pilot-cooling tower to analysis performance of different fills. To predict air flow distribution they have included the use of three dimensional version of the CFD code Fluent/UNS, version 4.2. Two dimensional CFD simulations have been used to predict the external air flow and recirculation around the tower.

Villiers et al. [12] examine the effect of non – uniform water distribution and theoretical analysis of natural draft cooling tower has been done with a CFD type model using an in-house natural draught cooling tower design package NDDES. The program allows variations in water loading and packing depth. Actual field tests were done on a large 900 MW natural draught cooling tower to conclude that cooling tower performance can be significantly affects water distribution and non uniform water distribution is a requirement in a large natural draught counter flow cooling tower. The results show that an improvement in performance of between 1.1°C and 1.3°C is achieved by changing water distribution and packing height. Improvement of 0.7 °C, in cold water temperature was obtained by having non- uniform water distribution only.

Fisenko et al. [13] have presented a mathematical model of evaporative cooling in natural draught cooling tower. They have neglected heat transfer in rain zone as drop size becomes too big and jets are formed. Heat transfer in the spray and fill zone has been investigated and they found that increase in height of sheets by more than 3 m does not increase the efficiency because the air becomes saturated. They have defined a nondimentional parameter $P=d * V_a / (D*h_f)$, where d is distance between sheets, V_a is velocity of air, D is diameter of the tower and h_f is height of sheets. Thermal efficiency of the film cooling does not practically increase when $p>10$.

Fisenko et al. [13] have simulated evaporative cooling of the water in mechanical draft cooling tower as a droplet flow. They found that the increase in the air velocity from 2 m/s to 4 m/s enhances efficiency by 25 to 33% (at given parameters). For all droplet sizes efficiency increases with increase in heighten to radius ratio, finer droplets give higher efficiency whereas increase L/G reduces tower efficiency. They later presented optimized air velocity versus humidity at different air inlet temperature. The curve becomes less steep as air temperature decreases.

Hawlader and Liu [14] have presented mathematical and physical model governing the flow mass and heat energy for an evaporative natural draught cooling tower. Average difference between measured and predicted temperature is 0.26°C.the simulation also proves that the main heat transfer takes place in the fill region where the percentage of latent heat transfer is predicted as 83% .However, about 90% of latent heat is transferred via evaporation in the rain although total heat transfer in rain region is very small in comparison to fill region. They have considered steady state two dimensional problems in r and x direction with uniform water distribution everywhere. So computational domain is half of the tower. The governing conservation equations were mass conservation of air and water, Momentum conservation equation of air flow in x and r direction, Energy conservation equation of air in the form of enthalpy, State equation of moist air.

Water flow has been treated in two-flow patterns continuous film flow and motion of discrete drops. Finite difference scheme with non-uniform grid is employed with more control volume located in fill and rain region, with grid points at centre of control volume. Grid independence has been shown above 1600 grids. They have analyzed with 85*45 = 3825 grids. The solution was assumed converged when temperature changes are less than 0.001 °C. Simulation results show that air flow is uniform in the radial direction. Water outlet temperature is non-uniform in the radial direction and is minimum at one fourth distance from the centre of the tower. Their prediction shows that both the local heat and mass transfer rate are non-uniformly distributed vertically and horizontally. It is seen that at each height level heat transfer rate decreases as the radius decreases(moving towards center) in vertical direction in the fill region the local heat transfer rate decreases as the height increases. However in the whole rain region the local heat transfer rate increases as it goes up from the basin. They have found that average heat transfer rate is about 72 kW/m^3 in fill region and 1 kW/m^3 in rain zone.

Bedekar et al. [15] have done experimental investigation of the performance of counter flow mechanical draft cooling tower to confirm that tower performance decreases with an increase in the L/G ratio. They have also mentioned of tower efficiency as range divided by water inlet minus wet bulb temperature.

Makkinejad [16] has presented a method to develop mathematical solution to cooling towers taking into account the main influencing parameters L/G ratio. In spray towers diameter of the equalized spheres is estimated as 63.2% of the passage diameter and is called Sauter diameter. The water distribution is assumed uniform and steady state analysis has been done for countercurrent cooling towers. It has been found that the water temperature falls with increase in Sauter diameter.

From the literature survey it can be concluded Merkel, e-NTU and Poppe methods are the most common & widely acclaimed by researchers. The formulation of these methods are described in chapter 3.

Chapter 3

Mathematical Model Formulation

This chapter discusses development of the formulation of Merkel, e-NTU method and Poppe method. Then governing equations of the rigorous Poppe method of analysis are derived from first principles. The governing equations of the Merkel method of analysis are subsequently derived after some simplifying assumptions are made. The equations of effective e-NTU method applied to wet cooling towers are also presented.

3.1 Poppe Method

The Poppe model was developed by Poppe and Rogener [4] in the early 1970s. The method of Poppe does not make the simplifying assumptions made by Merkel. The critical differences between the Merkel and Poppe methods are investigated by Kloppers and Kroger [6]. This method consider evaporation loss of water and Lewis factor not remain equal to one as for Merkel method. It works on realistic situations of cooling tower. Therefore it is most rigorous method among all other methods.

Consider an elementary control volume in the fill or packing of a counter flow wet-cooling tower (Fig. 3.1). Evaporation of the downward water occurs at the air-water interface where the air is saturated with has a lesser vapor concentration.

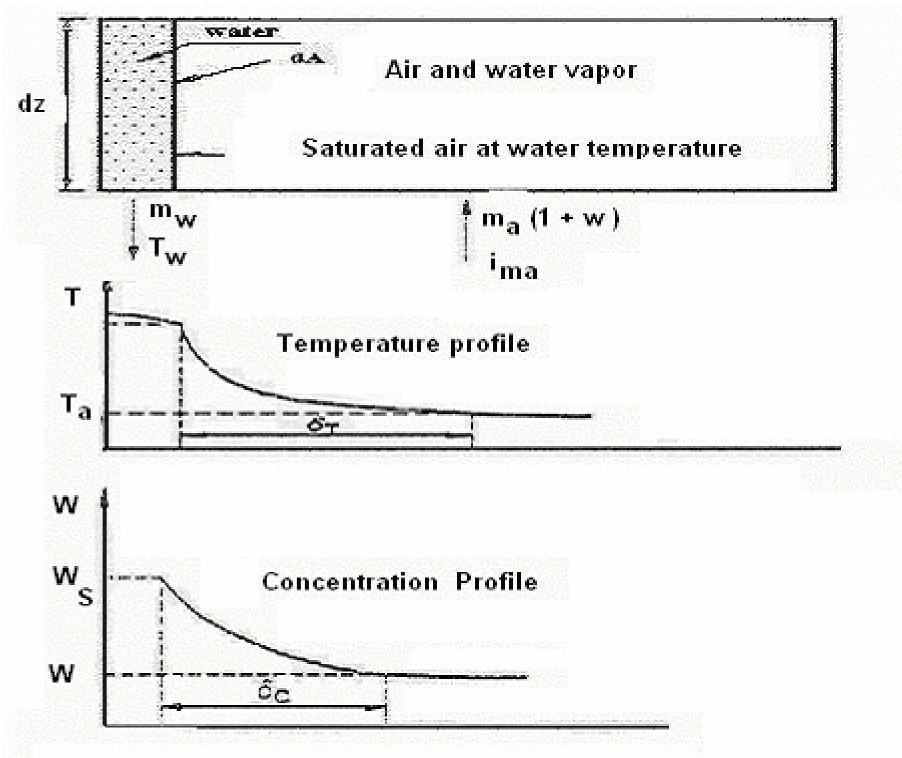


Fig 3.1 Control Volume For Deviation Of Governig equation For Counter Flow Fill

It will be assumed that the interface water temperature, T_s, is the same as the bulk water temperature, T_w. The effect of this assumption on the transfer process has been investigated by a number of researchers including Baker, Webb, and Marseille. Air and water properties at any horizontal cross section are assumed to be constant, and the area dA for heat and mass transfer is identical.

A mass balance for the control volume yields,

$$m_a(1+w) + mw + \left(m_w + \frac{dm_w}{dz}dz\right) = m_a\left[1 + \left(w + \frac{dw}{dz}dz\right)\right] + m_w$$

or

$$\frac{dm_w}{dz} = m_a \frac{dw}{dz} \tag{3.1}$$

Where

m_a = mass flow rate of the air constituent

Any energy balance for the control volume yields

$$m_a i_{ma} + \left(m_w + \frac{dm_w}{dz} dz \right) c_{pw} \left(T_w + \frac{dT_w}{dz} dz \right) = m_a \left(i_{ma} + \frac{di_{ma}}{dz} dz \right) + m_w c_{pw} T_w \quad (3.2)$$

(Where T_w is in °C.)

Neglecting second orders terms, Equation (3.2) simplifies to

$$m_w c_{pw} \frac{dT_w}{dz} + c_{pw} T_w \frac{dm_w}{dz} = m_a \frac{di_{ma}}{dz} \quad (3.3)$$

Where i_{ma} refers to the enthalpy of the air-water vapor mixture per unit mass of dry air, which is expressed as

$$i_{ma} = c_{pa} T_a + w \left(i_{fgwo} + c_{pv} T_a \right) \quad (3.4)$$

and where i_{fgwo} is evaluated at 0 °C and c_{pa} and c_{pv} at $T_a/2$ °C.

Substitute Equation 3.1 into Equation 3.3 to find

$$\frac{dT_w}{dz} = \frac{m_a}{m_w} \left(\frac{1}{c_{pw}} \frac{di_{ma}}{dz} - T_w \frac{dw}{dz} \right) \quad (3.5)$$

The total enthalpy transfer at the air-water interface consists of an enthalpy transfer associated with the mass transfer due to the difference in vapor concentration and the heat transfer due to the difference in temperature.

Accordingly, one has

$$dQ = dQ_m + dQ_c \tag{3.6}$$

Where the subscripts $_m$ and $_c$ refer to the enthalpies associated with mass transfer and convective heat transfer. The mass transfer at the interface is expressed by

$$\frac{dm_w}{dz} dz = h_d (w_{sw} - w) dA \tag{3.7}$$

Where w_{sw} is the saturation humidity ratio of air evaluated at the local bulk water temperature, T_w.

$$dQ_m = i_v \frac{dm_w}{dz} dz = i_v h_d (w_{sw} - w) dA \tag{3.8}$$

The enthalpy of the water vapor, i_v, at the bulk water temperature, T_w, is given by

$$i_v = i_{fgwo} + c_{pv} T_w$$

Where T_w is in °C and c_{pv} is evaluated at $T_w/2$ °C.

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

$$dQ_c = h(T_w - T_a) dA \tag{3.9}$$

The enthalpy of the saturated air evaluated at the local bulk water temperature is given by

$$i_{masw} = c_{pa} T_w + w_{sw} (i_{fgwo} + c_{pv} T_w) = c_{pa} T_w + w_{sw} i_v$$

This may be rewritten as

$$i_{masw} = c_{pa}\, T_w + w\, i_{v} + (w_{sw} - w)\, i_v \tag{3.10}$$

Where c_{pa} is evaluation at $T_w/2$ °C.

Subtract Equation 3.4 from Equation 3.10. The resultant equation can be simplified if the small differences in the specific heats, which are evaluated at different temperatures, are ignored, i.e.

$$i_{masw} - i_{ma} = (c_{pa} + w\, c_{pw})\,(T_w - T_a) + (w_{sw} - w)\, i_v$$

Or

$$T_w - T_a = [(i_{masw} - i_{ma}) - (w_{sw} - w) i_v]/c_{pma} \tag{3.11}$$

Where $c_{pma} = c_{pa} + w\, c_{pv'}$

Substitute Equations 3.8, 3.9, and 3.11 into Equation 3.6 to find upon re-arrangement

$$dQ = h_d \left[\frac{h}{c_{pma} h_d} (i_{masw} - i_{ma}) + \left(1 - \frac{h}{c_{pma} h_d} \right) i_v (w_{sw} - w) \right] dA \tag{3.12}$$

Where $h/\,(c_{pma} h_d) = Le_{f'}$ which is known as the Lewis factor and is indication of the relative rates of heat and mass transfer in an evaporate process.

The following equation to express the Lewis factor for air-water vapor systems:

$$L_{ef} = 0.866^{0.667} \left(\frac{w_{sw} + 0.622}{w + 0.622} - 1 \right) / \ln \left(\frac{w_{sw} + 0.622}{w + 0.622} \right) \tag{3.13}$$

Nothing the enthalpy transfer must be equal to the enthalpy change of the airstreams, one has from Equation 3.12

$$\frac{di_{ma}}{dz} = \frac{1}{m_a}\frac{dQ}{dz} = \frac{h_d}{m_a}\frac{dA}{dz}\ [\, L_{ef}\,(i_{masw}-i_{ma}) + (1-L_{ef})\, i_v\,(w_{sw}-w)] \tag{3.14}$$

For a one-dimensional model of the cooling tower fill where the available area for heat and mass transfer is the same at any horizontal section through the fill, the transfer area for a section dz deep is usually expresses as

$$dA = a_{fi}\, A_{fr}\, dz \tag{3.15}$$

Where

a_{fi} = the wetted area divided by the volume of the fill or area density

A_{fi} = the frontal area or face area

Substitute Equation 3.15 into Equation 3.14, and find

$$\frac{di_{ma}}{dz} = \frac{h_d\, a_{fi}\, A_{fi}}{m_a}\ [Le_f\,(i_{masw}-i_{ma}) + (1-Le_f)\, i_v\,(w_{sw}-w)] \tag{3.16}$$

When the ambient humidity is high enough, the air becomes saturated with water vapor prior to its exit from the fill. In this case, the previous equations fail to describe the evaporative process in the fill. Since the temperature of the saturated air the interface is still higher than the temperature of the now saturated free stream air, a potential for heat and mass transfer will still exist. The excess water vapor transferred to the free stream air will condense as a mist.

Assume that the heat and mass transfer coefficients for the mist zone is the same as those for unsaturated air as is proposed by Poppe and Rogener [4]. The evaporation rate in the mist zone depends on the difference in moisture content of the saturated air at the interface, at the local bulk water temperature, and the moisture content of the free stream air, thus

$$\frac{dm_w}{dz} = h_d a_{fi} A_{fr} \left[w_{sw} - w_{sa} \right] \tag{3.17}$$

Where w_{sa} is the humidity ratio of saturated air at temperature T_a.

Since the excess water vapor will condense, the enthalpy of supersaturated air is expressed by

$$i_{ss} = c_{pa} T_a + w_{sa} (i_{fgwo} + c_{pv} T_a) + (w - w_{sa}) c_{pw} T_a \tag{3.18}$$

Proceeding along the same lines as in the case of unsaturated air, find

$$\frac{di_{ma}}{dz} = \frac{h_d a_{fi} A_{fr}}{m_a} \left[L_{ef} (i_{masw} - i_{ss}) + (1 - L_{ef}) i_v (w_{sw} - w_{sa}) + L_{ef} (w - w_{sa}) c_{pw} T_w \right] \tag{3.19}$$

In addition to the assumption stated earlier, Merkel assumes the Lewis factor is equal to unity and the evaporation loss is negligible. Introducing these two assumptions, the governing Equations 3.16 and 3.5 simplify to

$$\frac{di_{ma}}{dz} = \frac{h_d a f_i A_{fr}}{m_a} (i_{masw} - i_{ma}) \tag{3.20}$$

and

$$\frac{dT_w}{dz} = \frac{m_a}{m_w} \frac{1}{c_{pw}} \frac{di_{ma}}{dz} \tag{3.21}$$

With only the previous equations, it is impossible to calculate the state of the air leaving the fill, since at least two properties must be known in order to achieve this. Hence, the exit air temperature, essential to calculating the airflow rate through a natural draft tower, is unknown. Merkel assumes that the air leaving the fill is saturated with water vapor, which enables him to determine the temperature and density of the air and the draft. In many practical cases, this assumption will yield reasonable results.

Traditionally, Equations 3.20 and 3.21 are combined to yield upon integration,

$$\frac{h_d A}{M_w} = \frac{h_d a_{fi} A_{fr} L_f i}{M_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{Two}^{Twi} \frac{c_{pw} dT_w}{(i_{masw} - i_{ma})} = Me \tag{3.22}$$

Commonly referred to as Merkel's equation. The non dimensional coefficient of performance or transfer characteristic $h_d a_{fi} L_{fi}/G_w$, is known as the Merkel number. In this equation, L_{fi} is the height of the fill or the air travel distance (ATD) and $G_w = m_w/A_{fr}$.

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

Poppe and Rogener [4] do not make the simplifying assumptions of Merkel. Their more rigorous approach is as follows:

Substitute Equations 3.7 and 3.14 into Equation 3.3, and find with $c_{pw}dT_w = di_w$

$$\begin{aligned} m_w c_{pw} dT_w &= m_w\, di_w \\ &= h_d\, dA\, [L_{ef}\,(i_{masw} - i_{ma}) + (1 - L_{ef})\, i_v\,(w_{sw} - w) - c_{pw} T_w\,(w_{sw} - w)] \end{aligned} \tag{3.23}$$

Equation 3.5 can be rearranged to give

$$\frac{d_w}{dT_w} = \frac{di_{ma}}{T_w c_{pw} dT_w} - \frac{m_w}{m_a T_w} = \frac{di_{ma}}{T_w di_w} - \frac{m_w}{m_a T_w} \tag{3.24}$$

Substitute Equations 3.14 and 3.23 into Equation 3.24, and find upon rearrangement

$$\frac{dw}{dTw} = \frac{c_{pw} m_w (w_{sw} - w)/m_a}{(i_{masw} - i_{ma}) + (L_{ef} - 1)[(i_{masw} - i_{ma}) - (w_{sw} - w)i_v] - (w_{sw} - w)c_{pw}T_w} \tag{3.25}$$

Upon substitution of Equation 3.1.25 into Equation 3.1.24, find

$$\frac{di_{ma}}{dt_w} = c_{pw}\frac{m_w}{m_a}$$

$$x\left[1 + \frac{c_{pw}T_w(w_{sw} - w)}{(i_{masw} - i_{ma}) + (L_{ef} - 1)[(i_{masw} - i_{ma}) - (w_{sw} - w)i_v] - (w_{sw} - w)c_{pw}T_w}\right] \tag{3.26}$$

Combine Equation 3.1 and 3.7 to find

$$h_d \; dA = \frac{m_a d_w}{(w_{sw} - w)} \tag{3.27}$$

Divide both sides of Equation 3.27 by m_w, introduce dT_w/dT_w to the right side of Equation 3.27 and integrate to find the Merkel number according to Poppe, i.e.,

$$\frac{h_d A}{m_w} = \int\left(\frac{m_a}{m_w}\right)\left(\frac{d_w/dT}{w_{sw} - w}\right)dT_w = Me_p \tag{3.28}$$

Upon substitution of Equation 3.25 into Equation 3.28 and differentiation of the latter with respect to water temperature, find

$$\frac{dme_p}{dT_w} = \frac{c_{pw}}{(i_{masw} - i_{ma}) + (L_{ef} - 1)[(i_{masw} - i_{ma}) - (w_{sw} - w)i_v] - (w_{sw} - w)c_{pw}T_w} \tag{3.29}$$

To evaluate the change in the ratio of m_w/m_a as the air flows upward through the fill, consider the elementary control volume in the fill (Fig. 3.2).

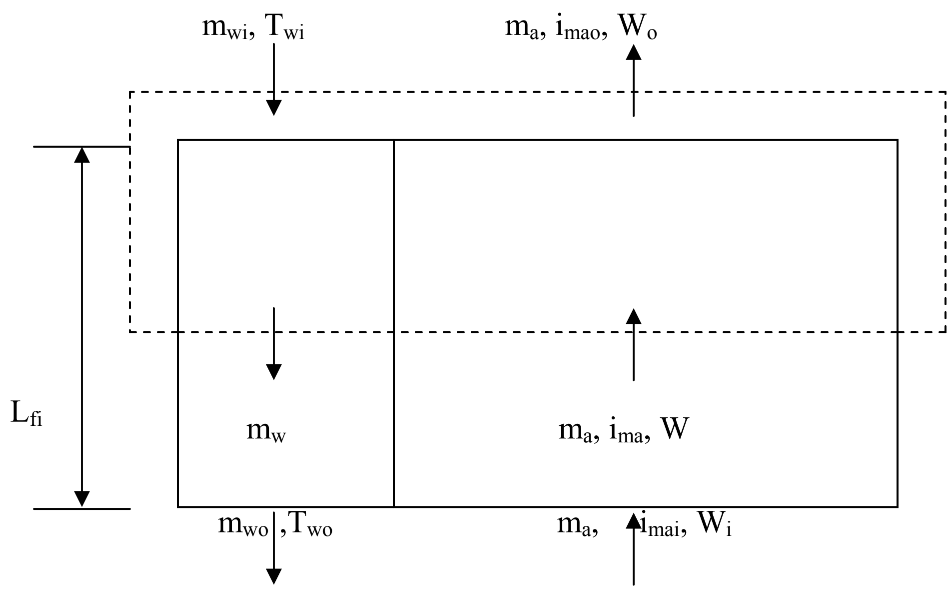


Fig. 3.2 Control Volume in Fill

A mass balance in terms of the inlet water mass flow rate, m_{wi}, applicable to the control volume is

$$m_{wi} = m_w + m_a (w_o - w) \tag{3.30}$$

Upon re-arrangement of Equation 3.30,

$$\frac{m_w}{m_a} = \frac{m_{wi}}{m_a}\left[1 - \frac{m_a}{m_{wi}}(w_o - w)\right] \tag{3.31}$$

The air outlet conditions can now be obtained from Equation 3.13, 3.25, 3.26, and 3.31 in terms of enthalpy and humidity ratio.

The previous equations are applicable if the air is unsaturated. For supersaturated air, the enthalpy i_{ss}, as given by Equation 3.18, is employed in the calculations instead of i_{ma}.

Equation 3.7 is not valid in the case of supersaturated air. The mass transfer at the interface of the water and supersaturated air is given by

$$\frac{dm_w}{dz} = dz = h_d\left(w_{sw} - w_{sa}\right)dA \tag{3.32}$$

The Bosnjakovic [7] equation for the Lewis factor for supersaturated air is expressed as

$$Lef = 0.8660^{0.667}\left(\frac{w_{sw}+0.622}{w_{sa}+0.622}-1\right)/\ell n\left(\frac{w_{sw}+0.622}{w_{sa}+0.622}\right) \tag{3.33}$$

By following the same procedures as for unsaturated as for unsaturated air but employing Equations 3.18, 3.32, and 3.33, find for supersaturated air

$$dw/\,dT_w = c_{pw}m_w/\,m_a\,(w_{sw} - w_{sa}) \\ /\,[(i_{masw}-i_{ss}) + (L_{ef} - 1)\,\{(i_{masw} - i_{ss}) - (w_{sw} - w_{sa})\,i_v + (w - w_{sa})\,c_{pw}\,T_w\} \\ + (w - w_{sw})\,c_{pw}T_w] \tag{3.34}$$

The enthalpy gradient is given by

$$di_{ma}/\,dT_w = (c_{pw}m_w/m_a)\,[1 + c_{pw}T_w\,(w_{sw} - w_{sa}) \\ /\,\{(i_{masw} - i_{ss}) + (L_{ef} - 1)\,\{(i_{masw} - i_{ss}) - (w_{sw} - w_{sa})\,i_v + (w - w_{sa})\,c_{pw}T_w\} \\ + (w - w_{sw})\,c_{pw}T_w\}] \tag{3.35}$$

While

$$dMe_p/dT_w = c_{pw}\,/[\left(i_{masw} - i_{ss}\right) + \left(L_{ef} - 1\right)\{\left(i_{masw} - i_{ss}\right) - \left(w_{sw} - w_{sa}\right)i_v + \left(w - w_{sa}\right)c_{pw}Tw\} \\ + (w - w_{sw})c_{pw}T_w] \tag{3.36}$$

The outlet air conditions in terms of enthalpy and humidity ratio can now be determined using Equations 3.31, 3.33, 3.34, and 3.35.

3.2 Merkel Method

Merkel developed the theory for the thermal evaluation of cooling towers in 1925.The basic theory of cooling tower operation was first proposed by walker et al, who developed the basic equations for total mass and energy transfer and considered each process separately.

The Merkel theory relies on several critical assumptions to reduce the solution to a simple hand calculation .Because of these assumptions; however, the Merkel method does not accurately represent the physics of heat and mass process in the cooling tower fill.

The critical simplifying assumptions of the Merkel theory are:-

(a). The Lewis factor, Le_f, relating heat and mass transfer is equal to 1.

(b). The air exiting the tower is saturated with water vapor and it is characterized only by its enthalpy.

(c). The reduction of water flow rate by evaporation is neglected in the energy balance. To simplify the analysis of an evaporative cooling process Merkel assumed that the evaporative loss is negligible, i.e. dw = 0

From Eq. (3.23) and that the Lewis number is equal to one. Eqs. (3.23) and (3.26) of the counter flow evaporative process simplify respectively to

$$di_{ma}/dz = (h_d.a_{fi}.A_{fi} / m_a) (i_{masw} - i_{ma}) \qquad (3.37)$$

And by dividing Eq. (3.37) by dz on both sides of Eq. (3.37) to

$$dT_w/dz = (m_a/ m_w C_{pw}) (di_{ma} /dz) \qquad (3.38)$$

Eqs (3.37) and (3.38) describe respectively the change in the enthalpy of the air water vapor mixture and the change in water temperature as the air travel distance changes. Eqs (3.37) and (3.38) can be combining to yield upon integration the Merkel equation

$$\frac{h_d A}{M_w} = \frac{h_d a_{fi} A_{fr} L_f i}{M_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{Two}^{Twi} \frac{c_{pw} dT_w}{(i_{masw} - i_{ma})} = Me \tag{3.39}$$

Where Me_M is the Merkel number (Merkel method)

3.3 Effectiveness-NTU Method

Jaber and Webb [3] developed the equations necessary to apply the Effectiveness-NTU method to counter flow or cross flow cooling towers. The approach is useful in cross flow and simplifies the method of solution when compared to a more conventional numerical procedure.

$$dQ = h_d \left[Le_f \left(i_{masw} - i_{ma} \right) + \left(1 - Le_f \right) i_v \left(w_{sw} - w \right) \right] dA \tag{3.40}$$

With the assumption of Merkel that the Lewis factor is equal to unity, Equation 3.40 reduces to

$$dQ = h_d \, (i_{masw} - i_{ma}) \, dA \tag{3.41}$$

Where $(i_{masw} - i_{ma})$ is the enthalpy driving potential used by the effectiveness-NTU method in the case of evaporative cooling.

For the control volume shown in Figure 3.40, it follows from Equations 3.14 and 3.21 that

$$dQ = m_w c_{pw} dT_w = m_a di_{ma} \tag{3.42}$$

It is convenient to relate dQ to the slope of the saturated air enthalpy (i_{masw}) water temperature (T_w) curve. Equation 3.42 is written as

$$dQ = m_w c_{pw} di_{masw}/(di_{masw}/dT_w) = m_a di_{ma} \tag{3.43}$$

From which if follows that

$$di_{masw} = dQ\ (di_{masw}/dT_w)\ /\ (m_w c_{pw}) \tag{3.44}$$

It follows from Equation 3.42 that $di_{ma} = dQ/m_a$. Subtract this relation from Equation 3.44, and find

$$di_{masw} - di_{ma} = d\left(i_{masw} - i_{ma}\right) = dQ\left[\left(di_{masw}/d\,T_w\right)/\left(m_w c_{pw}\right) - 1/m_a\right] \tag{3.45}$$

From Equations 3.45 and 3.41, it follows that

$$\frac{d\left(i_{masw} - i_{ma}\right)}{i_{masw} - i_{ma}} = h_d\left(\frac{di_{masw}/dT_w}{m_w c_{pw}} - \frac{1}{m_a}\right)dA \tag{3.46}$$

This equation, applicable in an evaporative system, will correspond to the heat exchanger design .if one defines the air capacity rate (cold fluid) as m_a and the water capacity rate (hot fluid) as $m_w c_{pw}/\ (di_{masw}/dT_w)$.

The maximum theoretical amount of enthalpy that can be transferred, is Q_{max}= (minimum capacity rate) x (i_{maswi} - i_{mai}), where i_{maswi} is the saturated air enthalpy at the water inlet condition and i_{mai} denotes air inlet enthalpy.

There are two possible cases to be considered.

Case 1

$$m_w\, c_{pw} / (di_{masw} / dT_w) < m_a$$

Where consistent with heat exchanger design terminology $C_{min} = m_w c_{pw} / (di_{masw}/dT_w)$ and $C_{min} = m_a$. The evaporative capacity rate ratio for this particular case is given by

$$C_R = C_{min} / C_{max} = m_w cpw / \left[\left(di_{masw} / dT_w\right) ma\right]$$

Substitute C_R into Equation 3.46 to find

$$\frac{d\left(i_{masw} - i_{ma}\right)}{i_{masw} - i_{ma}} = \frac{h_d\left(di_{masw} / dT_w\right)\left(1 - C_e\right) dA}{m_w c_{pw}} \tag{3.47}$$

Integration of Equation 3.47 between the entering and leaving air states, i_{ai} and i_{ao}, gives

$$(i_{maswo} - i_{mai}) / (i_{maswi} - i_{mao}) = \exp\left[-NTU\left(1 - C_R\right)\right] \tag{3.48}$$

where i_{maswo} and i_{maswi} refer to the saturated air enthalpy at the water outlet and inlet conditions. The analogous definition for NTU in this particular evaporative system or wet-cooling tower is

$$NTU = h_d\, A\, (di_{masw} / dT_w) / (m_w c_{pw}) \tag{3.49}$$

Where

A is the total wetted transfer area.

The heat exchange effectiveness is defined as

$$e = Q / Q_{max} \tag{3.50}$$

Integration of Equation 3.42 between inlet and outlet conditions gives

$$Q = m_w c_{pw} (T_{wi} - T_{wo}) = m_a (i_{mao} - i_{mai}) \tag{3.51}$$

The maximum enthalpy transfer rate can be expressed approximately ad

$$Q_{max} \approx m_w c_{pw} (i_{maswi} - i_{mai}) / (di_{masw} / dT_w) = m_a C_R (i_{maswi} - i_{mai}) \tag{3.52}$$

Where the gradient of the saturated air enthalpy-temperature curve over the control volume is

$$\frac{di_{masw}}{dT_w} \approx \frac{i_{maswi} - i_{maswo}}{T_{wi} - T_{wo}} \tag{3.53}$$

It follows from Equations 3.50, 3.51, and 3.53 that

$$e = (i_{maswi} - i_{maswo}) / (i_{maswi} - i_{mai}) \tag{3.54}$$

And from Equations 3.50, 3.51, and that 3.52

$$C_R .e = (i_{mao} - i_{mai}) / (i_{maswi} - i_{mai}) \tag{3.55}$$

From Equation 3.54 and 3.55 , it follows that

$$(e - 1) / (e.C_R - 1) = (i_{maswo} - i_{mai}) / (i_{maswi} - i_{mao}) \tag{3.56}$$

Equating equations 3.48 and (3.56) gives the Effectiveness-NTU equation for a counter flow evaporative system or cooling tower

$$e = \frac{1 - \exp[-NTU\, e(1 - C_R)]}{1 - C_R \exp[-NTU\, e(1 - C_R)]} \tag{3.57}$$

Case 2

$$m_a < m_w c_{pw} / (di_{masw} / dT_w)$$

In this case,

$$C_R = m_a (di_{masw} / dT_w) / (m_w c_{pw})$$

Follow a procedure similar to that of Case 1, and again find the effectiveness given by Equation 3.57 .

The Effectiveness-NTU method is subject to approximations involved in linearizing the i_{masw} versus T_w curve as a straight line. The accuracy of the method can be increased by breaking up the design into a number of increments.

An analytical method was developed by Berman to improve the approximation of the i_{masw} verses T_w curve as a straight line. He proposed a correction factor δ or λ given by

$$\delta = \lambda = (i_{maswo} + i_{maswi} - 2\ i_{masw}) /4 \tag{3.58}$$

Where i_{masw} denotes the enthalpy of the saturated air at the mean water temperature $T_{wm} = (T_{wi} + T_{wo}) /2$.
This factor is used to obtain a more correct value for Q_{max} as follows:

$$Q_{max} = C_{min} (i_{maswi} - \delta - i_{mai}) \tag{3.59}$$

The use of the correction factor gives a two-increment design.
The Effectiveness-NTU equations described previously show the Effectiveness-NTU equations for a counter flow heat exchanger can be applied to a counter flow cooling tower. The equations for a cross flow heat exchanger can be applied to a cross flow cooling tower. Jabber and Webb [3] recommend the use of the unmixed/unmixed Effectiveness-NTU equation.

Chapter 4

Simulation and Solution Techniques

This chapter represents the solution techniques applied for the Merkel, e-NTU and Poppe methods. To validate the accuracy of the output, the results often are compared with the examples available in the literature. Computer programs are developed in C++ language. There are two types of programs associated with cooling towers i.e., Sizing and Rating. In sizing problem Merkel number and fill volume is calculated, while in performance analysis problem outlet water temperature is calculated.

4.1 Merkel Method

The details of Merkel method is described in chapter 3. The main equation used in this method is given below:

$$\frac{h_d A}{m_w} = \frac{h_d a_{fi} A_{fr} L_f i}{m_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{Two}^{Twi} \frac{c_{pw} dT_w}{(i_{masw} - i_{ma})} = Me \tag{4.1}$$

$$m_w.C_{pwm}.dT_w = m_a.di_{ma} \tag{4.2}$$

The non dimensional coefficient of performance or transfer characteristic $h_d a_{fi} L_{fi}/G_w$, is known as the Merkel number. In this equation, L_{fi} is the height of the fill or the air travel distance (ATD) and $G_w = m_w/A_{fr}$.

4.1.1 Solution Procedure Applied

The integral in Eq. (4.1) needs to be evaluated by numerical integration techniques. The J.C. Kroger and D.G. Klopper [6] recommended that the four point Chebyshev integration techniques be employed. A discussion of the Chebyshev integration technique can also be found in Mohiuddin and Kant [6]. LI and Piddy [2] states that the Chebyshev procedure lacks accuracy when the approach (i.e. the difference between the water outlet temperature and the air wet bulb temperature) is small (down to 0.56 ºC). Any integration technique can be employed to solve Eq. (4.1) but it is strongly recommended that same integrations technique be employed in the fill performance analysis and the subsequently cooling tower performance analysis.

The four point Chebyshev integration techniques essentially use four intervals for determination of the integral. Li and Priddy [2] use thirteen and seven intervals respectively for numerical integration to determine the change of water and air enthalpy through the fill for a cooling range of approximately 14 ºC. It was found by the authors that the Chebyshev procedure is generally very accurate when compared to the composite Simpson rule with 100 intervals.

A computer programs “Merkel Size “was developed to determine the size of the cooling tower. In the above mentioned program the cooling range (Integration limits in equation 4.1) are divided in 100intervals to get reasonable accuracy. Input/ output parameters are represented in Fig 4.1 and Flow chart of the program “Merkel Size” is shown in Fig. 4.2. A program “Merkel Rate” was developed to determine the outlet water temperature of cooling tower. Input/output diagram & the flow chart are shown in Fig. 4.3 and Fig. 4.4 respectively.

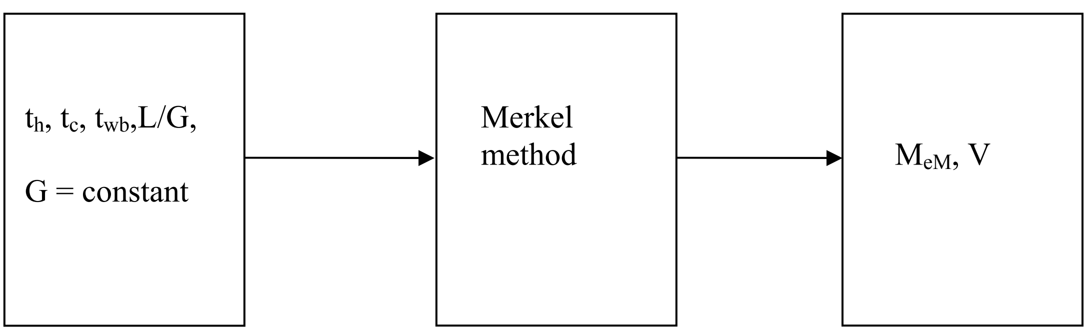


Fig. 4.1 Input/output of program “Merkel Size” (For design problem)

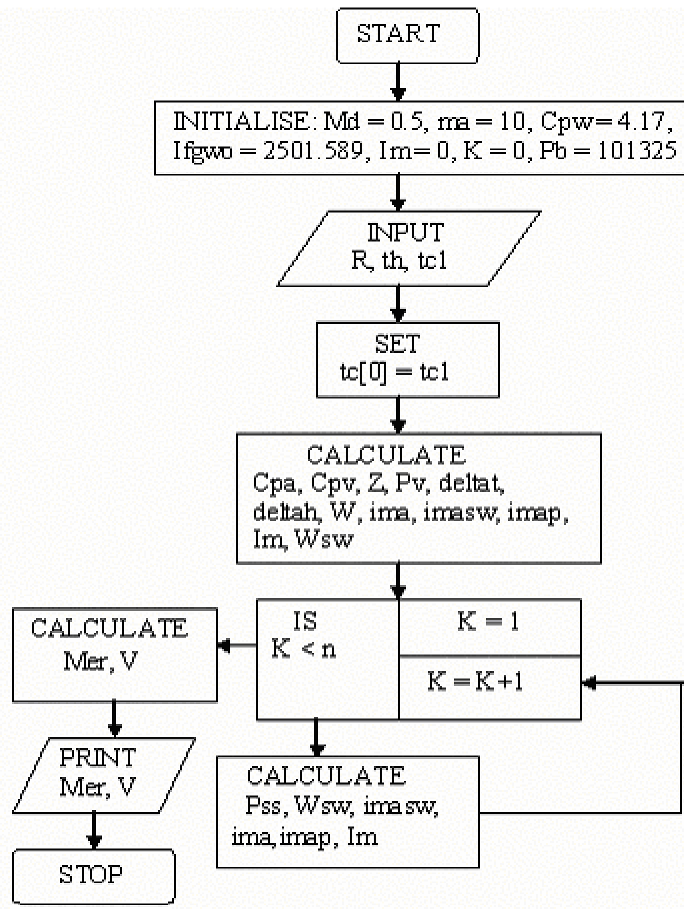


Fig 4.2 Flow Chart Of Merkel Method Computer Program For Design Problem Of Cooling Tower.

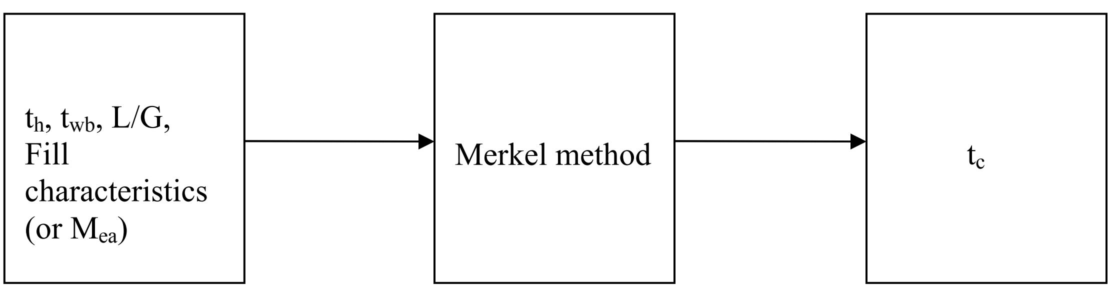
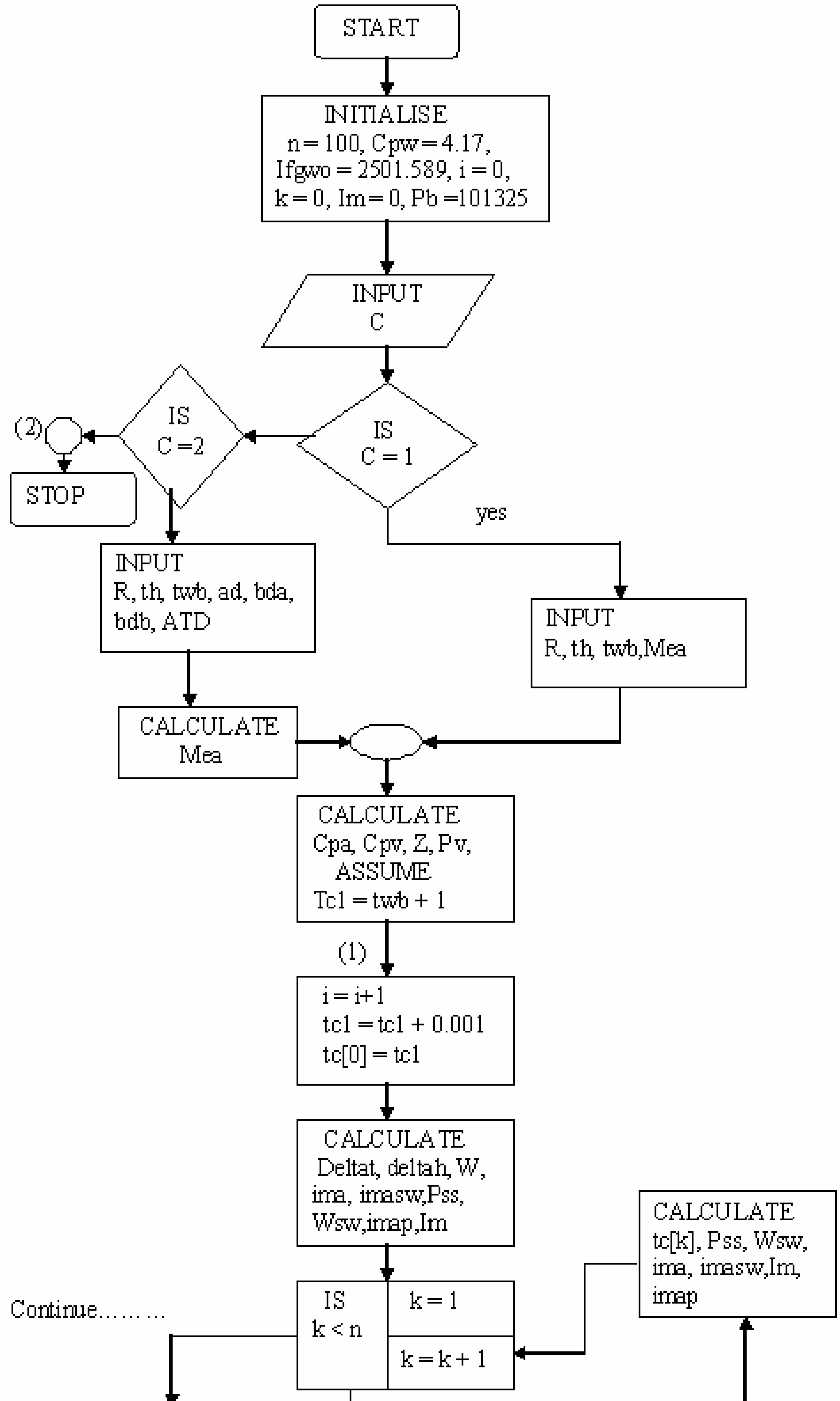


Figure 4.3 Input/output of program "Merkel Performance" (For performance analysis problem)


START
INITIALISE
n = 100, Cpw = 4.17,
Ifgwo = 2501.589, i = 0,
k = 0, Im = 0, Pb =101325
INPUT
C
IS
C = 1
IS
C =2
(2)
STOP
yes
INPUT
R, th, twb, ad, bda,
bdb, ATD
INPUT
R, th, twb,Mea
CALCULATE
Mea
CALCULATE
Cpa, Cpv, Z, Pv,
ASSUME
Tc1 = twb + 1
(1)
i = i+1
tc1 = tc1 + 0.001
tc[0] = tc1
CALCULATE
Deltat, deltah, W,
ima, imasw,Pss,
Wsw,imap,Im
CALCULATE
tc[k], Pss, Wsw,
ima, imasw,Im,
imap
IS
k < n
k = 1
k = k + 1
Continue………

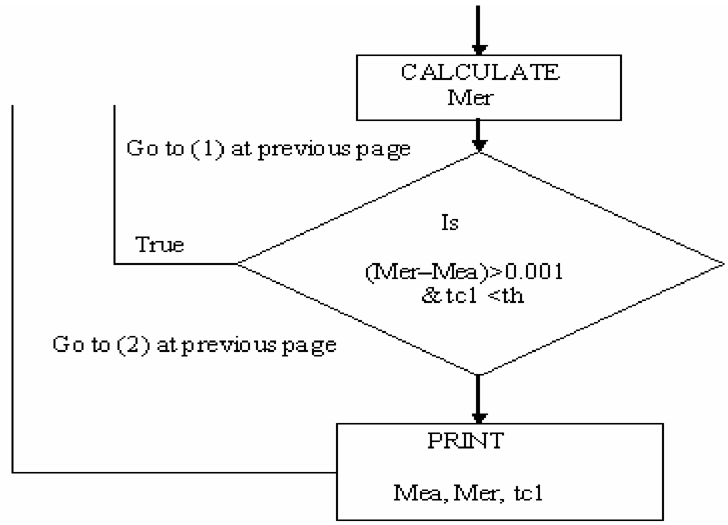


Fig.4.4 Flow Chart Of Computer Program Of Merkel Method For Analysis Problem Of Cooling Tower

4.1.2 Validation of Numerical Modeling

For validation of numerical model Merkel method for design problem of cooling tower and performance analysis of cooling tower, results are compared respectively with data taken from Li and Priddy's handbook [2] and from Goel thesis 1986 [17].

Table 4.1 Data from Li and Priddy's Hand Book [2] of a sizing problem.

Initial Conditions	Initial Values
Inlet hot water temperature (t_h)	43.33 ºC
L/G ratio	1.3
Wet bulb temperature (t_{wb})	20.55 ºC
Outlet cold water temperature (tc)	28.88 ºC
Air mass flow rate (G)	10 gm/sec

Table 4.2 Results compared with Li and Priddy's Hand Book [2] for sizing problem.

	Merkel number (M_{eM})	Volume of cooling tower, m^3 (V_m)
Li and Priddy's handbook (10 Intervals)	1.51	30.20
Numerical model of Merkel method (10 Intervals)	1.57	31.5
Numerical model of Merkel method (100 Intervals)	1.44	28.88

Percentage error in Merkel number for 10 and 100 intervals is found 3.9 and 4.6 when numerical model compared with Li and Priddy's handbook output results. While percentage error of fill volume is calculated 3.3 % and 4.3 % for respectively 10 and 100 intervals. Results in Table 4.2 show that Merkel number and fill volume calculated by numerical model decreases when number of interval increases.

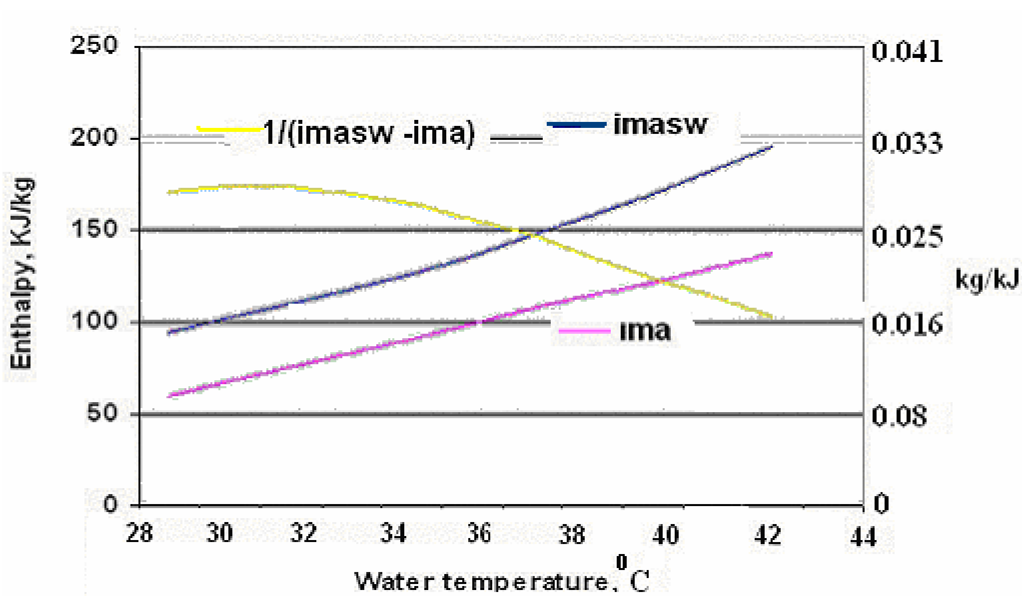


Fig.4.5 Enthalpy diagram of the Merkel method

Figure 4.5 shows the enthalpy curves of the air in a counter flow wet cooling tower. This Fig.4.5 plotted on the basis of results calculated by Merkel method for the Kelly's hand book data input parameters which has been shown in Appendix (B) Table B.2. The i_{ma} curve i.e. the enthalpy of the air as it moves through the fill, shown in Figure 4.5 is linear due to the linear nature of Eq. (A.1). The i_{masw} curve is the saturation curve of the air at the water interface temperature. The potential for heat and mass transfer at a particular water temperature is the difference between i_{masw} and i_{ma}.the Merkel number, M_{eM}, of Eq. (4.1.1), is a function of the area under the $1/(\ i_{masw} - i_{ma}\ )$ as shown in Figure 4.3

Table 4.3 Data from Goel thesis [17] (For performance analysis problem).

Initial Conditions	Initial Values
Inlet hot water temperature (t_h)	44.5 ºC
L/G ratio	1.55
Wet bulb temperature (t_{wb})	30 ºC
Merkel number available (M_{ea})	1.68

Table 4.4 Result compared with Goel thesis [17] (For performance analysis problem).

	Prediction of outlet cold water temperature (tc)
Goel thesis	33.85 ºC
Numerical model of Merkel method.	33.45 ºC

Percentage error calculated for outlet water temperature is 1.1 % when numerical model compares with Goel thesis [17] results. It is clear from results shown in Table 4.2. and Table 4.4 that numerical model for Merkel method predicts results are quite close to the results shown by Li and Priddy's handbook [2] and Goel thesis [17] for both design and performance analysis problem of cooling tower.

4.2 Poppe Method

The main equations used in this method are 3.25, 3.26 and 3.29 for unsaturated air and 3.34, 3.35 and 3.36 for saturated air flow in cooling tower.

4.2.1 Solution Procedure Applied

The forth order Runga – Kutta method [6] is employed to solve the system of differential equations for unsaturated and supersaturated air. The system of equations for unsaturated air (including saturated air) is represented by Eqs. (3.25), (3.26) and (3.29). The system of equations for the supersaturated air is represented by Eqs. (3.1.34), (3.35) and (3.36). In the equations that follow, i_{ma} must be replaced by i_{ss} for supersaturated air. The first step in the solution process is to divide the fill into a number of intervals where the water temperature difference is equal across each interval, i.e.

$\Delta T_w = (T_{wi} - T_{wo})/$ (Number of intervals)

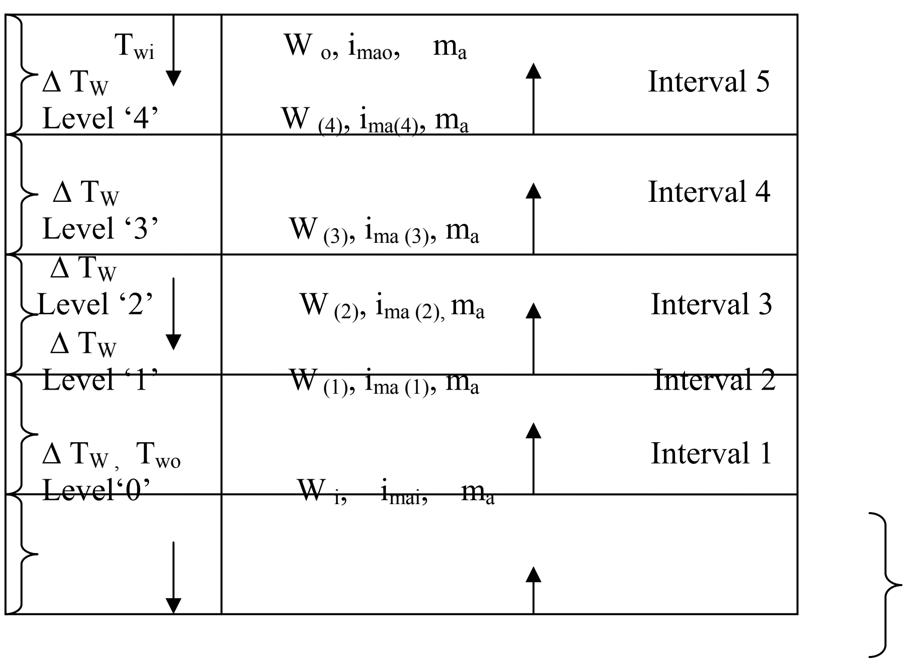


Figure 4.6 Counter flow fill divided into five intervals.

Fig. 4.6 shows an example where the fill is divided into five intervals. It is necessary to divide the fill into more than one interval to capture, as accurately as possible, the point at which the air becomes supersaturated. This is because a different set of equations is applicable for supersaturated air. Approximately five intervals are generally sufficient to obtain accurate results. It was mentioned that the value of w_o is not known a priori. A value of m_{wo} is guessed and a new value of w_o is subsequently determined the equations are solved until the value of w_o converges. Only a few of these iterations are generally necessary to obtain convergence.

The equations are solved across one interval at a time by Runga-Kutta method, which is explained below. The air, which is generally unsaturated, enters the fill at Level (0) in Fig. 4.6 with w_i, i_{mai}, m_a known. The values of $w_{(1)}$ and $i_{ma\ (1)}$ are then determined by the Runga-Kutta method with the set of equations for unsaturated air, ma remains constant. It is then determined whether the air is still unsaturated or if it is supersaturated at the outlet of the first interval (i.e., at Level (1) in Fig 4.6).if the air is supersaturated, the set of equations for the supersaturated air must be solved across the next interval. If the air is the supersaturated it will generally remain in the supersaturated state through the rest of the fill.

The following procedure can be followed to determine whether the air at outlet of an interval, as indicated in Fig.4.6, is unsaturated or supersaturated. Assume that the air at Level (1), for example, is unsaturated and determine $T_{a\ (1)}$ from Eq.(A.1) in appendix (A) by iterative means with $w_{(1)}$ and $i_{ma(1)}$ known. Then assume that the air is saturated and determine the wet bulb temperature, $T_{wb\ (1)}$ from Eq. (A.6). $T = T_{wb}$ in Eq. (A.6). If $T_{a\ (1)} > T_{wb\ (1)}$ then the assumption that the air is unsaturated is correct. If $T_{wb\ (1)}) > T_{a\ (1),}$ which is impossible, the air is supersaturated. The actual value of the wet bulb temperature is then $T_{wb\ (1)} = T_{a\ (1).}$

Eqs. (3.25), (3.26) and (3.29) for unsaturated and saturated air or Eqs(3.34), (3.35) and (3.36).

For supersaturated air can be respectively written as

$$dw/dT_w = f\,(w, i_{ma,}\,T_w\,) \tag{4.1}$$

$$di_{ma}/dT_w = g\,(w, i_{ma}, T_w\,) \tag{4.2}$$

$$dM_{ep}/dT_w = h\,(w, i_{ma}, T_w) \tag{4.3}$$

Refer to Fig.4.6. The cooling tower fill is divided into one or more intervals with the same water temperature difference across each interval. In addition to the intervals, levels are specified (A level is an imaginary horizontal plane through the fill at top and bottom of the fill and between two fill intervals). Initial values of the variables w, i_{ma} and T_w are required on a particular level, say level (n). The values of the variables can then be determined at the level (n+1) with the aid of Eqs. (4.4) and (4.5).

$$w_{(n+1)} = w_{(n)} + (\,j_{(n+1,\,1)}) + 2j_{(n+1,\,2)} + 2j_{(n+1,\,3)+}j_{(n+1,\,4))/}6 \tag{4.4}$$

$$i_{ma\,(n+1)} = i_{ma(n)} + (\,k_{(n+1,\,1)} + 2k_{(n+1,\,2)} + 2k_{(n+1,\,3)} + k_{(n+1,\,4))/}6 \tag{4.5}$$

$$M_{ep\,(n+1)} = M_{ep(n)} + (\,l_{(n+1,\,1)} + 2l_{(n+1,\,2)} + 2l_{(n+1,\,3)} + k_{(n+1,\,4))/}6 \tag{4.6}$$

Where

$$j_{\,(n+1,\,1)} = \Delta T_w\,.\,f\,(\,T_{w(n),}\,i_{ma(n)}\,,\,w_{(n)}\,) \tag{4.7}$$

$$k_{\,(n+1,\,1)} = \Delta T_w\,.\,g\,(\,T_{w(n),}\,i_{ma(n),}\,w_{(n)}\,) \tag{4.8}$$

$$l_{\,(n+1,1)} = \Delta T_w\,.\,h\,(\,T_{w(n),}\,i_{ma(n),}\,w_{(n)}\,) \tag{4.9}$$

$$j_{\,(n+1,2)} = \Delta T_w\,.\,f(\,T_{w(n)} + \Delta T_{w/2},\,i_{ma(n)} + k_{(n+1,1)/2},\,w_{(n)} + j_{(n+1,1)/2}\,) \tag{4.10}$$

$$l_{\,(n+1,2)} = \Delta T_w\,.\,g(\,T_{w(n)} + \Delta T_{w/2},\,i_{ma(n)} + k_{(n+1,\,1)/2},\,w_{(n)} + j_{(n+1,1)/2}\,) \tag{4.11}$$

$$j_{(n+1,3)} = \Delta T_w\,.f(\,T_{w(n)} + \Delta T_{w/2},\,i_{ma(n)} + k_{(n+1,\,2)/2},\,w_{(n)} + j_{(n+1,2)/2}\,) \tag{4.12}$$

$$k_{(n+1,3)} = \Delta T_w \,.\, g(T_{w(n)} + \Delta T_{w/2}, i_{ma(n)}) + k_{(n+1,2)/2}, w_{(n)} + j_{(n+1,2)/2}) \tag{4.13}$$

$$l_{(n+1,3)} = \Delta T_w \,.\, h(T_{w(n)} + \Delta T_{w/2,} ima_{(n)} + k_{(n+1,2)/2}, w_{(n)} + j_{(n+1,2)/2}) \tag{4.14}$$

$$j_{(n+1,4)} = \Delta T_w \,.\, f(T_{w(n)} + \Delta T_w, i_{ma(n)} + k_{(n+1,3),} w_{(n)} + j_{(n+1,3)})) \tag{4.15}$$

$$k_{(n+1,4)} = \Delta T_w \,.\, g(T_{w(n)} + \Delta T_w, i_{ma(n)} + k_{(n+1,3),} w_{(n)} + j_{(n+1,3)}) \tag{4.16}$$

$$l_{(n+1,4)} = \Delta T_w \,.\, h(T_{w(n\,)} + \Delta T_{w,} i_{ma(n)} + k_{(n+1,3),} w_{(n)} + j_{(n+1,3)}) \tag{4.17}$$

The four variables in the Runga- Kutta method are T_w, w, i_{ss} or i_{ma}, M_{ep} from the left- hand side of Eqs. (3.25), (3.26) and (3.29) for unsaturated air and Eqs. or Eqs (3.34), (3.35) and (3.36) for supersaturated air. For this reason Eqs. (1), (2) and (3) are functions of only w, i_{ma}, or i_{ss} and T_w. Most of the other variables are functions of these variables. Eqs. (4.1), (4.2) and (4.3) are not functions of M_{ep} because dM_{ep}/dT_w is function of dw/dT_w as can be seen from Eqs. (3.34), (3.35) for supersaturated air can be solved without Eq. (3.36) or Eq. (3.36) respectively.

A computer program "Poppe Size "was developed to determine the size of the cooling tower. In the above mentioned program the cooling range are divided in 100 intervals to get reasonable accuracy. Input/ output parameters are represented in Fig 4.7 and Flow chart of the program "Merkel Size" is shown in Fig. 4.9. A program "Merkel Rate" was developed to determine the outlet water temperature of cooling tower. Input/output diagram & the flow chart are shown in Fig. 4.8 and Fig. 4.10 respectively.

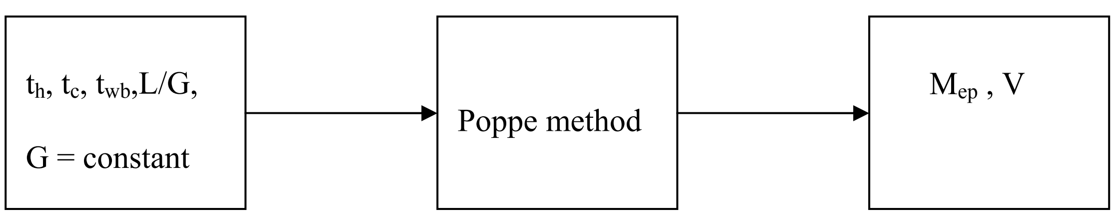
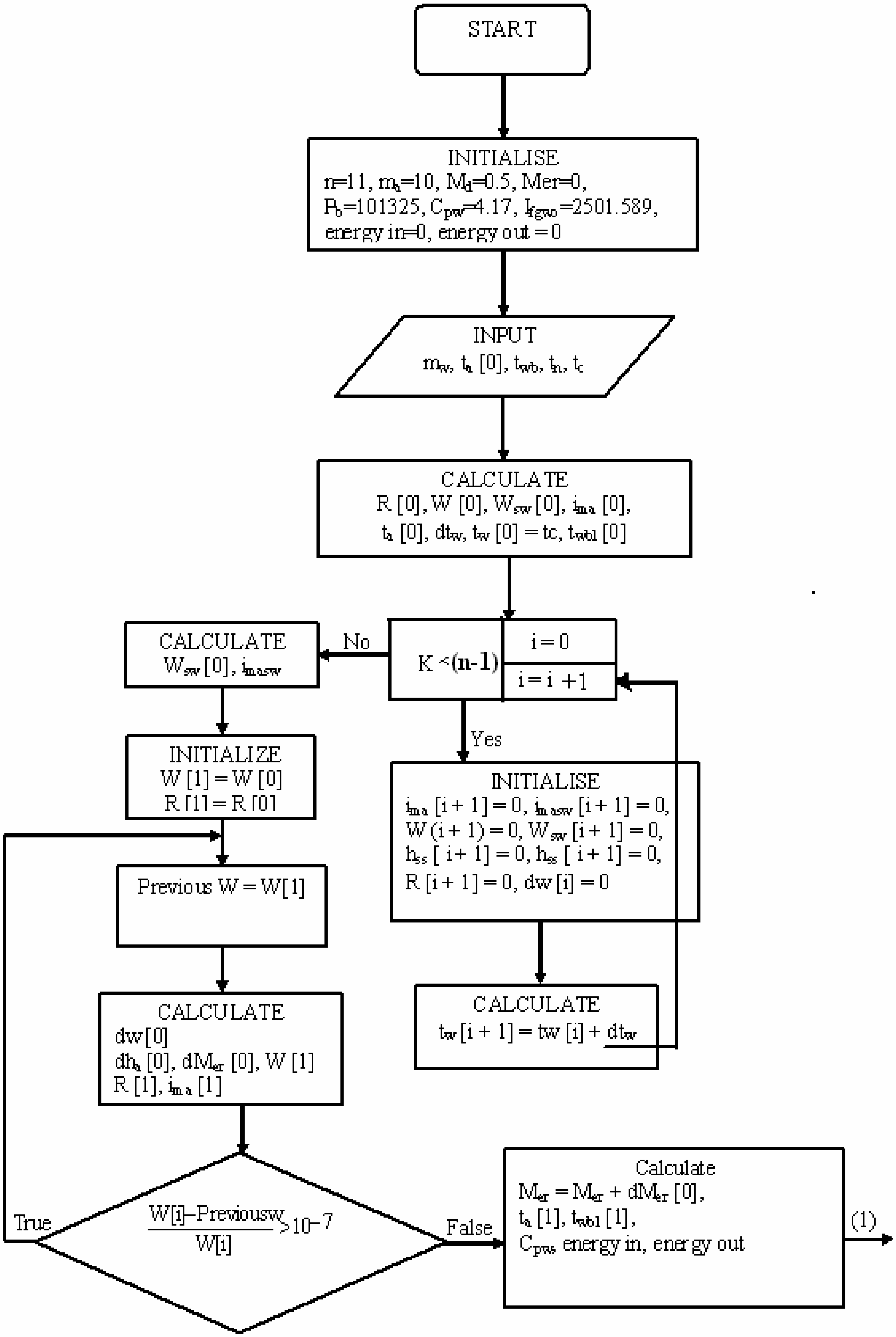


Fig 4.7 Input/output of computer program "Poppe size".

(B).


START
INITIALISE
n=11, ma=10, Md=0.5, Mer=0,
Pb=101325, Cpw=4.17, Ifgwo=2501.589,
energy in=0, energy out = 0
INPUT
mw, ta [0], twb, tn, tc
CALCULATE
R [0], W [0], Wsw [0], ima [0],
ta [0], dtw, tw [0] = tc, twb1 [0]
K < (n-1)
i = 0
i = i +1
No
Yes
CALCULATE
Wsw [0], imasw
INITIALIZE
W [1] = W [0]
R [1] = R [0]
INITIALISE
ima [i + 1] = 0, imasw [i + 1] = 0,
W (i + 1) = 0, Wsw [i + 1] = 0,
hss [i + 1] = 0, hss [i + 1] = 0,
R [i + 1] = 0, dw [i] = 0
Previous W = W[1]
CALCULATE
tw [i + 1] = tw [i] + dtw
CALCULATE
dw [0]
dha [0], dMer [0], W [1]
R [1], ima [1]
True
W[i]–Previousw / W[i] > 10–7
False
Calculate
Mer = Mer + dMer [0],
ta [1], twb1 [1],
Cpw, energy in, energy out
(1)

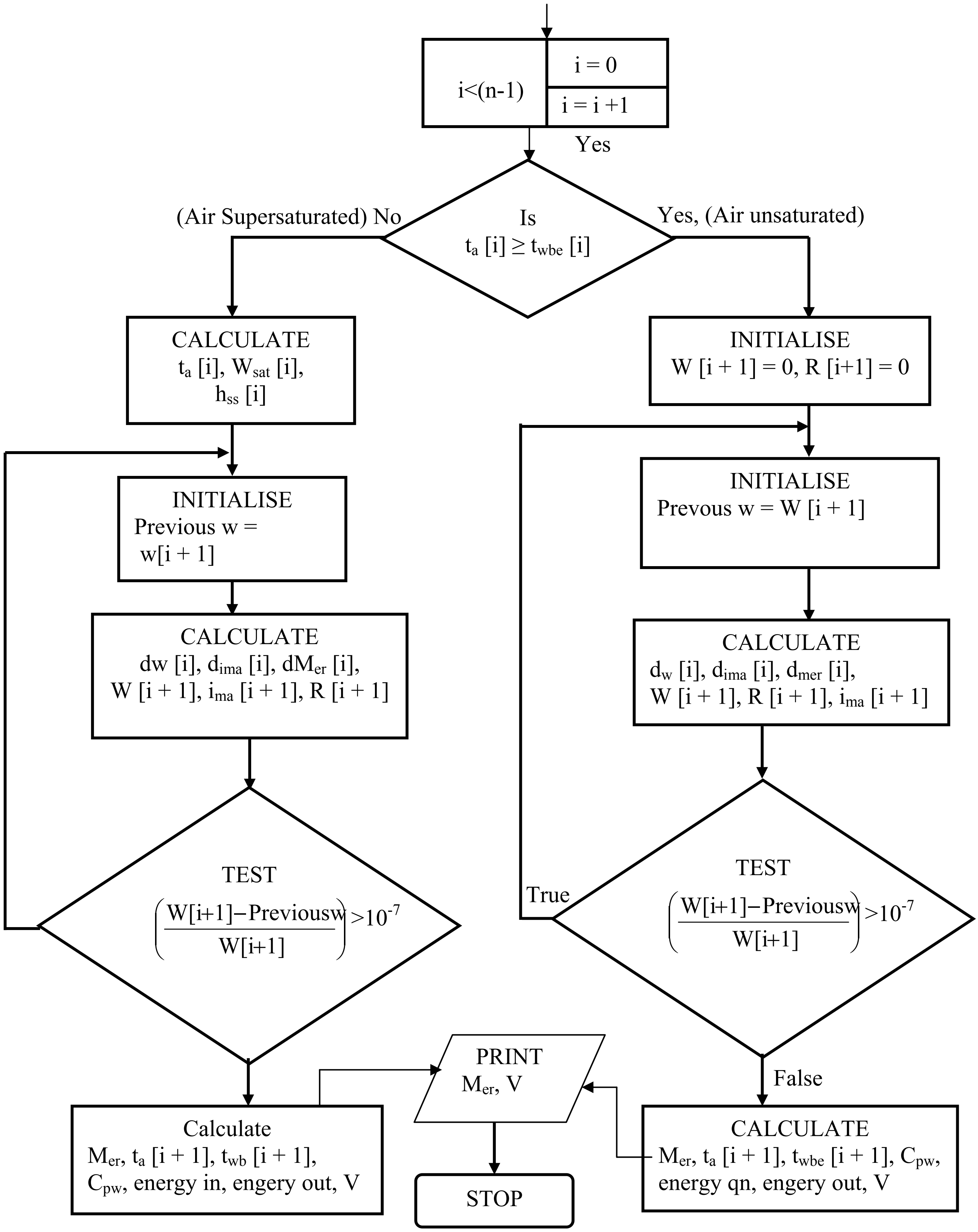
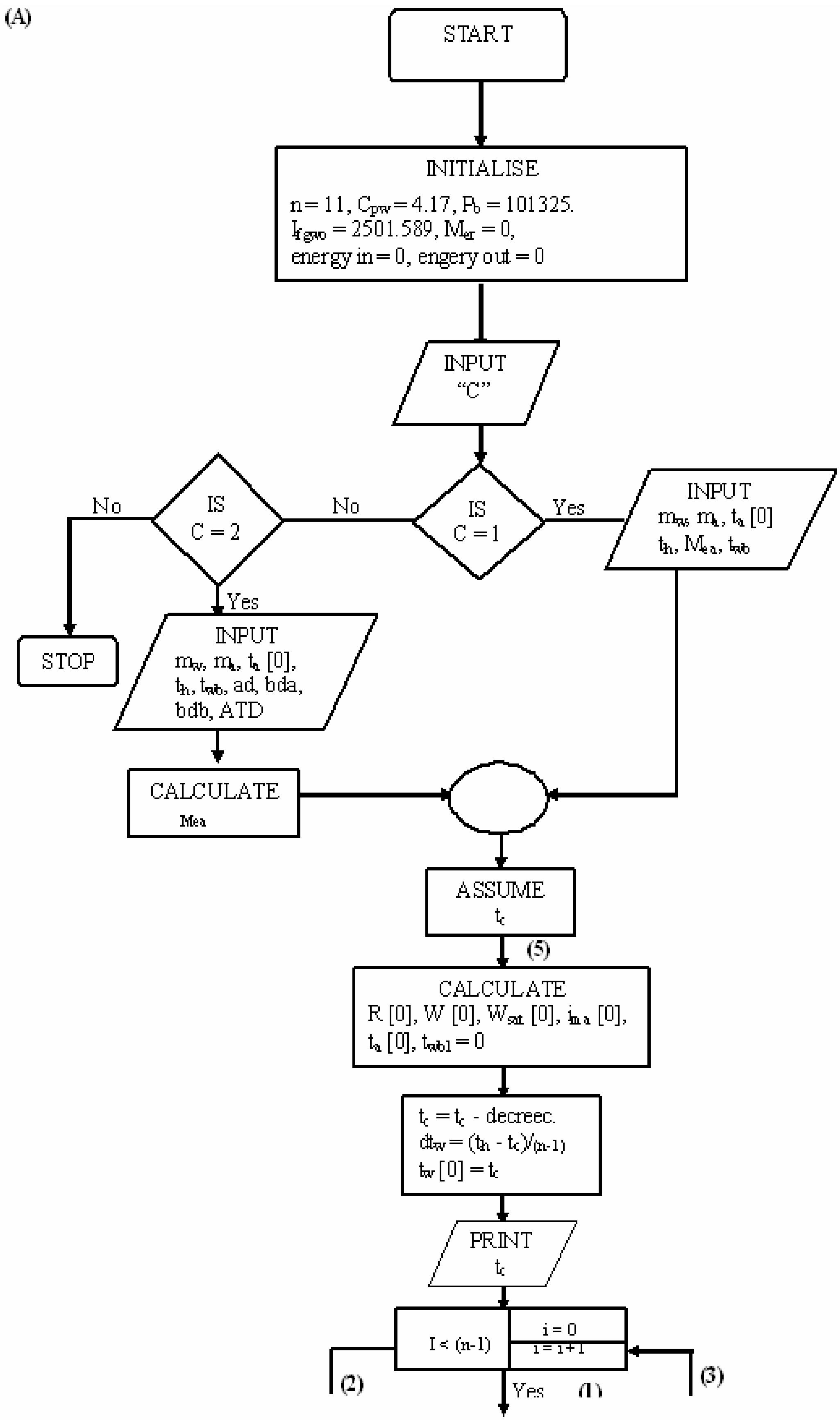
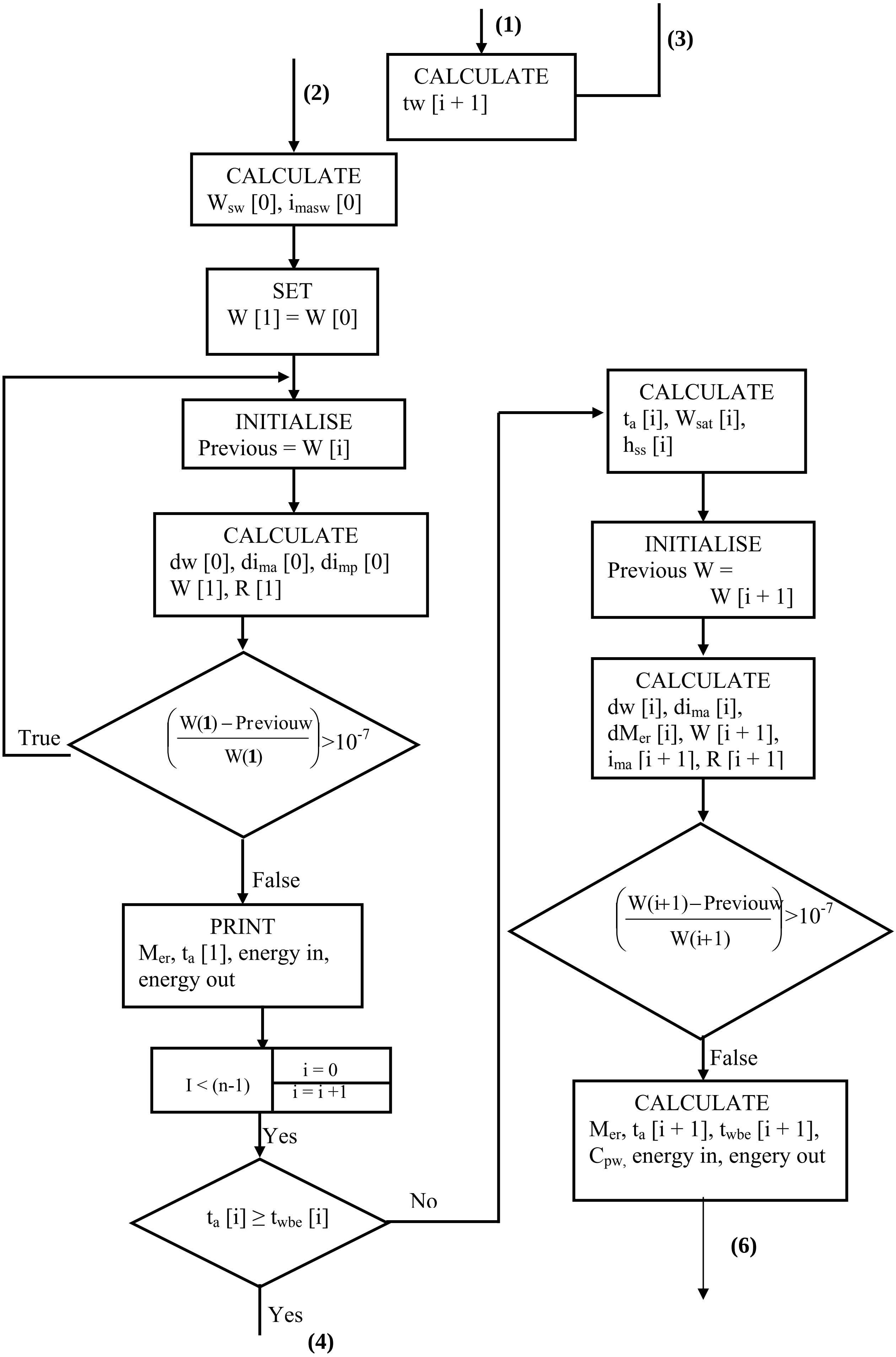


Fig.4.8 Flow Chart of Programming of Poppe Method (Design Problem)

(A)



START
INITIALISE
n = 11, Cpw = 4.17, Pb = 101325.
Ifgwo = 2501.589, Mer = 0,
energy in = 0, engery out = 0
INPUT
"C"
IS
C = 1
Yes
No
INPUT
mw, ma, ta [0]
th, Mea, twb
IS
C = 2
No
Yes
STOP
INPUT
mw, ma, ta [0],
th, twb, ad, bda,
bdb, ATD
CALCULATE
Mea
ASSUME
tc
(5)
CALCULATE
R [0], W [0], Wsat [0], ima [0],
ta [0], twb1 = 0
tc = tc - decreec.
dtw = (th - tc)/(n-1)
tw [0] = tc
PRINT
tc
I < (n-1)
i = 0
i = i + 1
(2)
Yes
(1)
(3)



(1)
CALCULATE
tw [i + 1]
(3)
(2)
CALCULATE
W_{sw} [0], i_{masw} [0]
SET
W [1] = W [0]
INITIALISE
Previous = W [i]
CALCULATE
dw [0], di_{ma} [0], di_{mp} [0]
W [1], R [1]
True
$\left(\frac{W(1) - Previouw}{W(1)}\right) > 10^{-7}$
False
PRINT
M_{er}, t_a [1], energy in, energy out
I < (n-1)
i = 0
i = i +1
Yes
t_a [i] ≥ t_{wbe} [i]
No
Yes
(4)
CALCULATE
t_a [i], W_{sat} [i], h_{ss} [i]
INITIALISE
Previous W = W [i + 1]
CALCULATE
dw [i], di_{ma} [i], dM_{er} [i], W [i + 1], i_{ma} [i + 1], R [i + 1]
$\left(\frac{W(i+1) - Previouw}{W(i+1)}\right) > 10^{-7}$
False
CALCULATE
M_{er}, t_a [i + 1], t_{wbe} [i + 1], C_{pw}, energy in, engery out
(6)

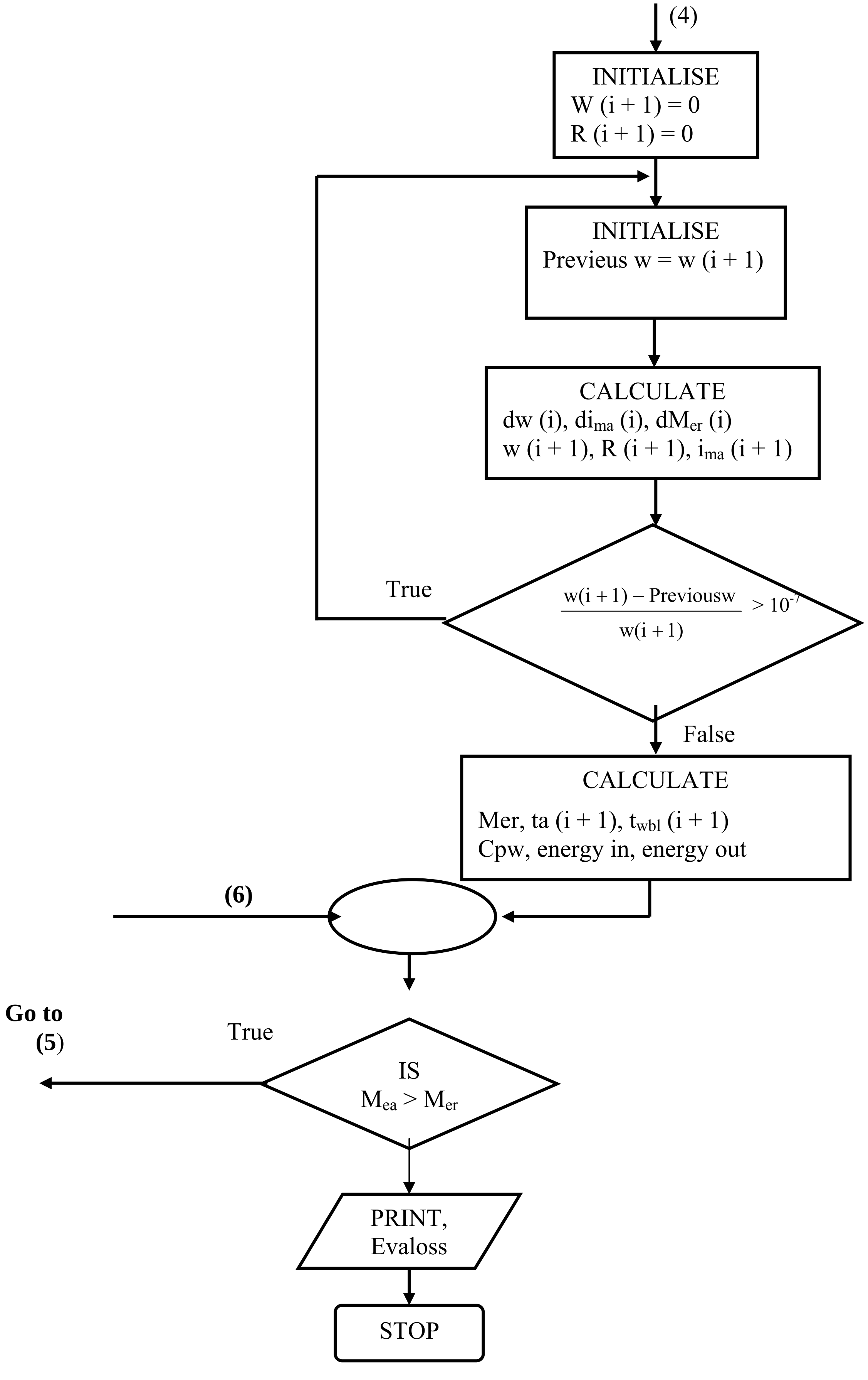


Fig. 4.9 Flow Diagram of Programming Of Poppe Method (Analysis Problem)

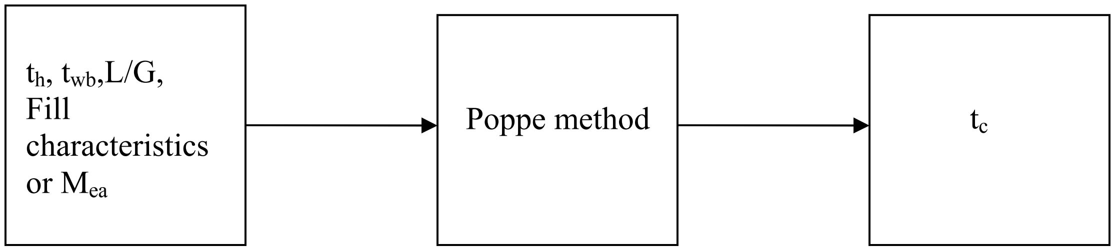


Fig 4.10 Input/output of computer program “Poppe Rate”.

4.2.2 Validation of Numerical Modeling

For validation of numerical model Poppe method for design problem of cooling tower and performance analysis of cooling tower, results are compared respectively with data taken from Li and Priddy’s handbook [2] and from Goel thesis [17].

Table 4.5 Results compared with Li and Priddy’s Hand Book [2] (For sizing problem).

	Merkel number (M_{ep})	Volume of cooling tower,m^3(V_p)
Li and Priddy’s handbook (10 Intervals)	1.51	30.20
Numerical model of Poppe method (10 Intervals)	1.83	36.71

Error percentage in Merkel number and in fill volume found 21.1 % and 21.5 % respectively. Difference in results is found because Li and Priddy’s results are based on simplifying assumptions, while Poppe consider realistic conditions and considers evaporation loss and actual condition of air during flow in cooling tower.

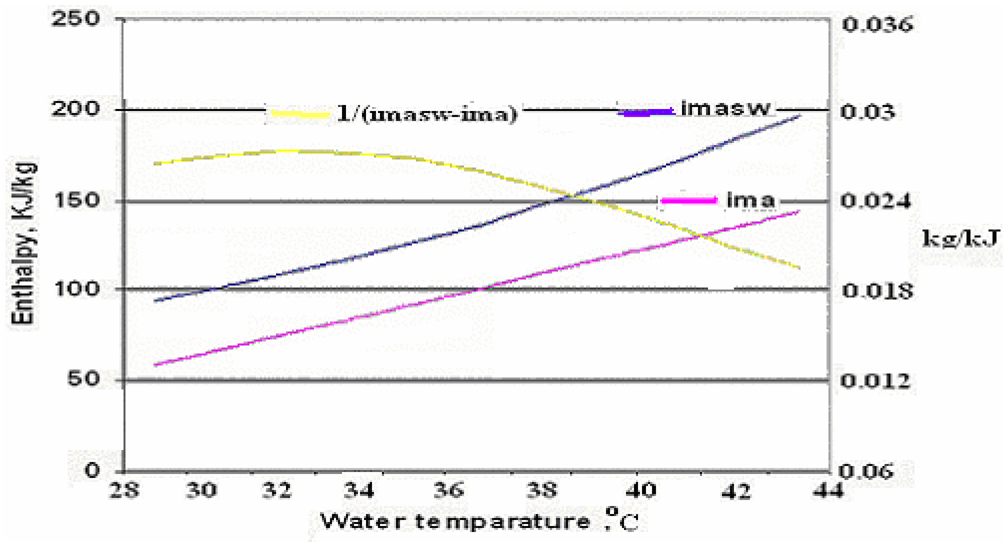


Fig.4.11 Enthalpy diagram for Poppe method

Fig.4.11 shows the enthalpy curves of the air in a counter flow wet cooling tower. Results of Fig.4.11 have been shown in Table B.1. In the i_{ma} curve i.e. the enthalpy of the air as it moves through the fill, shown in Fig. is linear due to the linear nature of Eq.(A.1). The i_{masw} curve is the saturation curve of the air at the water interface temperature. The potential for heat and mass transfer at a particular water temperature is the difference between i_{masw} and i_{ma}.

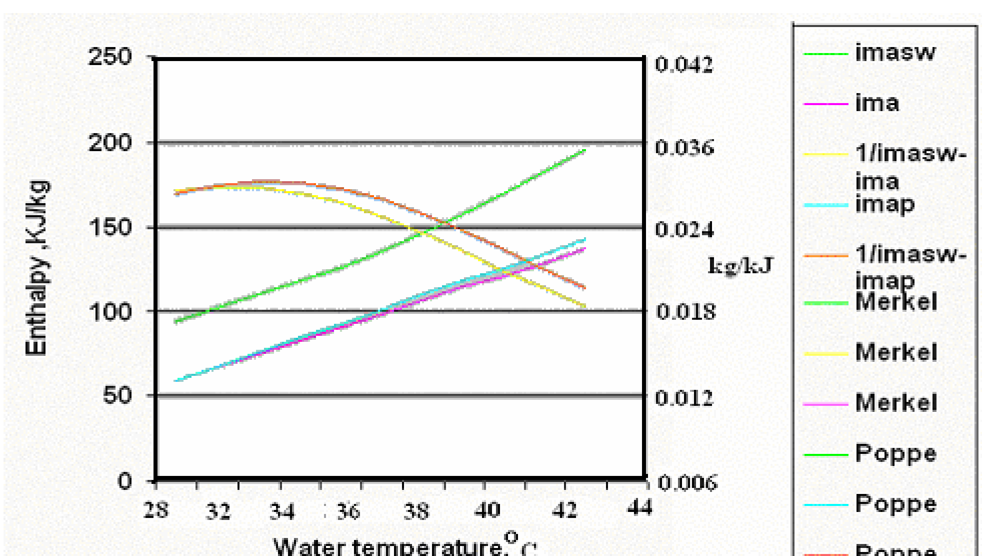


Fig.4.12 Enthalpy diagram for Merkel and Poppe method

Fig 4.12 shows the difference in the enthalpy diagrams between the Merkel and Poppe methods. The i_{masw} curves of the two methods fall on the top of each other. There is a small discrepancy in the i_{ma} curves of the two different methods, especially at the hot water side. It can be seen that the Poppe method predicts an approximately linear variation of the air enthalpy for this specific case but the gradient is different from that predicted by Merkel method. The $1/(i_{masw} - i_{ma})$ curve of Poppe method lies above the $1/(i_{masw} - i_{ma})$ curve of the Merkel method. As the transfer characteristic, or Merkel number, is a function of area under the $1/(i_{masw}-i_{ma})$ curve, the Merkel number predicted by the Merkel method

Table 4.6 Data from Goel thesis [17] (For performance analysis problem).

Initial Conditions	Initial Values
Inlet hot water temperature (t_h)	44.5 ºC
L/G ratio	1.55
Wet bulb temperature (t_{wb})	30 ºC
Merkel number available (M_{ea})	1.68

Table 4.7 Result compared with Goel thesis [17] (performance analysis problem).

	Outlet cold water temperature (tc)	Percentage evaporation Loss
Goel thesis	33.85 ºC	-
Numerical model of Poppe method.	34.48 ºC	1.64

It is clear from results shown in Table 4.5 and Table 4.7 that numerical model for Poppe method predicts results are quite close to the results shown by Li and Priddy's handbook [2] and Goel thesis [17] for both sizing and performance analysis problem of cooling tower. Percentage variation in outlet temperature value is 1.8 % and evaporation loss found 1.64 % of cooling fluid.

4.3 e-NTU METHOD

The details of Merkel method is described in chapter 3. Main equations of e-NTU method are 3.46, 3.47, 3.49, 3.54 and 3.57.

4.3.1 Solution Procedure Applied

A one –increment design (N=1) may be performed very quickly with e-NTU method. The e-NTU method is subject to approximations involved in linear zing the i_{masw} versus T curve as a straight line. However, the desired accuracy can be obtained by breaking the design down into N increments. Traditional cooling tower design methods typically use an incremental method. One may use the correction factor (S) given by equation (4) for the e-NTU method, which essentially gives a two- increment design to do this, one redefines i_{masw1} and i_{masw2} as (i_{masw1} - S) and (i_{masw2} - S), respectively. Hence, the definition of e is rewritten as

$$e = m_w.C_{pw}.(T_{w1} - T_{w2})/ m_{min} (i_{masw1} - S - i_{ma1}) \quad (4.3.1)$$

A typical problem that often arises in the cooling tower design is the determination of the NTU when T_{wb}, R, A, and m_a/m_w are given. The traditional method of solution is to use the curves given in publications by Kelly and Cooling Tower Institute [3]. These curves are based on use of the Merkel method, and were generated for a wide range of practical operating conditions. The Cooling Tower Institute curves [3] were generated using the Tchebychev integration method with three increments (N=3). A simple procedure for a one-increment design using enthalpy correction factor is outlined below:

1. Calculate the slope of saturation line, $f = \Delta i_{masw1}/R$.
2. Calculate $m_{w*} = m_w.C_{pw}/f$ and compare to ma to determine $C_R = m_{min}/m_{max.}$
3. Find $\Delta i_{ma} = (m_w/m_a).C_{pw}.R$.
4. Calculate the effectiveness $e = (m_a.\Delta i)/ [m_{min}. (i_{masw}1 - S - i_{ma1})]$.
5. Read (or calculate) the e-NTU from chart (or equation).

Counter flow rating calculation may be performed without iterations using the following procedure:

1. Specify the leaving water temperature.
2. Set several ΔT_w increments and calculate the $K_m.A/m_a$ values for each increment.
3. When the calculations for the last increment yields $\sum K_m. A/m_a$ greater than the given value decrease the ΔT_w for the last increment and continue until $\sum K_m.A/m_a$ equals the given value.

A computer program "e-NTU Size "was developed to determine the size of the cooling tower. In the above mentioned program the cooling range are divided in 11 intervals to get reasonable accuracy. Input/ output parameters are represented in Fig 4.13 and Flow chart of the program "e-NTU Size" is shown in Fig. 4.14. A program "e-NTU Rate" was developed to determine the outlet water temperature of cooling tower. Input/output diagram & the flow chart are shown in Fig. 4.15 and Fig. 4.16 respectively.

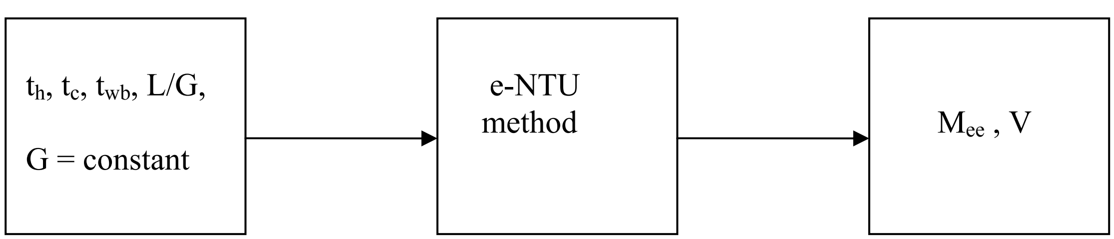


Figure 4.13 Input/output of program "e-NTU Size".

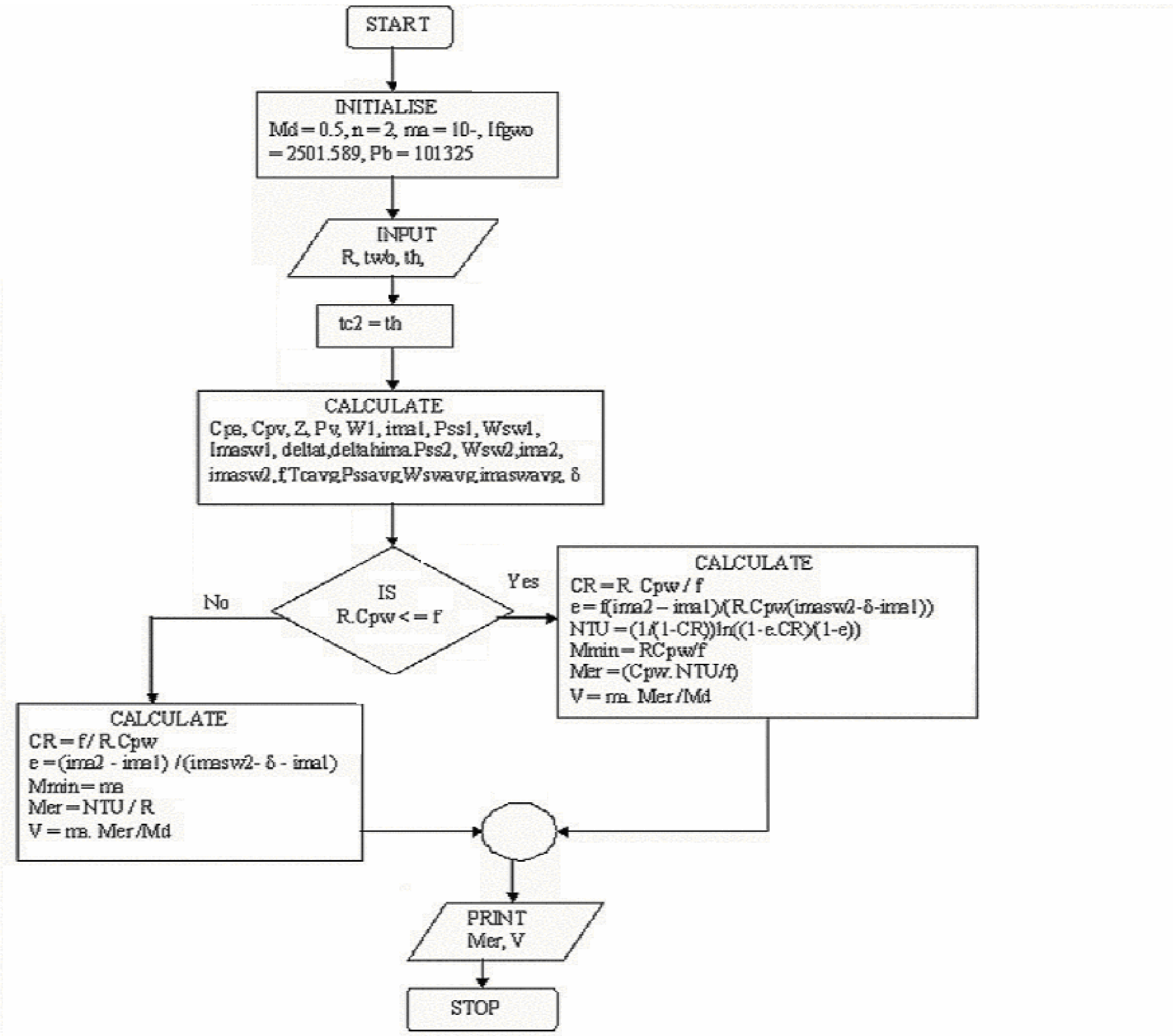


Fig. 4.14 Flow Chart Of Program Of e-NTU Method For Design Problem Of Cooling Tower.

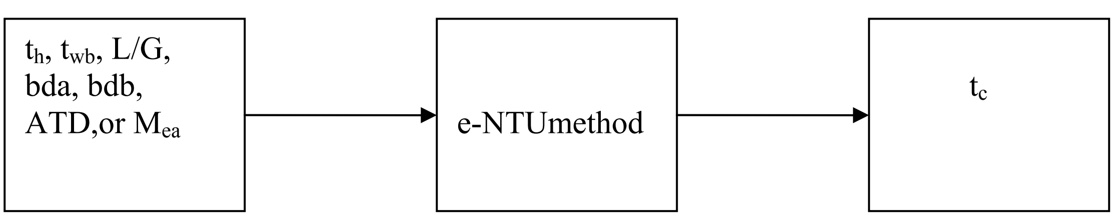
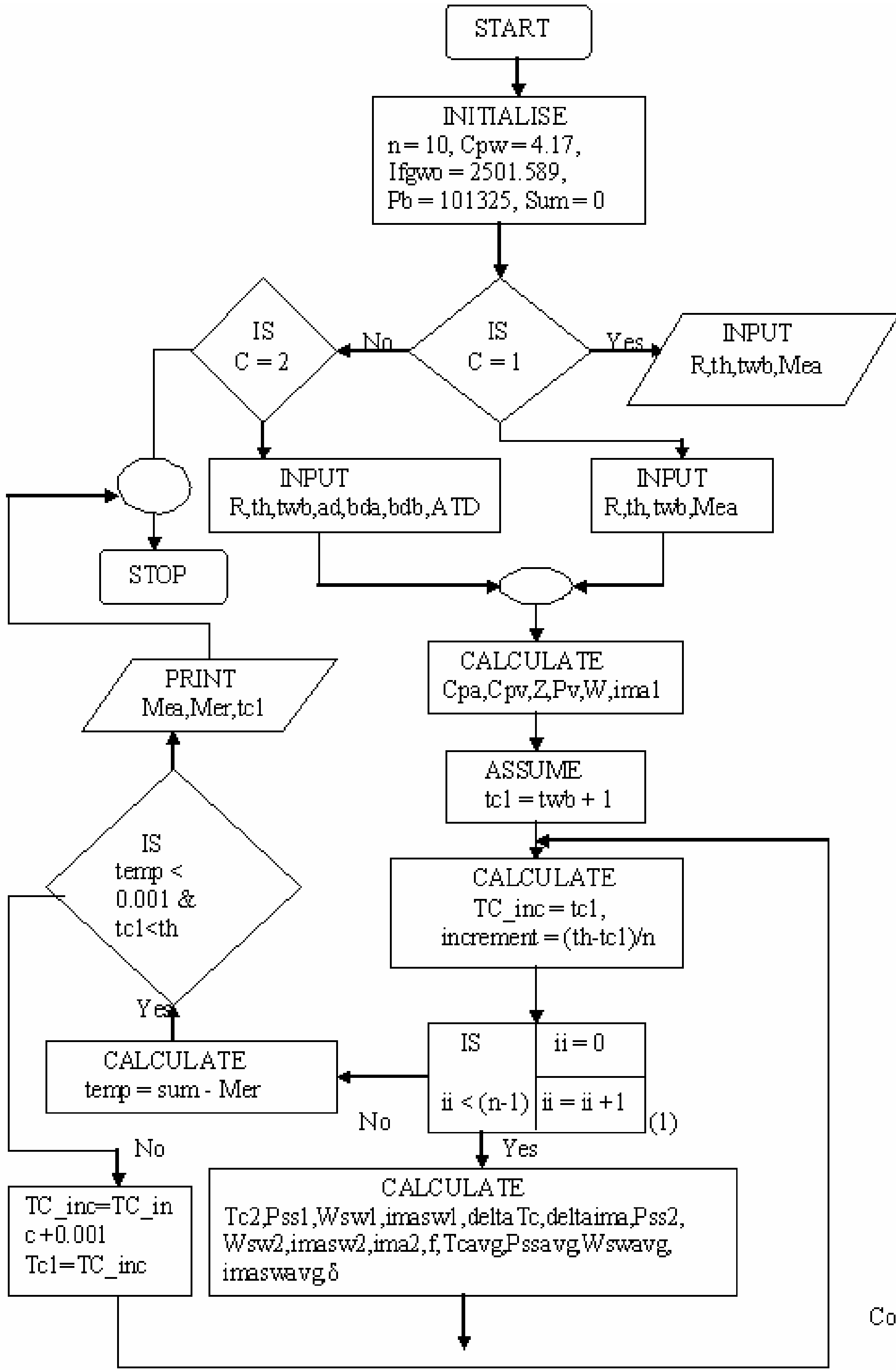


Fig 4.15 Input/output of program "e-NTU Size".


START
INITIALISE
n = 10, Cpw = 4.17,
Ifgwo = 2501.589,
Pb = 101325, Sum = 0
IS
C = 2
No
IS
C = 1
Yes
INPUT
R,th,twb,Mea
INPUT
R,th,twb,ad,bda,bdb,ATD
INPUT
R,th, twb,Mea
STOP
CALCULATE
Cpa,Cpv,Z,Pv,W,ima1
ASSUME
tc1 = twb + 1
PRINT
Mea,Mer,tc1
IS
temp <
0.001 &
tc1<th
CALCULATE
TC_inc = tc1,
increment = (th-tc1)/n
Yes
IS
ii = 0
ii < (n-1)
ii = ii +1
(1)
CALCULATE
temp = sum - Mer
No
No
Yes
TC_inc=TC_in
c+0.001
Tc1=TC_inc
CALCULATE
Tc2,Pss1,Wsw1,imasw1,delta Tc,deltaima,Pss2,
Wsw2,imasw2,ima2,f,Tcavg,Pssavg,Wswavg,
imaswavg,δ

Continue……

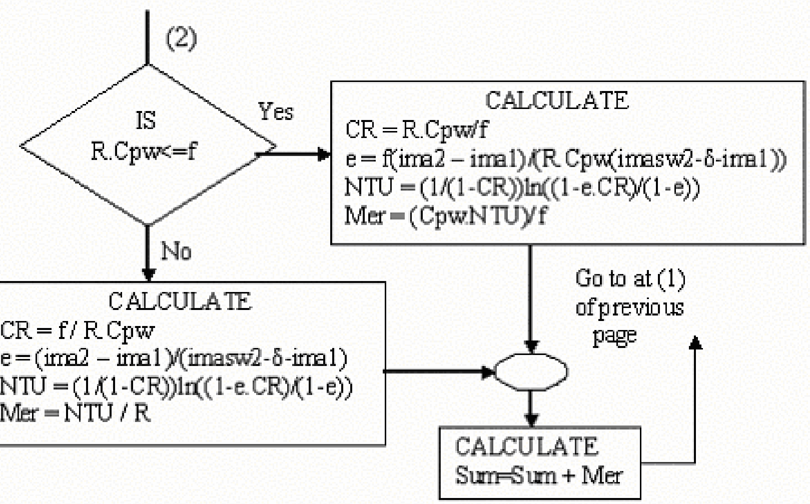


Fig. 4.16 Flow Diagram Of e-NTU Method Program For Analysis Problem Of Cooling Tower.

4.3.2 Validation of Numerical Modeling

For validation of numerical model e-NTU method for design problem of cooling tower and performance analysis of cooling tower, results are compared respectively with data taken from H. Jabber and R. L.Webb [3].

Table 4.8 Data from H. Jabber and R. L.Webb [3] (For sizing problem).

Initial Conditions	Initial Values
Inlet hot water temperature (t_h)	35 ºC
L/G ratio	1
Wet bulb temperature (t_{wb})	25 ºC
Outlet cold water temperature (tc)	30 ºC
Air mass flow rate (G) (gm/sec)	10

Table 4.9 Results compared from H. Jabber and R. L.Webb [3] (For sizing problem).

	Merkel number (M_{ee})	Volume of cooling tower(V_e), m^3
From Jabber and Webb	0.74	14.80
Numerical model of e-NTU method	0.76	15.34

Results displayed in Table 4.9 shows that error percentage in Merkel number and in fill volume found 2.7 % and 3.6 % respectively when results data of model compares with Jaber and Webb [3] results. Results are very close and error percentage is very less it validate this model with data available in literature.

Table 4.10 Data from H. Jabber and Webb [3] (Performance analysis problem).

Initial Conditions	Initial Values
Inlet hot water temperature (t_h)	35 ºC
L/G ratio	1
Wet bulb temperature (t_{wb})	25 ºC
Merkel number available (M_{ea})	0.74

Table 4.11 Result compared with H. Jabber and Webb [3] (Performance analysis problem).

	Outlet cold water temperature (tc)
From Jabber and Webb	30 ºC
Numerical model of e-NTU method.	33.82 ºC

A result in Table 4.11 shows that numerical model for e-NTU method predicts quite close results to the results shown by H. Jabber and R.L.Webb [3] performance analysis problem of cooling tower. Percentage error in predicted outlet temperature found 10.6 %.

Chapter 5
Results and Discussion

Various computer programs related to Merkel, e-NTU, and Poppe methods are discussed in chapter 4. In the present chapter output of these programs are computed for wide range of initial conditions. Numerical models of Merkel, e-NTU and Poppe methods for both sizing and performance analysis of cooling tower are compared in the subsequent section. Comparison of results is based on wide range of initial conditions [1]. Various graphs are plotted for Merkel number, fill volume, outlet temperature of cooling tower and percentage error in these parameters with respect to water to air mass flow rate ratio for Merkel, e-NTU, and Poppe methods. In this present chapter some valuable conclusion also drawn with graphs.

5.1 Selection of Initial Conditions

It is now necessary to compare the values of the cooling tower volume for designing of cooling tower and outlet cold water temperature for performance analysis of cooling tower calculated by Merkel, e-NTU and Poppe methods.

Initially, cover a wide range of operating conditions in broad steps; the values given in Table 5.1 are taken from J.W. Sutherland [1]. These seven combinations were used with four values of L/G, namely, 0.5, 1.0, 1.5, and 2.0, keeping G at 10 kg/s throughout. The value of atmospheric pressure, P_b, was taken as the standard value of 101325 Pa. However, in order to compare the different methods mentioned above, a typical value of 0.5 kg/m^3.s was taken for $h_d.A_v$ [1] and the values of tower volumes $V_{M,}$ V_E and V_P compared.

Table 5.1 Initial operating conditions for input in Merkel, e-NTU and Poppe method computer programs (For sizing problem of cooling tower)

Run	t_{wb} (ºC)	t_d (ºC)	t_h (ºC)	t_c (ºC)	A (ºC)	R (ºC)
1.	30	35	50	40	10	10
2.	20	25	50	40	20	10
3.	20	25	40	30	10	10
4.	10	15	50	40	30	10
5.	10	15	40	30	20	10
6.	20	35	50	40	20	10
7.	20	35	40	30	10	10
8.	30	35	60	40	10	20

Fill characteristic values for Marley MC 67 are given below : -(Refer Appendix C)

(a). Fill characteristic value $a_d = 1.495$, $b_{da} = - 0.63$, $b_{db} = - 0.35$.

(b). Fill height value ATD = 0.9

For sizing and performance analysis problem of cooling towers different computer programs are developed. For computer programming of Merkel method cooling range (integration limit) is divided in 100 parts to improve the accuracy of results, while for Poppe method cooling range divided in 10 intervals, It is necessary to divide the fill into more than one interval to capture, as accurately as possible, the point at which the air becomes supersaturated air. For program in of e-NTU method fill divided into 11 divisions to get accurate results.

5.2 Comparison of Models for Sizing Problem

In design (sizing) problem of cooling tower Merkel number and volume of cooling tower is calculated by Merkel and e-NTU methods. In the literature it is mentioned that both Merkel and e-NTU methods work on same simplifying assumptions. Therefore these models are compared first in this chapter.

Percentage error in cooling tower volume given as:

$\% E = 100 X (1 - V_e/V_M)$

Where

$V_{e =}$ Tower volume calculated by e-NTU method;

V_M = Tower volume calculated by Merkel method

Fig. 5.1 represents the variation of Merkel number calculated by Merkel method with water to air mass flow rate ratio for initial conditions represented in Table 5.1. Results data of Fig. 5.1 and Fig. 5.2 are represented in Table B.3 & Table B.4.

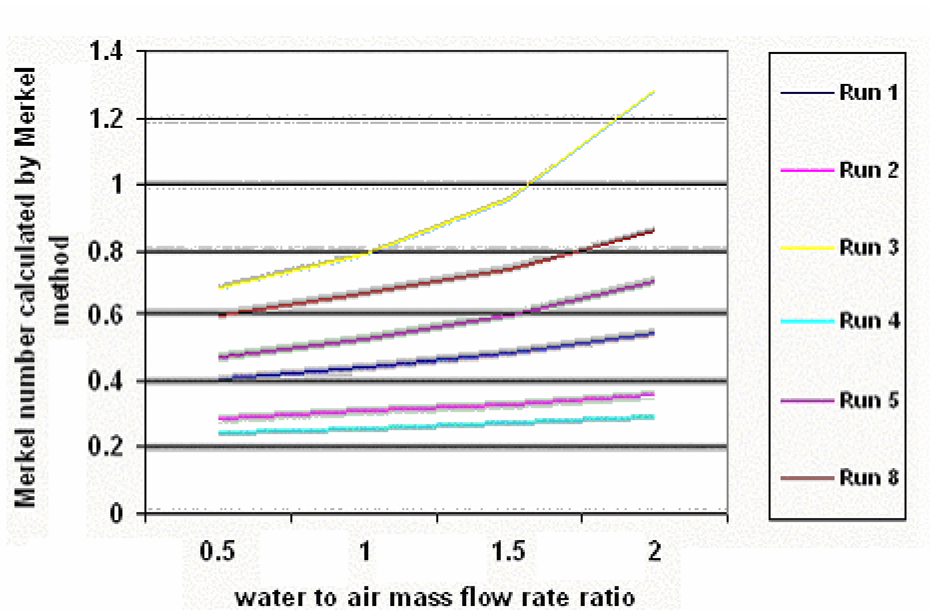


Fig. 5.1 Variation in Merkel number calculated by Merkel method with water to air mass flow rate ratio

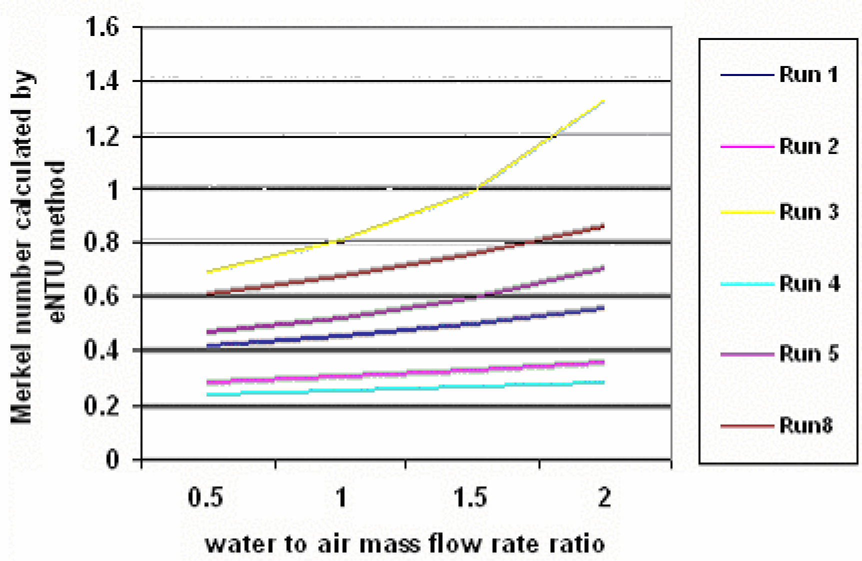


Fig. 5.2 Variation in Merkel number calculated by e-NTU method with water to air mass flow rate ratio

Results compared in Fig 5.1 & 5.2 are similar to each other and maximum percentage error variation calculated less than 1.08 %. Merkel number calculated by e-NTU method is approximately 1.6 % less than, calculated by Merkel method.

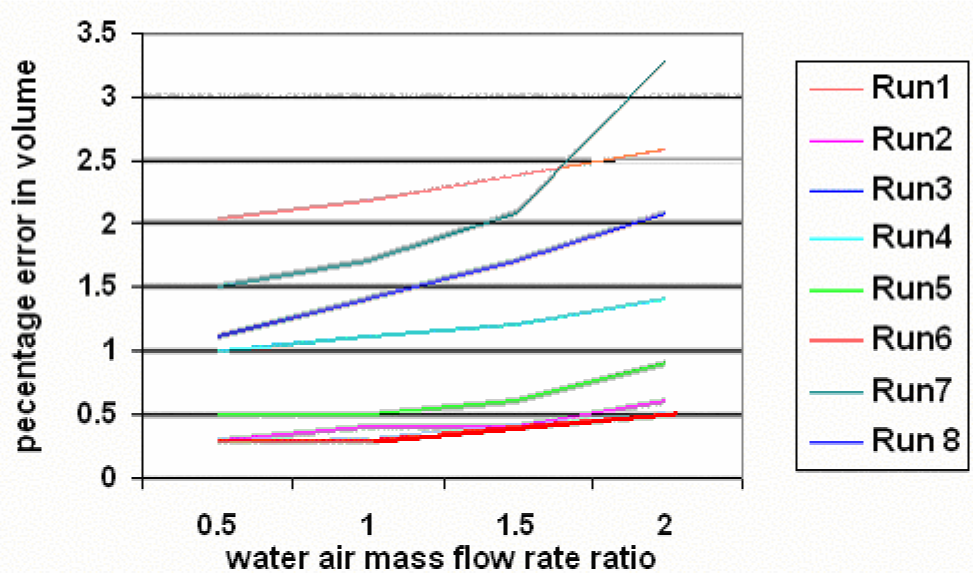


Fig. 5.3 Variation of tower volume calculated by Merkel and e-NTU methods with water to air mass flow rate ratio

Fig 5.3 shows percentage error in volume to compare Merkel and e-NTU method varies from 0.3 % to 3.3 %. These results are shown in Table B.5. The error is insignificant because both Merkel and e-NTU methods work on same simplifying assumptions. Therefore main comparison will be between Merkel and Poppe method taken in to account in further discussion.

In design problem of cooling tower Merkel number and volume of cooling tower is calculated by Merkel and Poppe method. Percentage error in cooling tower volume plotted here in Fig.5.3 with water to air mass flow rate ratio.

Percentage error in cooling tower volume calculated by Merkel and Poppe method is given as:

$\%E=100 \times (1-V_M/V_P)$

Where

V_M = Tower volume calculated by Merkel method

V_P = Tower volume calculated by Poppe method

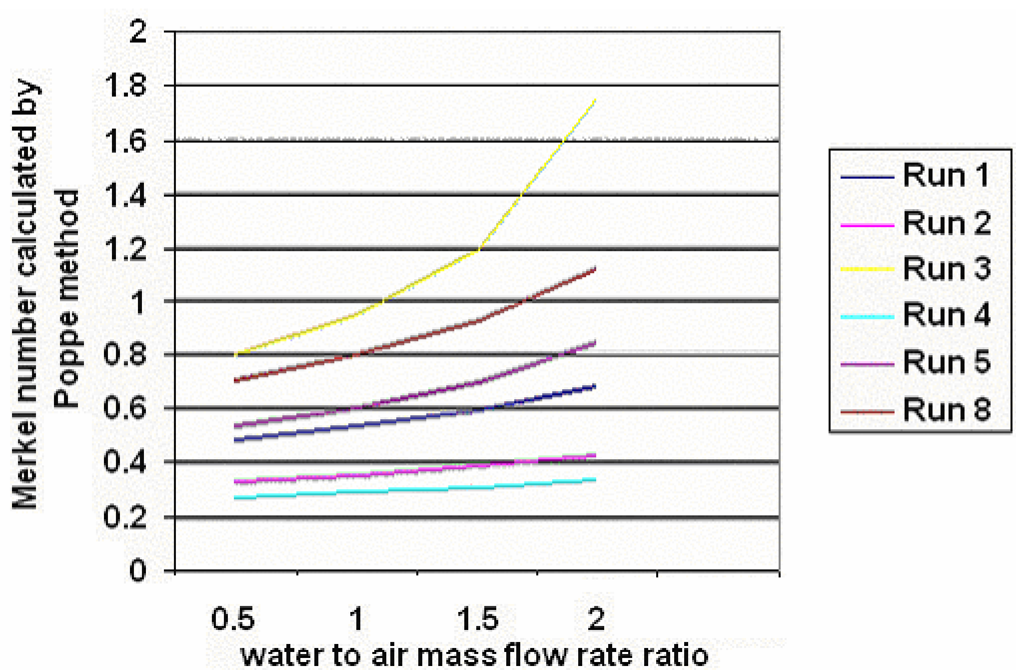


Fig. 5.4 Variation in Merkel number calculated by Poppe method with water to air mass flow rate ratio

Fig. 5.4 represents variation in Merkel number calculated by Poppe method with water to air mass flow rate ratio. Result varies from 0.2 % to 1.8 % for various initial conditions. Results data of Fig. 5.4 is represented in Table B.6.

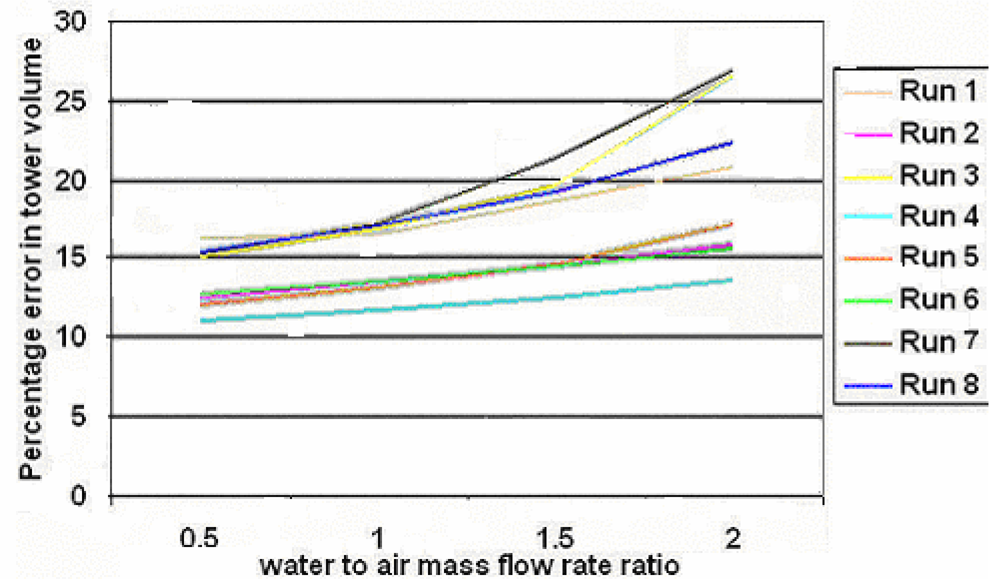


Fig. 5.5 Variation of tower volume error calculated by Merkel and Poppe method with water to air mass flow rate ratio

Percentage error variation in tower volume for Poppe method has shown in Fig.5.5. Results data of Fig. 5.5 is represented in Table B.7. From Fig. 5.5, it can be seen that for a value of m_w/m_a 0.5 of the range of errors in tower volume experienced is from 11 to 16.2 (average13.6); for 1.0 from 17.12 to 11.7 (average14.45); for 1.5 from 19.8 to 12.5(average 16.15 percentage); and for m_w/m_a equal to 2.0 the range of E is from 13.6 to 27.0 percentage (average20.3 percentage). The overall average error for the four values of m_w/m_a is 16.3 percentages. There is no significance in the crossing over of some of the error curves in Fig 4. It can be explained by the behavior of the difference $w_{sw} – w$ in the Poppe method in the relation to that of $h_{sw} – h$ in the Merkel method.

The results for the first sever sets of conditions in Table 5.1 are shown in fig. 5.3 and 5.5. As the ratio m_{w1}/m_a increases at a fixed value of m_a, and fixed air and water states, the same quantity of air is required to cool more water. Thus the tower volumes increases Fig.5.5, because when Merkel number increases tower volume also increases and the percentage error, E, (i.e. the underestimation of tower volume if the Merkel method is used in comparison of Poppe method is used) also increases Fig. 5.3 this means that effect of water loss by evaporation is more important the higher the water flow rate. It should be remembered that the value of m_w/m_a reduces from top to the bottom of the tower for Poppe method Fig. 5.4, whereas is a constant for Merkel method. The value of V_p is obtained from M_{ep} by multiplying M_{ep} by m_a and divided by $h_d.A_v$

A number of calculations regarding cooling tower behavior at affixed value of m_w/m_a can be drawn from Fig. 5.4 to Fig. 5.7 as follows:

1. For the same range and approach, the higher the wet bulb temperature the smaller the tower volume (Merkel number): i.e., M_{ep} (Run 1) < M_{ep} (Run 3) and M_{ep} (Run2) < M_{ep} (Run 5) (The numbers in parenthesis refer to the appropriate set of conditions in Table 5.1). Tower size varies inversely with wet-bulb temperature. When heat load, range, and approach values are fixed, reducing the design wet-bulb temperature increases the size of the tower see Fig. 5.6. Results data of Fig. 5.6 is represented in Table B.8.

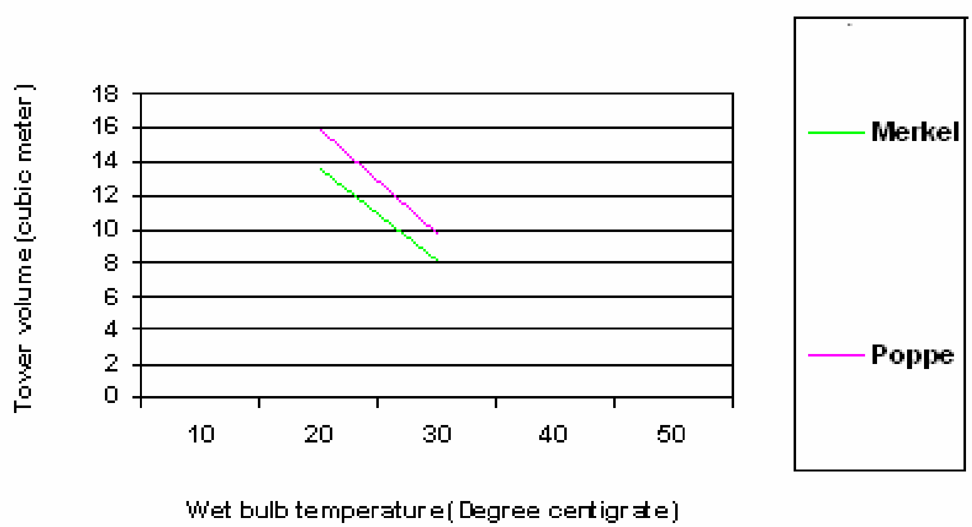


Fig. 5.6 Variation in tower volume with Wet bulb temperature for same range and approach (Data taken for Run 1 and Run 3)

2. For the same wet- bulb temperature and the larger the approach the smaller the tower: i.e., M_{ep} (Run 2) < M_{ep} (Run 3), M_{ep} (Run 4) < M_{ep} (Run 5) and M_{ep} (Run 6) < M_{ep} (Run 7) Tower size varies inversely with approach. A longer approach requires a smaller tower see Fig. 5.7. Conversely, a smaller approach requires an increasingly larger tower. Results data of Fig. 5.7 is represented in Table B.9.

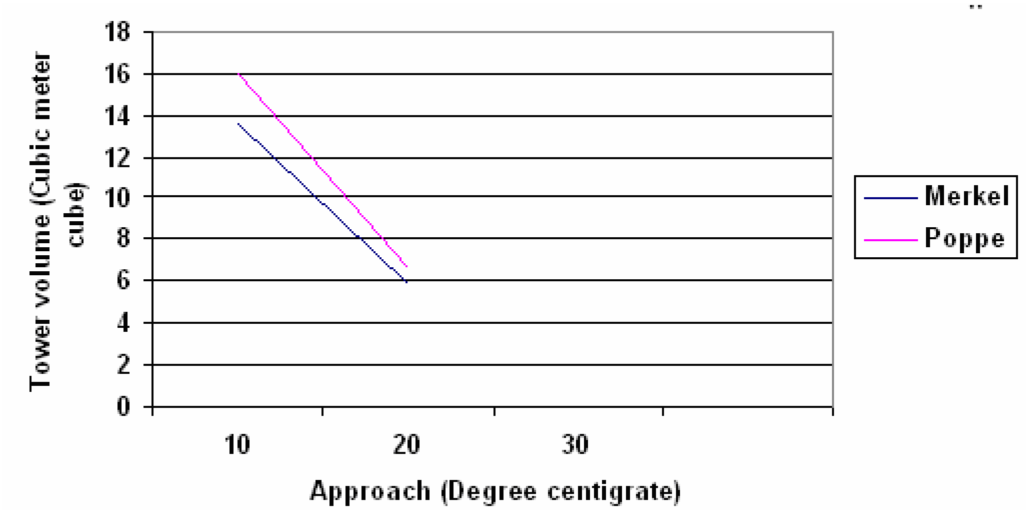


Fig. 5.7 Variation tower volume with approach for same wet bulb temperature (for Run 2 and Run 3 at $m_w / m_a = 0.5$)

3. For the same initial and final water temperatures, the lower the inlet air wet bulb temperature the smaller the tower: i.e., M_{ep} (Run 2) < M_{ep} (Run 1).

The results in Fig. 5.5 which are not obvious, e.g., cooling water from 50 to 40 °C with an inlet air wet bulb temperature of 30 °C (Run 1) requiring a smaller tower than for cooling water from 40 to 30 °C at 10°C wet bulb temperature (Run 5), can be explained by the behavior of the difference w_{sw} - w and its effects on M_{ep}. The explanation is more readily understandable in terms of the enthalpy driving force h_{sw} – h and thus M_{eM}, which are encountered in the Merkel method (Enthalpy diagram for Merkel method in previous chapter).Merkel number, determined by Poppe and e-NTU approaches are respectively approximately 16.3% higher and 1.6% lower than Merkel number determined by Merkel method.

5.3 Comparison of Models in Performance Analysis Problem

In performance analysis problem of cooling tower outlet water temperature of cooling tower is calculated by Merkel and Poppe method. Outlet water temperature of cooling tower plotted here in Figs. (5.9) and (5.10).Fig (5.11) shows error variation in outlet water temperature calculated by Merkel and Poppe method with water to air mass flow rate ratio. Results data of outlet water temperature of cooling tower for Figs. (5.9) (5.10) and (5.11) are represented in Table B.10, B.11 and in Table B.12 respectively.

$\% E = 100 \times (1 - t_m/t_p)$

Where

t_m = Outlet water temperature calculated by Merkel method

t_p = Outlet water temperature calculated by Poppe method

Fig. 5.8 represents outlet water temperature calculated by Merkel method with water to air flow rate ratio. Result data are shown in Table B.10.

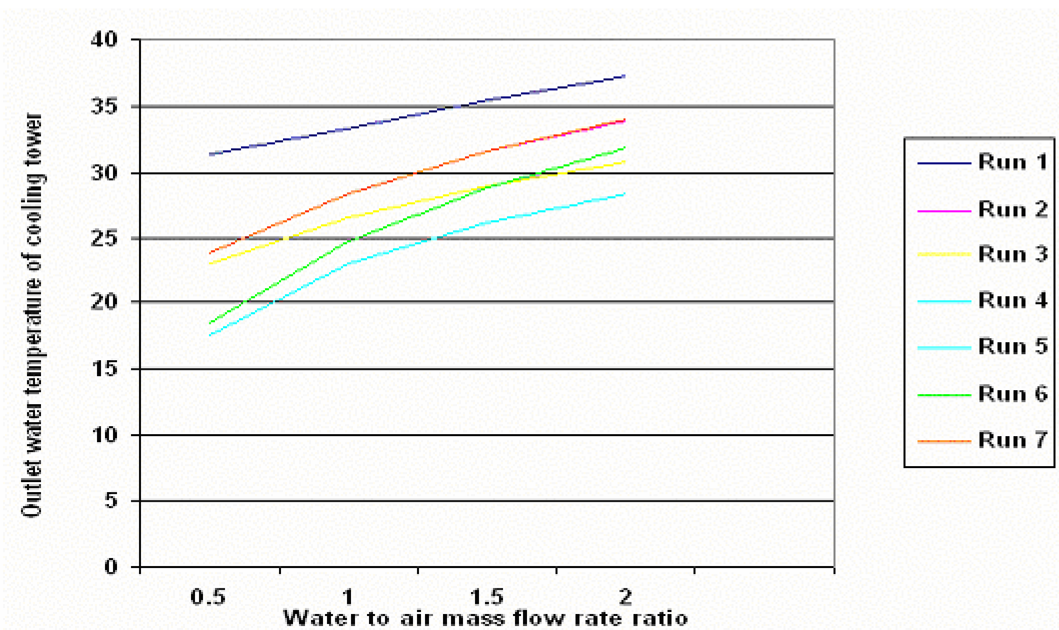


Fig. 5.8 Variation of outlet water temperature calculated by Merkel method with water to air flow rate ratio

Fig. 5.8 represents that as the ratio m_{w1}/m_a increases at a fixed value of m_a, and fixed air and water states, the same quantity of air is required to cool more water. Thus the cooling tower outlet temperature increases.

Fig. 5.9 represents variation outlet water temperature calculated by Poppe method with water to air flow rate ratio. Result data are shown in Table B.11.

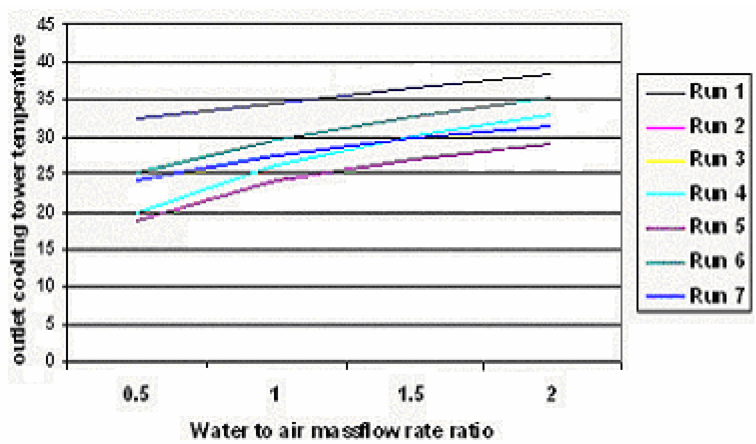


Fig. 5.9 Variation of water outlet temperature calculated by Poppe method with water to air mass flow rate ratio

Fig. 5.9 represents that as the ratio m_{w1}/m_a increases at a fixed value of m_a, and fixed air and water states, the same quantity of air is required to cool more water. Thus the cooling tower outlet temperature increases. It is found that outlet water temperature varies minimum 7.7 % (For Run 1) to maximum 65.9 % (For Run 4) for various initial conditions.

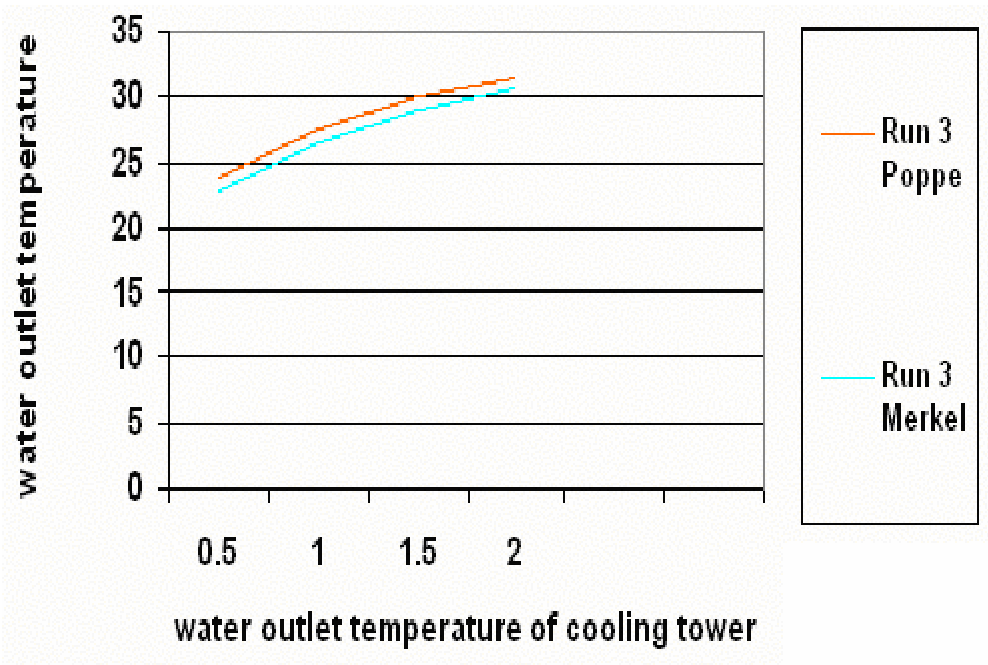


Fig. 5.10 Variation in Outlet Water Temperature Calculated By Poppe and Merkel Methods With Water to Air Mass Flow Rate Ratio.

Fig 5.10 shows that outlet temperature predicted smaller Merkel Method than calculated by Poppe method, predict nearly the same cooling tower water outlet temperature as obtained by more rigorous Poppe method.

A very small difference in water outlet temperature is due to the fact that the Merkel and Poppe methods predict different air outlet conditions causing the draft to be different in two cases. But Poppe method calculates accurate water outlet temperature because Merkel number work on some simplifying assumptions. While Poppe method work on actual conditions and consider evaporation loss in cooling tower.

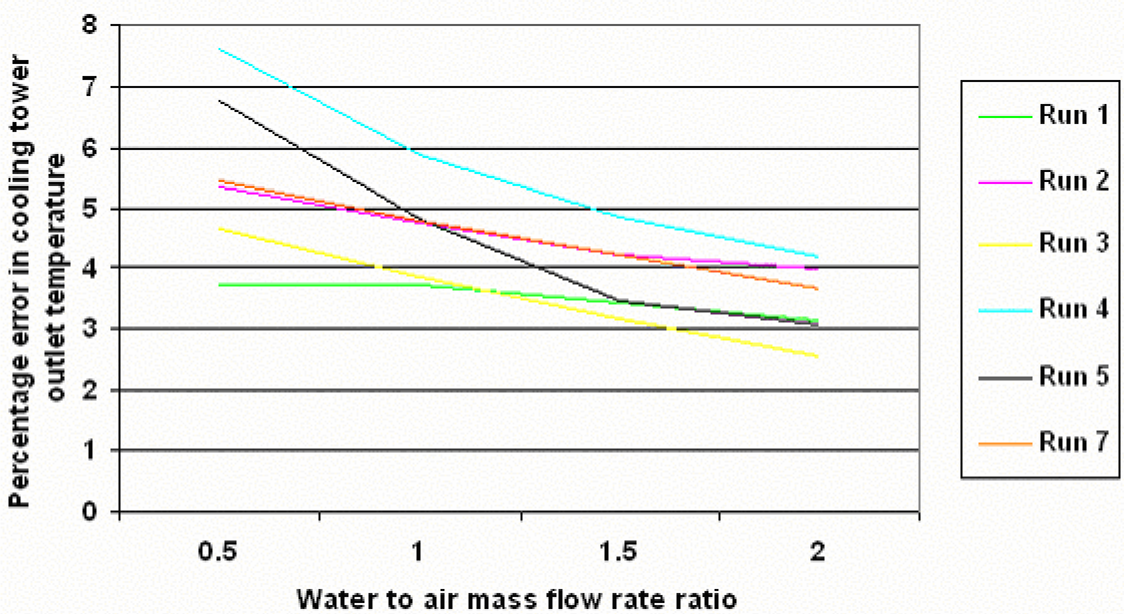


Fig. 5.11 Variation in the percentage error in cooling tower outlet water temperature with water to air flow rate ratio for Poppe and Merkel method

The results for the first sever sets of conditions in Table 5.2 are shown in Fig 5.11. Results data of Fig. 5.11 is represented in Table B.13. As the ratio m_{w1}/m_a increases at a fixed value of m_a, and fixed air and water states, the same quantity of air is required to cool more water. Thus the cooling tower outlet temperature percentage error decreases.

From Fig. 5.11 , it can be seen that for a value of m_w/m_a of the 0.5 range of errors in outlet temperature of cooling tower experienced is from 3.7 %(for Run 1) to 7.6 %(for run 4) (average 5.65 %; for 1.0 from 3.7 % (for run 1) to 5.87 % (for run 4)(average 4.78 %); for 1.5 % from 3.16 % (for run 3) to 4.87 % (for run 4)(average 4.01 %); and for m_w/m_a equal to 2.0 the range of E is from 2.53 % (for run 3) to 4.15 % (for run 4) percentage (average 3.34 %). The overall average error for the four values of m_w/m_a is 5.25 %. There is no significance in the crossing over of some of the error curves in fig 4. It can be explained by the behavior of the difference $w_{sw} – w$ in the Poppe method in the relation to that of $h_{sw} – h$ in the Merkel method.

A number of calculations regarding cooling tower behavior at affixed value of m_w/m_a can be drawn from Fig 5.8 to Fig. 5.11 as follows:

1. For the same range and approach, the higher the wet bulb temperature the higher the outlet cold water temperature: i.e., T_{wo} (Run 3) < T_{wo} (Run 1) and T_{wo} (Run 5) < T_{wo} (Run 2). (The numbers in parenthesis refer to the appropriate set of conditions in Table 2.)

2. For the same wet- bulb temperature and the larger the approach the smaller the outlet water temperature: i.e., T_{wo} (Run 3) < T_{wo}(Run2), T_{wo}(Run 5) < T_{wo}(Run 4) and T_{wo}(Run 7) < T_{wo}(Run 6)

3. For the same initial and final water temperatures, the lower the inlet air wet bulb temperature the smaller outlet water temperature: i.e., T_{wo} (Run 2) < T_{wo} (Run 1).

A number of calculations regarding cooling tower behavior at affixed value of m_w/m_a can be drawn from fig. as follows:

1. For the same range and approach, the higher wet bulb temperature the smaller outlet water temperature: i.e., T_{wo} (Run 3) < T_{wo} (Run 1) and T_{wo} (Run 5) < T_{wo} (Run 2). (The numbers in parenthesis refer to the appropriate set of conditions in Table 2.)

2 For the same wet- bulb temperature and larger the approach higher the tower outlet cold water temperature: i.e., T_{wo} (Run 3) < T_{wo} (Run2), T_{wo} (Run 5) < T_{wo} (Run 4).

3. For the same initial and final water temperatures, the lower the inlet air wet bulb temperature the higher the cooling tower outlet cold water temperature: i.e., T_{wo} (Run 2) > T_{wo} (Run 1).

5.4. Determination of Evaporation Loss by Poppe Method

In performance analysis problem of cooling tower percentage evaporation loss can be calculated by Poppe method. Percentage evaporation loss of cooling tower plotted here in Figs. 5.13 shows variation in percentage evaporation loss calculated by Poppe method with water to air mass flow rate ratio. Results data of Fig. 5.12 is represented in Table B.14.

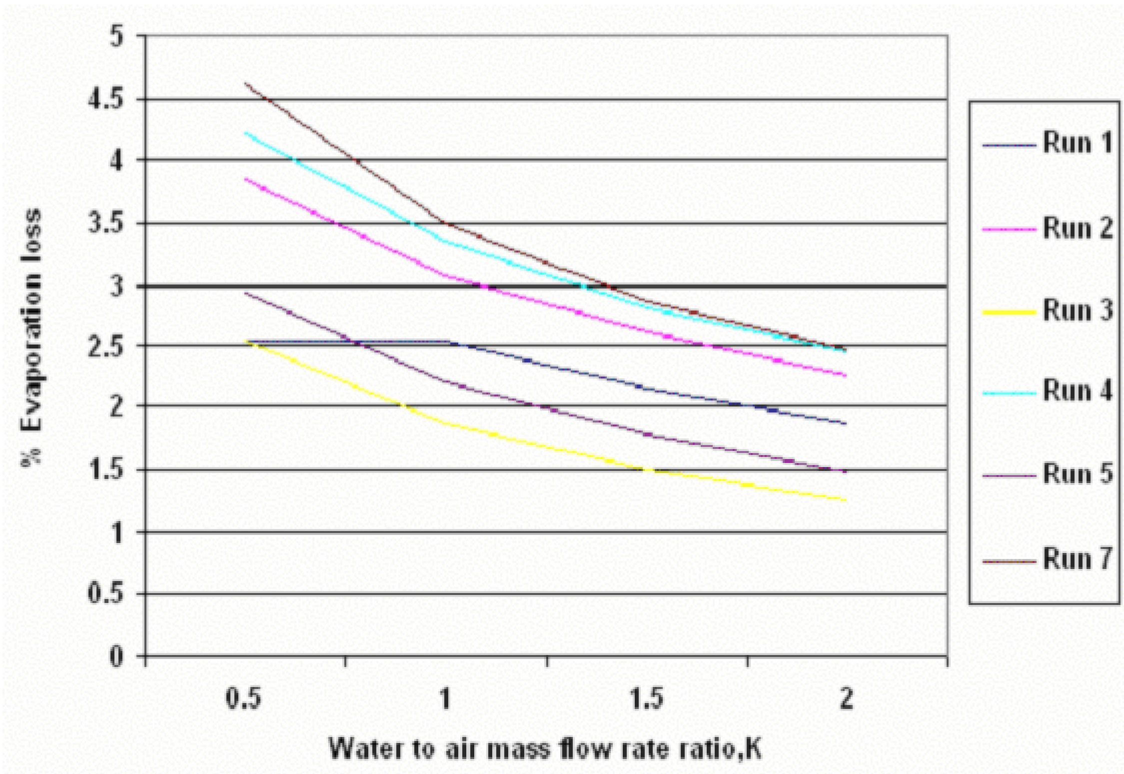


Fig. 5.12 Variation of evaporation loss calculated by Poppe method with water to air mass flow rate ratio

Fig. 5.12 shows that evaporation loss decreases as water to air mass flow rate ratio increases. Percentage evaporation varies from 1.26 % (For Run 3) to 4.62 % (For Run 7) for different initial conditions.

Chapter 6

Application of Model

In this numerical modeling, thermal analysis of cooling tower has been done using conventional Merkel approach and by elaborated method. The elaborated model is due to Poppe, which considers variation of Lewis number and includes evaporation loss. The model is capable of calculating air temperature, humidity ratio cooling tower water outlet temperature and evaporation loss.

The model can be used to predict cooling tower outlet temperature under varying ambient conditions for different inlet water temperatures. This program has been written for performance prediction of natural daft cooling tower which takes cooling tower geometry parameter, water flow and ambient conditions as input and predicts outlet conditions.

The model has been validated from available data for cooling tower. For validation addition has been made in the program to accommodate varying height too. This program applicable to predict outlet condition of cooling tower and designing of cooling tower according to given conditions wherever cooling tower applicable. Some applications of this model follow as: -

6.1 Air Conditioning

The most common cooling tower application is for air conditioning with electric chillers. Typical conditions are 35 ºC inlet water temperature, 29 ºC water outlet temperatures and a water flow rate of 3gpm /ton. Our numerical modeling can design the cooling tower consider optimizing the chiller/ cooling tower combination. This is especially beneficial where the design wet bulb temperature is less than 26 ºC. In Denver, for example, the design wet bulb is typically 20 ºC and the cooling tower selected for 35 in and 29 out will be very small. It is clearly advantageous to reduce the condenser water temperature to

32/26, 31/25, 29/23, etc.. The tower will grow in size but the chiller- more importantly its compressor horsepower- will decline making a more economical selection. Operational savings can be substantial.

The system designer should try to get the cooling water temperature as high as practical for the most economical cooling tower selection. The greater this 'approach' the more economical the tower selection. This numerical modeling helpful to design cooling tower for economical benefits.

6.2 Hot Well/Cold Well

Certain industrial applications are cyclical in nature imposing substantial loads for short periods of time. Here, the amount of water in the system becomes important. A closely coupled system with minimal water volume will have rapidly varying temperatures while a large water volume will smooth out the temperature spikes.

Other systems requiring intermittent process water flow are inconsistent with the cooling tower's desire for a constant flow while in operation. Still others have process water temperatures that are too high for direct introduction into a conventional cooling tower. In such case this numerical modeling helpful to design cooling tower to fulfill needs.

6.3 Design and Performance Analysis in Power plant

This numerical modeling of different methods is helpful to design and predicts performance of cooling tower. If experimental outlet conditions are differ from , outlet condition of cooling tower predicted by numerical modeling , it means some thing is wrong in cooling tower whether in design, or any part of cooling tower is not performing well or scale formation takes place. This numerical modeling can design cooling tower for different whether and for different fill design. Design problem for Dadri plant given below for validation of results found by numerical modeling.

Table.6.1 Dadri plant data for input to the program [Technical detail handbook NTPC]

Initial Conditions	Initial Values
Hot water temperature	43 ºC
m_w/m_a	1.35
Merkel number available (Packing function)	1.366
Wet bulb temperature	27 ºC
Approach	5 ºC
Range	11 ºC
Dry bulb temperature	35 ºC

Table.6.2 Results Output of designing problem of cooling tower for Dadri data plant

	Outlet cold water temperature (ºC)	% Evaporation Loss
Merkel method	31.91	-
Poppe method	32.9	1.70
NTPC handbook data	32	-

Sutherland [1] has shown graphical representation of variation of Merkel number with water to air mass flow rate ratio and predicted results for different approach conditions. For modification of results , graphical representation of variation of Merkel number with water to air mass flow rate, calculated by Merkel and Poppe method shown below :-

Table.6.3 Input Data from Sutherland [1] for numerical modeling.

Run	t_{wb}	t_d	t_h	t_c	A $=(t_c - t_{wb})$	R $=(t_h - t_c)$
1.	30	35	50	40	10	10
2.	20	25	50	40	20	10
3.	10	15	50	40	30	10

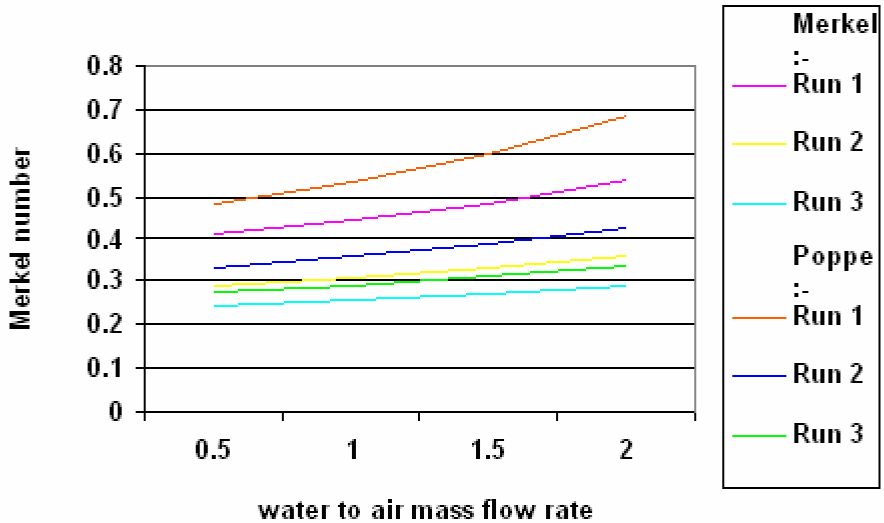


Fig. 6.1 Variation in Merkel number with water flow rate ratio for different Approach calculated by Merkel and Poppe methods

Results predicted by Merkel and Poppe methods are shown in Fig.6.1 in graphical form. Results data are shown in Table B.15 and results are validated by Sutherland [1].Fig. 6.1 shows that smaller the approach, higher the Merkel number (Tower volume). Fig shows that for cooling the water near to the wet bulb temperature higher tower volume is required.

Chapter 7

Conclusion

Based upon the results and discussion in chapter 5 following conclusion can be draw:

1. The most commonly used models available in the literatures are Merkel, e-NTU and Poppe methods.

2. Computer programs for sizing and performance analysis of wet cooling tower prepared. The output of the programs matches satisfactorily with the available data in the literatures.

3. The comparison of the enthalpy diagrams between the Merkel and Poppe methods shows that i_{masw} curves of the two methods fall on the top of each other. There is a small discrepancy in the i_{ma} curves of these two different methods, especially at the hot water side. Poppe method predicts an approximately linear variation of the air enthalpy but the gradient is different from that predicted by Merkel method. The $1/(i_{masw} - i_{ma})$ curve of Poppe method lies above the $1/(i_{masw} - i_{ma})$ curve of the Merkel method. As the transfer characteristic, or Merkel number, is a function of area under the $1/(i_{masw}-i_{ma})$ curve, the Merkel number predicted by the Merkel method. It is therefore very important that the same method of method (i.e. Merkel, Poppe or e-NTU) be employed in the fill performance test and subsequent cooling tower performance method.

4. Percentage error in fill volume calculated by Merkel and e-NTU method varies from 0.3 to 3.3 percentages. Merkel number calculated by e-NTU method is approximately 1.6% less than calculated by Merkel method. This error is insignificant because both Merkel and e-NTU method work on same simplifying assumptions.

5. Counter flow cooling tower can be significantly undersized if the Merkel method is used. For an extensive series of runs of Sutherland data [1] for Merkel and Poppe method, errors in tower volume of up to 27 percentage were obtained with the average value being approximately 16.3 percentage. Similarly performance analysis of cooling tower error in outlet water temperature calculated by both Merkel and Poppe method is up to 7.6 percentages were obtained with the average value being approximately 5.25 percentages.

   This gives valuable information to designer that this error percentage in outlet water temperature of cooling tower is very less it means both Merkel and Poppe methods predicts outlet temperature of tower very close to each other. Although Poppe predicts higher temperature than Merkel method yet its amplitude is very less. So it is suggest to designer to use Merkel method.

6. When designer emphasis on calculation of evaporation loss in cooling tower flow, Poppe method will be the right choice. Because both e-NTU and Merkel method work on simplifying assumptions and do not count evaporation loss during flow through cooling tower. Percentage evaporation loss calculated by Poppe method varies from 1.261 to 4.621.

7. This work gives some valuable results for designing the cooling tower that is very helpful for designer. For fixed heat load, range and approach tower size varies inversely with wet-bulb temperature, reducing the design wet-bulb temperature increases the size of the tower. This is because most of the heat transfer in a cooling tower occurs by virtue of evaporation and air's ability to absorb moisture reduces with temperature.

8. Tower size varies inversely with approach. A longer approach requires a smaller tower. Conversely, a smaller approach requires an increasingly larger tower. This work also gives the information of variation of cooling tower outlet temperature with

different operating condition which is helpful to predict the cooling tower performance.

Validation of numerical modeling of Merkel, e-NTU and Poppe method has been done with cooling tower data available from Dadri power plant of NTPC, Sutherland [1] data for different operating condition , Jaber and Webb [3] data and Li and Priddy's hand book data [2] for different operating conditions for both design and performance analysis of cooling tower.

Chapter 8
Scope of Future Work

Performance prediction of cooling tower with non-uniform water distribution by numerical modeling and simulation of cooling tower performance with non-uniform water distribution for force draught cooling tower is still to be done. This can be done either by writing codes in programming or by using soft wares CFD and CFX etc.

Future researcher can do simulation of cooling tower performance and designing with uniform and non-uniform water distribution for both natural and force draught cooling tower which is still left and further programming of this model will make easies of this application. One can predict the formulation of Poppe, Merkel and e-NTU method, which is applicable for cross flow cooling tower. Modeling and simulation will be further step in this area.

References

[1]. Sutherland, J. W., 1983, *‘‘Analysis of Mechanical-Draught Counter flow Air/ Water Cooling Towers,’’* ASME J. Heat Transfer, 105, Pp. 576–583.

[2]. Kam W. Li and A. Paul Priddy, 1985, “*Hanbook of Power Plant System Design*,” John Wiley & Sons, Canada.

[3]. Jaber, H., and Webb, R. L., 1989, *‘‘Design Of Cooling Towers By The Effectiveness -NTU Method,’’* ASME J. Heat Transfer, 111, Pp. 837–843.

[4]. M. Poppe, H. Rogener, Derechnung Von Rockkohlwerken, VDI- War Meatlas (1991) Mi1 – Mi15.

[5]. J C Kloppers, D.G Kroger, “*Cooling Tower Performance Evaluation-Merkel,Poppe and NTU Methods Of Analysis,”* Trans, ASME:J Engg. Gas Turbines Power, In Press.

[6]. J.C.Kloppers, And D.G. Kroger, 2004 *‘‘A Critical Investigation into the Heat and Mass Transfer Analysis of Counter Flow Wet Cooling Towers,”* Science Direct, International Journal Heat and Mass Transfer 48, (2005), P.P 765 – 777

[7]. Kroger, D. G., 1998, “*Air-Cooled Heat Exchangers and Cooling Towers Thermal-Flow Performance, Evaluation and Design*,” Pennwell Corp., Tulsa, Ok.

[8]. A.K.M Muhiuddin , K. Kant, “*Knowledge Base For The Systematic Design of Wet Cooling Towers, Part I: Selection And Cooling Tower Characteristics”*, Int. J.

[9]. C. S. Mishra M-Tech thesis on *“Simulation of Cooling Tower Performance with Non-Uniform Water Distribution*”, May 2005, Indian Institute of Technology, Delhi.

[10]. Webb, R.L., 1988, *“A Critical Review of Cooling Towers Design Methods”* In: Heat Transfer Equipments Design, R.K. Shah, E.C.Subbarao, and P.A. Mashelkar. Eds., Hemisphere Pub. Corp., Washington, DC, PP. 547-558, Refrigeration 19(1) (1996) 43-51

[11]. Milosavljevic Nenad, Heikkila Pertti. *“A Comprehensive Approach to Cooling Tower Design.”* Applied Thermal Engineering 21 (2001) PP. 899 -915.

[12]. Villiers Adrianj. De, Bosman B Peter, Knight Piesold Energy South Africa. *"Enhancing Tower Performance Using Non- Uniform Water Distribution."* PP. TPN96-15. Cooling Tower Institute Annual Conference Boston, Texas – February 1996.

[13]. Fisenko S.P., Petruchik A.I., Solodukhin A.D., *"Evaporative Cooling Of Water in A Mechanical Draft Cooling Tower"*, International Journal of the Heat and Mass Transfer 47 (2004) PP. 165-177.

[14]. Hawlader and Liu M.N.A., Liu B.M., "*Numerical Study of the Thermal Hydraulic Performance of Evaporative Natural Draft Cooling Towers,"* Applied Thermal Engineering 22 (2002) 41-59.

[15]. Bedekar S.V., Nithiarasu P., Seetharamu K.N., "*Experimental Investigation of The Performance of a Counter Flow, Packed Bed Mechanical Cooling Tower,"* Energy Vol.23, No.11, PP. 943-947, 1998.

[16]. N. Makkinejad., *"Temperature Profile in Countercurrent/Cocurrent Spray Towers.*" International Journal of Heat and Mass Transfer, 44 (2001) 429 – 442.

[17]. Goel Yashvir Singh, *"Computer Aided Modeling of the Heat Rejection System of Thermal Power Plant",* P.H.D, Thesis 1993, Indian Institute Of Technology, New Delhi India.

[18]. Marley Cooling Technologies, 2004, *''Cooling Tower Performance Basic Theory nd Practice", Overland Park KS USA.*

Appendix (A)

Thermo Physical Properties

The thermo physical properties summarized here are presented in Kroger [6]. Refer to Kroger [6] for the ranges of applicability of the following equations of the thermo physical properties. All the temperature is expressed in Kelvin.

The enthalpy of the air water mixtures given by

$$i_{ma} = C_{pa}\,(T-273.15) + w\,(I_{fgwo}+C_{pv}\,(T-273.15)) \qquad \text{kJ/kg dry air} \qquad (A.1)$$

Where the specific heats, C_{pa} and C_{pv}, are evaluated at (T+273.15)/2 by Eqs. (A.2) and (A.4) respectively. The latent heat i_{fgwo} is evaluated at 273.15 K according to Eqs. (A.8).

The specific heat of the dry air given by

$$C_{pa} = 1.045356 - 3.161783 \times 10^{-4}\, T + 7.083814 \times 10^{-7}\, T^2 - 2.705209 \times 10^{-10}\, T^3 \quad \text{kJ/kgK} \qquad (A.2)$$

The vapor pressure of the saturated water vapor is given by

$$P_v = 10^Z \quad \text{N/m}^2 \qquad (A.3)$$

Where

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

The specific heat of saturated water vapor is given by

$$C_{pv} = 1.3605 + 2.31334 x 10^{-3} T - 2.46784 x 10^{-13} T^5 + 5.91332 x 10^{-16} T^6 \quad k J/kg.K \tag{A.4}$$

The specific heat of the mixture of air and water vapor is given by

$$C_{pma} = (C_{pa} + w C_{pv}) \quad kJ/K. kg dry air \tag{A.5}$$

The humidity ratio is given by

$$w = ((2501.6 - 2.3263(T_{wb} - 273.15)/ (2501.6 + 1.8577(T - 273.15) - 4.184(T_{wb} - 273.15)))) (0.62509P_{vwb}/ (P_a - 1.005 P_{vwb})) x ((1.00416(T - T_{wb}))/ (2501.6 + 1.8577(T - 273.1) - 4.184 (T_{wb} - 273.15))) \tag{A.6}$$

Where P_{vwb} is the vapor pressure from Eqs. (A.3) evaluated at the wet bulb temperature. The specific heat of the water is given by

$$C_{pw} = 8.15599 -2.80627 x 10^{-2} T +5.11283 x 10^{-5} T^2 -2.17582 x 10^{-16} T^6 \quad kJ/kgK \tag{A.7}$$

The latent heat of water is given by

$$I_{fgw} = 3.4831814 x 10^3 -5.862703T +12.139568 X 10^{-3} T^2-1.40290431 X 10^{-5} T^3 \quad k J/kg \tag{A.8}$$

I_{fgwo} is obtained from Eqs. (A.8) where T = 273.15

Appendix (B)

Results data of graphical representation shown in previous part of this thesis, given below in table form.

Table B.1. Results of Li and Priddy's handbook [2] problem solved by Poppe method

t_d (ºC)	t_c (°C)	i_{masw} (kJ/kg)	i_{ma} (kJ/kg)	$I = 1/(i_{masw} - i_{ma})$, (kg/kJ)
32.00	28.88	94.12	58.53	0.0284
30.83	30.32	101.52	67.26	0.0291
30.28	31.77	109.47	75.61	0.0295
30.20	33.21	117.91	83.96	0.0294
30.47	34.66	126.98	92.34	0.0288
31.00	36.10	136.63	100.74	0.0278
31.69	37.55	147.03	109.17	0.0264
33.13	38.99	158.09	117.67	0.0247
34.50	40.44	170.03	126.21	0.0228
35.80	41.88	182.75	134.8	0.0208
37.02	43.33	196.51	143.44	0.0188

Table B.2 Results of Li and Priddy's handbook [2] problem solved by Merkel method

t_c (ºC)	i_{masw} (kJ/kg)	i_{ma} (kJ/kg)	$I=1/(i_{masw} - i_{ma})$ (kg/kJ)
28.88	94.12	59.3	0.0287
30.32	101.52	67.13	0.0290
31.77	109.47	74.96	0.0289
33.21	117.91	82.79	0.0284
34.66	126.98	90.62	0.0275
36.10	136.63	98.45	0.0261
37.55	147.03	106.28	0.0245
38.99	158.09	114.11	0.0227
40.44	170.03	121.94	0.0207
41.88	182.75	129.71	0.0188
43.33	196.51	137.6	0.0169

Table B.3 Variation of Merkel number calculated by Merkel method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Run 1	0.4054	0.4404	0.4843	0.5414
Run 2	0.2888	0.3073	0.3293	0.3558
Run 3	0.6791	0.7894	0.9617	1.2887
Run 4	0.2426	0.256	0.2713	0.2894
Run 5	0.4685	0.5216	0.594	0.7009
Run 8	0.5965	0.6611	0.7455	0.8617

Table B.4 Variation in Merkel number calculated by e-NTU method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Run 1	0.4139	0.4503	0.496	0.5558
Run 2	0.2879	0.3064	0.3283	0.3549
Run 3	0.6896	0.8034	0.9828	1.334
Run 4	0.2401	0.2532	0.2684	0.2862
Run 5	0.466	0.5189	0.5911	0.698
Run 8	0.6099	0.6715	0.7504	0.8562

Table B.5 Variation in percentage error calculated by Merkel and e-NTU method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Run 1	2.05	2.2	2.4	2.6
Run 2	0.3	0.4	0.4	0.6
Run 3	1.5	1.7	2.1	3.3
Run 4	1	1.1	1.2	1.4
Run 5	0.5	0.5	0.6	0.9
Run 6	0.3	0.3	0.4	0.5
Run 7	1.5	1.7	2.1	3.3
Run 8	1.1	1.4	1.7	2.1

Table B.6 Variation in Merkel number calculated by Poppe method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Run 1	0.4835	0.5327	0.5968	0.6847
Run 2	0.3304	0.355	0.3848	0.422
Run 3	0.7994	0.9485	1.199	1.758
Run 4	0.2728	0.29	0.3104	0.3349
Run 5	0.5328	0.6005	0.6962	0.8458
Run 8	0.7046	0.7969	0.9242	1.1115

Table B.7 Variation in tower volume error calculated by Merkel and Poppe method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Run 1	16.2	16.5	18.8	20.9
Run 2	12.5	13.5	14.4	15.8
Run 3	15	16.8	19.8	26.7
Run 4	11	11.7	12.5	13.6
Run 5	12.01	13.1	14.6	17.1
Run 6	12.7	13.5	14.5	15.6
Run 7	15.17	17.12	21.5	27
Run 8	15.3	17	19.3	22.4

Table B.8 Variation in tower volume with wet bulb temperature with same range and approach.

t_{wb} →	10°C	20°C	30°C	40°C
$\mathbf{V_M}$	-	13.58	8.1	-
$\mathbf{V_P}$	-	15.98	9.67	-

Table B.9 Variation in tower volume with different approaches for same wet bulb temperature.

A →	10°C	20°C	30°C	40°C
V_M	13.58	5.77	-	-
V_P	15.98	6.6	-	-

Table B.10 Variation in tower outlet temperature calculated by Merkel method with water to air mass flow rate.

L/G → Run ↓	0.5	1	1.5	2
Run 1	31.42	33.25	35.47	37.23
Run 2	23.69	28.35	31.6	33.9
Run 3	22.93	26.61	29.04	30.78
Run 4	18.48	24.85	28.91	31.8
Run 5	17.48	22.9	26.18	28.4
Run 6	18.48	24.85	28.91	31.8
Run 7	23.69	28.35	31.6	34.02

Table B.11 Variation in tower water outlet temperature calculated by Poppe method with water to air mass flow rate ratio.

L/G → Run ↓	**0.5**	**1**	**1.5**	**2**
Run 1	32.63	34.53	36.73	38.43
Run 2	25.03	29.77	32.98	35.31
Run 3	24.04	27.67	29.99	31.58
Run 4	20	26.4	30.39	33.18
Run 5	18.75	24.07	27.21	29.3
Run 6	25.06	29.78	32.99	35.31
Run 7	24.07	27.69	30	31.59

Table B.12 Variation in tower water outlet temperature calculated by Merkel and Poppe method with water to air mass flow rate ratio.

L/G →	0.5	1	1.5	2
t_{CP}	24.04	27.67	29.99	31.58
t_{CM}	22.93	26.61	29.04	30.78

Table B.13 Variation in percentage error in tower water outlet temperature with water to air mass flow rate ratio for Merkel and Poppe method.

L/G → Run ↓	0.5	1	1.5	2
Run 1	3.708	3.706	3.43	3.12
Run 2	5.35	4.76	4.18	3.98
Run 3	4.61	3.83	3.16	2.53
Run 4	7.6	5.87	4.87	4.15
Run 5	6.77	4.86	3.46	3.07
Run 6	5.46	4.8	4.21	3.64

Table B.14 Variation in evaporation loss calculated by Poppe method with water to air mass flow rate ratio

L/G → Run ↓	0.5	1	1.5	2
Run 1	2.5283	2.528	2.145	1.863
Run 2	3.855	3.081	2.595	2.246
Run 3	2.519	1.87	1.504	1.261
Run 4	4.211	3.359	2.827	2.447
Run 5	2.948	2.194	1.769	1.486
Run 7	4.621	3.498	2.872	2.454

Table B.15 Variation in Merkel number for different approach condition calculated by Merkel and Poppe method with water to air mass flow rate ratio.

L/G → Run ↓	0.5	1	1.5	2
Merkel				
Run 1	0.4054	0.4404	0.4843	0.5414
Run 2	0.2888	0.3073	0.3293	0.3558
Run 3	0.2426	0.256	0.2713	0.2894
Poppe				
Run 1	0.4835	0.5327	0.5968	0.6847
Run 2	0.3304	0.355	0.3848	0.422
Run 3	0.2728	0.29	0.3104	0.3349

Appendix (C)

Fills or Packs

Cooling tower fills or packs have been developed from the simple timber or bamboo splash bar to the modern vacuum formed or injection molded plastic fills now in common use according to Mirsky. Fills should be structurally strong, chemically inactive, fire resistant, resistant to fouling and erosion, and have a low airflow resistance. A critical comparative study is presented by Monjoie. His main objective is to identify the deficiencies of seven different plastic fill materials. Among others, properties concerning forming, assembly, fire, chemical, thermal, recycling, and environmental impact are evaluated. Examples of some plastic fills are shown in Figure C.1.

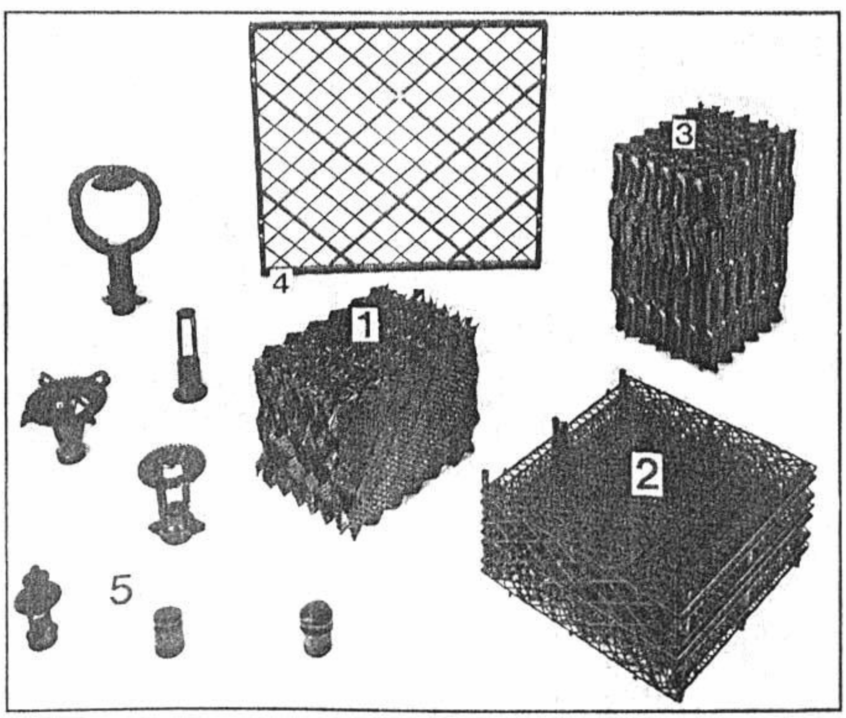


Fig. C.1 Plastic Fills and Spray Nozzels (1) Film (2) Tric Grid (3) Film (4) Splash (5) Spray Nozzels

Further examples of fills are shown in Figure C.2 from Lowe and Figure C.3 from Johnson. Many other configurations find application in practice according to Dumitru. PVC can be used as fill material up to a water temperature of about 50 °C, chlorinated PVC (CPVC) up to 65 °C, while Burger finds higher temperatures require polypropylene or stainless steel.

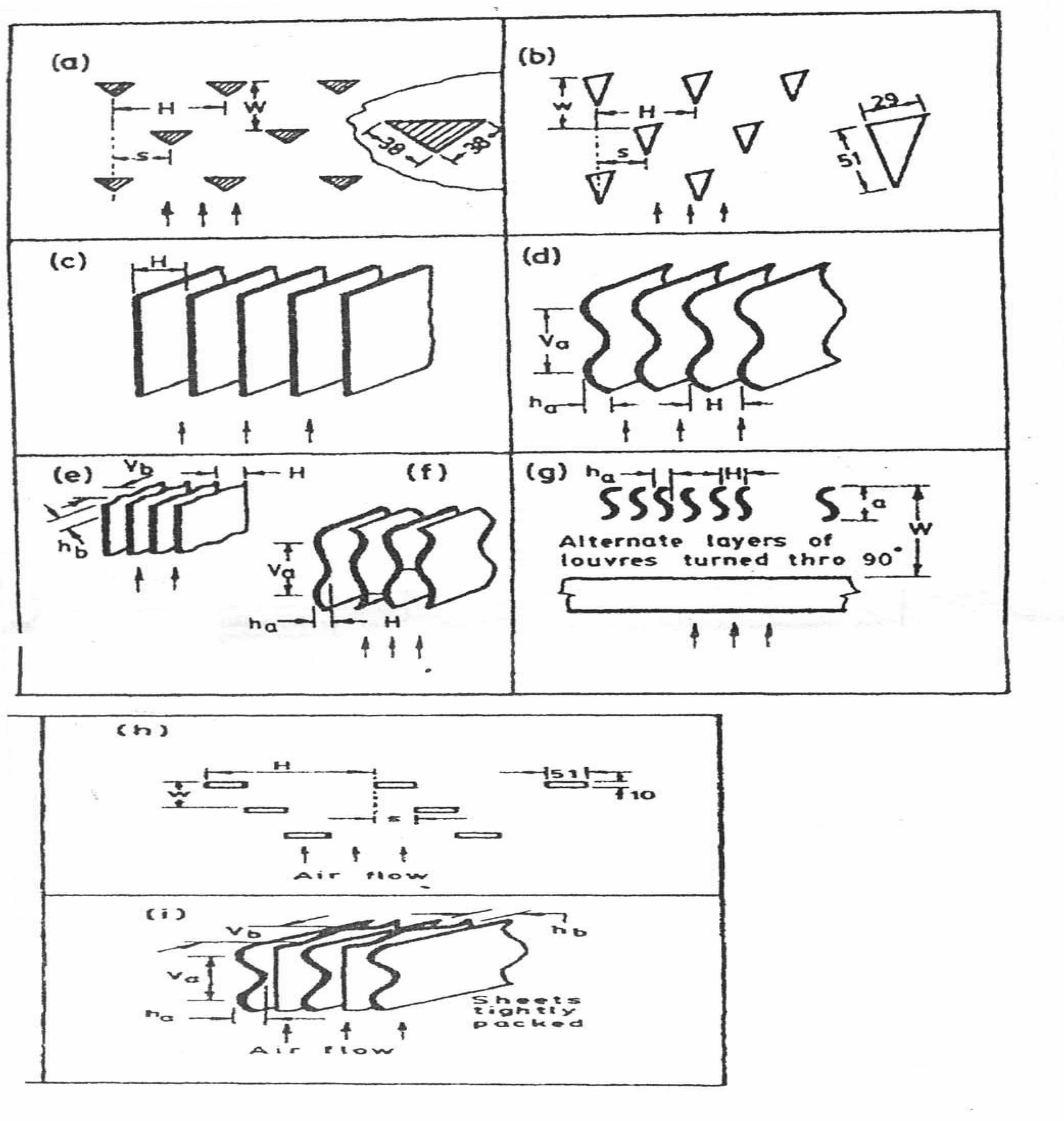


Fig. C.2 Fills (a) (b) and (h) Splash, (c) (d) (e) (f) (g) and (i) Film

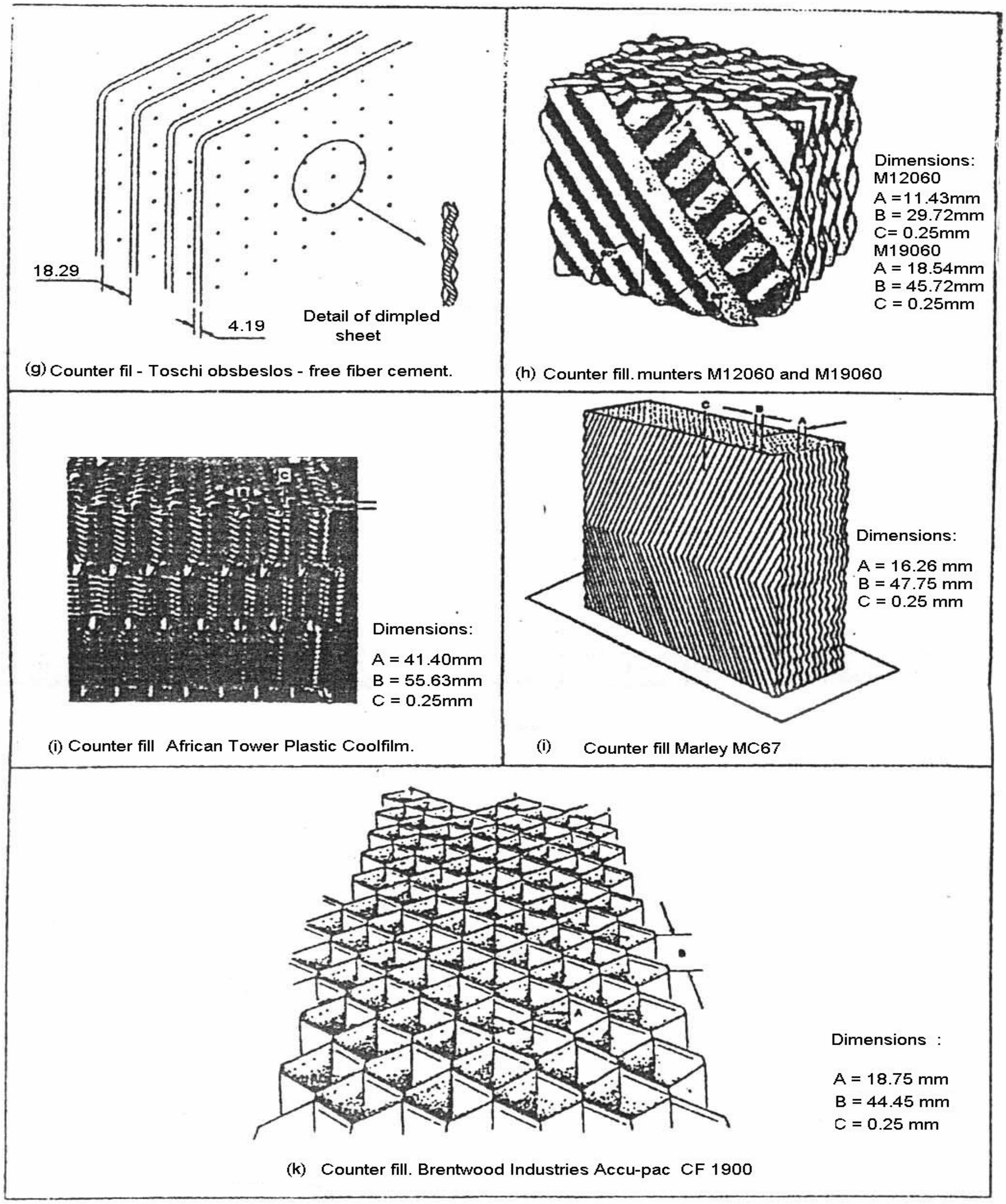


Fig. C.3 (b) Fills

In some applications, fill materials that can be recycled may be preferred. Many of the deficiencies of plastic fills may be avoided by employing stainless steel fills, an example is shown in Figure C.4.

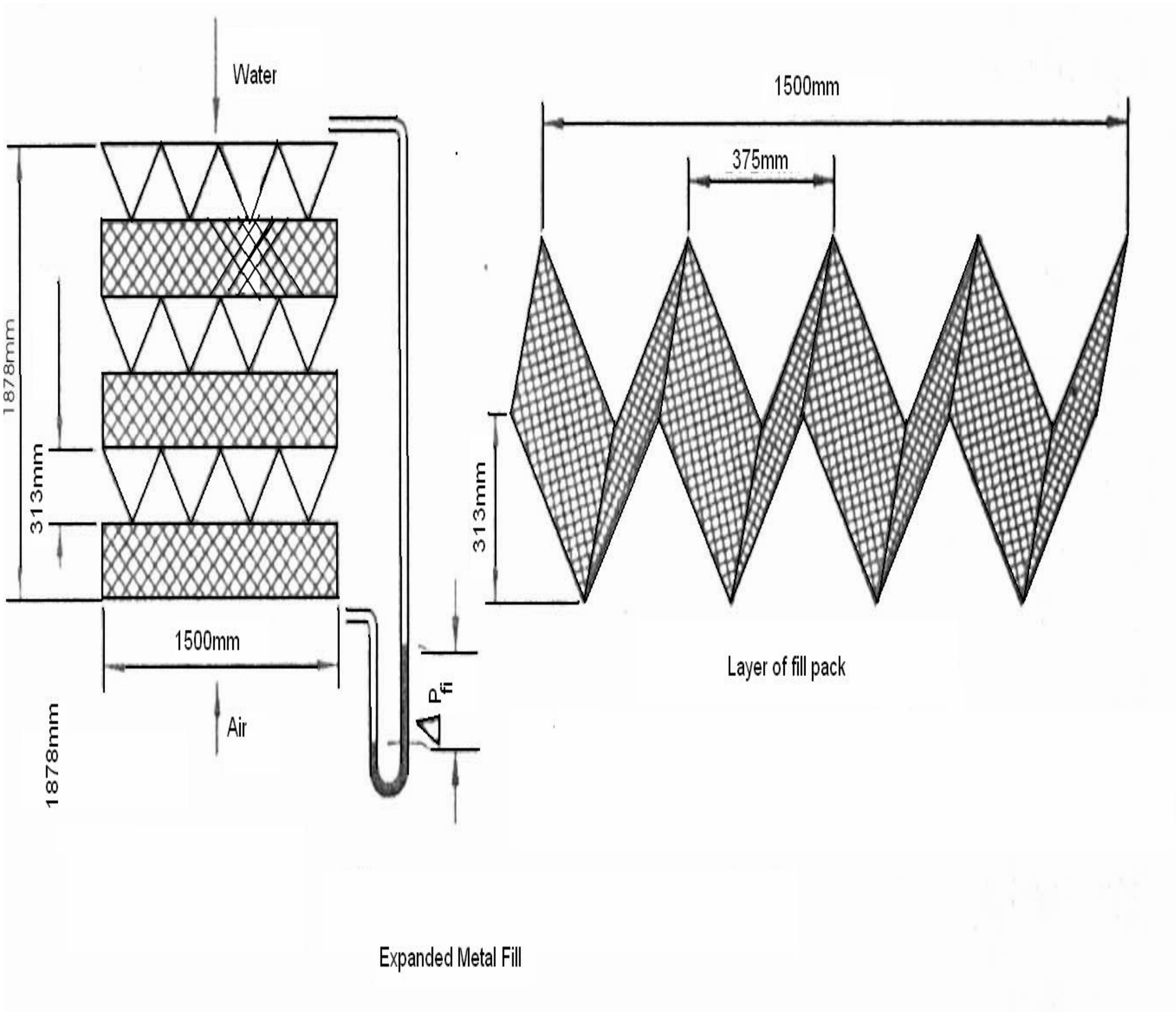


Fig. C.4 Expanded Metal Fill

The water distribution system for a counter flow wet-cooling tower usually consists of a piping manifold that serves to support and supply an array of low pressure spray nozzles. A few examples are shown in Figure C.1. In large natural draft cooling towers, nozzles require relatively low pressures of between 5000 to 15,000 N/m2. Medium pressure nozzles requiring 20,000 to 100,000 N/m^2 and producing smaller droplets are employed in mechanical draft industrial cooling towers according to Thacker. A most comprehensive source of nozzle information is presented by Lefebvre. Nozzles should be arranged so the distribution of water entering the fill is as uniform as possible. Kranc found non-uniformity of flow occurred where nozzles produced circular spray patterns with radial variation overlap patterns of the sprays from adjacent nozzles. He also found non-uniform flow of water may be corrected partially in certain fills.

Improved distribution may also be achieved by employing nozzles that give almost square spray patterns. In a counter flow cooling tower, the entire cooling region including the spray, fill, and rain zones are influenced by the characteristics of the droplets introduced through the spray nozzles. Up to 15% of the cooling may occur in the spray zone above the fill. The spray may be directed downward or upward. In the case of the latter, the longer droplet residence time improves the transfer process in the spray zone. The spray produced in a cooling tower depends on the type of nozzle employed according to both Scriven and Bellagamba. The lightest drops (less than about 0.3 mm in diameter) are carried upward by the air to the droplet eliminators where most are collected and returned downward to the fill in the form of larger drops.

Splash fill

The splash type fill or pack is designed to break the mass of water falling through the cooling tower into a large number of drops. The water surface area exposed to cooling air increases as well as the amount of heat transferred to the surrounding air by conduction, convection, radiation, and evaporation. As water falls through the fill, droplets collide with successive layers of splash bars, which cause redistribution of water and heat due to the formation of fresh droplets.

As a further benefit, the retention time of water falling through the tower is prolonged by contact with the fill, extending the period during which the water is exposed to cooling air. The disadvantage of splash fill is, by its very nature, a large volume is required to break up the water flow, which in turn necessitates large towers. There is a natural tendency for free falling droplets to agglomerate.

The effectiveness of a particular splash type fill is governed by its ability to form droplets. Its efficiency also depends upon its airside flow resistance and, to a lesser extent, on economical use of material. Treated timber is used due to its availability, structural strength, relative cheapness, and long working life under most conditions. Injection molded plastic fills have been in service for many years.

Splash fills tend to produce more carryover than other types of fill, particularly if high air rates are used. Efficient spray eliminators are employed to overcome this potential disadvantage. However, the increased air resistance of the complete tower demands additional draft or fan capacity and additional running costs.

The inherent disadvantages of splash fills have created a demand for the development of film type fills, which are more compact and preferred. The latter require less material and water pumping power due to the lower fill height.

Trickle pack

Trickle packs or grids are much finer than splash packs and are made up of plastic or metal grids onto which the water is sprayed. It runs down the grid rather than splashing. This type of fill has been introduced in recent years with the advances in plastic injection molding. Because of the much finer mesh than the splash type fill, they tend to clog more easily and have a greater pressure drop.

Film fill

Although the purpose is to produce a large water surface area, film fills are different from splash fills because this is achieved by allowing the water to spread in a thin layer over a large area of fill rather than forming droplets. This reduces the problem of carryover of water droplets into the atmosphere and allows higher air velocities to be used.

Fill types may be placed in several categories, the simplest is the timber grid. This consists of a series of closely spaced slats placed in tightly packed layers, each layer at right angles to the previous layer. This arrangement provides good water distribution over a large area, but air resistance is high. Thin timber sections have low structural strength and limited resistance to chemical attack and distortion. Corrugated or flat asbestos sheeting was used in the past, but resin impregnated cardboard, metal, and more effective plastics are preferred now.

A theoretical examination by Kelly suggests a fill consisting of a series of close parallel vertical film surfaces would give good transfer with low pressure drop. Various manufacturers have designed packs along these lines but found in practice they were not reliable. The packs have a tendency towards uneven water distribution, which reduces heat transfer effectiveness. Fouling is more of a problem in this fill than in the splash type.

Extended film fills

Although problems were encountered with early thermoplastic film type fills, these difficulties have been largely overcome. There are now a wide variety of pressed or vacuum formed fills available. These vary in design but have high transfer characteristics, low weight, acceptable strength, and adequate durability. The problems of water and air distribution have been reduced by the development of geometrical designs, which incorporate interconnected channels and secondary profiles. Both these features improve water and air distribution and encourage greater mixing of the layer of saturated air. The air layer forms adjacent to both the water layer and the bulk of air traveling through the fill, and it further improves performance.

A large variety of proprietary cooling tower fills are produced by commercial manufacturers. The results of studies on the performance characteristics of fills have been reported in the literature by authors such as Lowe, the Cooling Tower Institute (CTI), Kelly, Cale, Fulkerson, and Johnson. Some of the more recent studies reported by Johnson have evaluated test facilities and methods of data evaluation. Thermal and pressure drop data obtained in one facility does not always agree with that obtained in another facility. Some of the reasons for these discrepancies are:

- Distorted flow patterns.
- Test facility edge effects.
- Influences due to the type of spray nozzle and spray or rain zone according to Fulkerson.
- Different test temperatures and pressures.
- Changes in fill wetting patterns (the degree of wetting of the fill surface may change with time).
- Errors in measurement.

In a large modem test facility, the cross section of the fill may have dimensions up to 7 m x 7 m for counter flow and 5 m x 10 m for cross flow and most of the previously mentioned problems can be greatly reduced according to Fabre and Caytan. It is important to elaborate on the method employed in evaluating the test data, i.e., Merkel, Poppe, or others. Where the Merkel method is employed, it is convenient to present the transfer and pressure drop characteristics per meter of fill depth as follows:

$$h_d\, a_{ii}\, A_{fr} / m_w = h_d a_{fi} / \mathrm{G}_w = a_d (G_w / \mathrm{G}a)^{-b\ d} \qquad \text{(C.1)}$$

and

$$K_{fi1} = a_p (G_w / G_a) + b_p \qquad \text{(C.2)}$$

Where mean mass flow rates through the fill are given by $G_w = m_w/A_{fr}$ and $G_a = m_a/A_{fr.}$ The subscript 1 refers to one meter height of the fill or air travel distance. In fill performance characteristics presented in the literature, some correlations may be expressed in terms of G_{av} based on inlet or mean conditions through the fill. If not clearly defined, this may lead to errors, especially when evaluating the draft equation in the case of a natural draft wet-cooling tower.

Other forms for approximating the previous characteristics empirically are as follows

$$h_d a_{fi} / G_w = a_d \; G_w^{bd} G_a^{cd} \qquad \text{(C.3)}$$

and

$$h_d a_{fi} / G_w = a_d \; (G_w / G_a)^{-bd} L_{fi}^{cd} \qquad \text{(C.4)}$$

or

$$Kfi1 = ap \; G_w^{b_{pa}} G_a^{b_{pb}} \qquad \text{(C.5)}$$

Where the values of $a_{d'}$ $a_{p'}$ $b_{d'}$ and b_p are determined experimentally in each case.

Obviously, these simple correlations cannot consider all variables, resulting in considerable scatter of test data which may lead to less reliable cooling system designs. Ideally, fill performance tests for a particular cooling tower should be conducted under conditions similar to those specified for the tower design operating point according to Kloppers.

The publication includes:

- Complete data listing of the calculated mass transfer coefficients.
- Resulting correlations and confidence limits associated with the correlations.
- Statistical summary of the capability of the codes to predict large scale cooling. tower performance.

Some of the results are shown in Tables C.2 (a) and C.2 (b) taken from Johnson. Dreyer presents a mathematical model to predict the performance characteristics of splash fill material and lists extensive experimental performance data.

The performance characteristics of a few fills are listed in Table C.1 from Lowe.

Table C.1 Data for Counter flow Fills (Merkel's Theory)

Mass transfer per meter of fill of ATD, $^{h}{}_{d}\,{}^{a}{}_{fi}/G_w = {}^{a}{}_{d}\,(G_w/G_a)^{-b}{}_{d}$ **Loss coefficient per meter of fill height or ATD, $K_{fi1} = {}^{a}{}_{p}\,(G_w/G_a) + {}^{b}{}_{p}$**										
Fill type	Description	Fig. no. 4.3.2	Dimensions				Mass Transfer		Pressure	
			a_a m	P_a m	P_t M	P_1 m	a_d	^{b}d	^{a}P	^{b}P
1.	Triangular splash bar	a	Staggered		0.1524	0.2286	0.2950	0.50	2.62	5.00
2.	Triangular splash bar	a	Staggered		0.1524	0.1524	0.3084	0.50	2.73	9.15
3.	Triangular splash bar	a	Staggered		0.1524	Altern 0.1270 0.3302	0.3150	0.45	1.57	4.5
4.	Triangular splash bar	a	Staggered		0.1524	0.3048	0.246	0.42	1.89	3.0
5.	Triangular splash bar	a	Staggered		0.1143	0.4572	0.236	0.47	2.16	3.75
6.	Flat asbestos sheets	c			0.0444		0.2887	0.70	0.725	1.37
7.	Flat asbestos sheets	c			0.0381		0.361	0.72	0.936	1.30
8.	Flat asbestos sheets	c			0.0318		0.394	0.76	0.77	1.70
9.	Flat asbestos sheets	c			0.0254		0.459	0.73	0.89	1.70
10.	Triangular splash bar (Bar upside down)	a	Staggered		0.1524	0.2286	0.276	0.49	4.15	6.35
11.	Corrugated asbestos sheets	d	0.054	0.1461	0.4450		0.69	0.69	1.93	7.80
12.	Corrugated asbestos sheets	d	0.054	0.1461	0.3175		0.72	0.61	3.61	8.10
13.	Corrugated asbestos sheets	d	0.054	0.1461	0.0572		0.59	0.68	1.39	1.50
14.	Corrugated asbestos sheets	e	^{a}b = 0.54	P_b = 0.1461	0.0445		0.36	066	1.93	0.44
15.	Corrugated asbestos sheets	f	0.054	0.1461	0.0254		0.56	0.58	1.74	12.4
16.	Triangular splash bar	b	In line		0.1016	0.2032	0.24	0.52	2.51	0.35
17.	Triangular splash bar	b	Staggered		0.1016	0.2032	0.29	0.55	2.18	1.55
18.	Triangular splash bar	b	Staggered		0.1016	0.2540	0.26	0.58	1.69	1.45
19.	Triangular splash bar	b	In line		0.1016	0.2540	0.24	0.54	1.61	1.45
20.	Triangular splash bar	b	Staggered		0.1016	0.1950	0.31	0.53	2.35	1.50
21.	Triangular splash bar	b	Staggered		0.1016	0.1524	0.32	0.54	2.32	2.80
22.	Triangular splash bar	b	Staggered		0.1270	0.2032	0.31	0.46	2.10	1.30
23.	Triangular splash bar	b	Staggered		0.0508	0.1524	0.61	0.65	4.08	11.0
24.	Triangular splash bar	b	Staggered		0.1270	0.1905	0.31	0.49	2.59	1.00
25.	Triangular splash bar	b	Staggered		0.1524	0.1905	0.29	0.47	2.64	0.60
26.	Asbestos louvers	g	0.0254	0.1461	0.0254	0.2731	0.67	0.70	1.08	7.55
27.	Asbestos louvers	g	0.0254	0.1461	0.0254	0.1715	0.94	0.68	2.78	12.0
28.	Asbestos louvers	g	0.0254	0.1461	0.0254	0.5271	0.39	0.69	1.06	4.30
29.	Asbestos louvers	g	0.0254	0.1461	0.0254	0.4001	0.51	0.67	1.41	5.05
30.	Asbestos louvers	g	0.0381	0.1334	0.0254	0.1588	1.15	0.66	3.71	25.0
31.	Asbestos louvers	g	0.0381	0.1334	0.0381	0.1588	0.81	0.66	4.04	17.6
32.	Asbestos louvers	g	0.0381	0.1334	0.0381	0.3874	0.55	0.65	2.55	11.5

Table C.1 Data for Counter flow Fills (Merkel's Theory) - continued

Fill type	Description	Fig. no. 4.3.2	Dimensions				Mass Transfer		Pressure	
			a_a m	P_a m	P_t M	P_1 m	a_d	$^b d$	$^a P$	$^b P$
33.	Asbestos louvers	g	0.0381	0.1334	0.0381	0.5144	0.33	0.63	2.22	6.20
34.	Rectangular splash bar	h	L_t= 0.05		0.2032	0.2286	0.28	0.52	2.08	5.40
35.	Rectangular splash Bar	h	L_t= 0.05		0.2032	0.3048	0.26	0.53	1.90	3.40
			Corrugations horizontal		Corrugations vertical					
36.	Corrugated splash sheets	i	0.0540	0.1461	$^a b$= 0.0540	Pb= 0.1461	0.61	0.73	1.82	9.70
37.	Corrugated splash Sheets	i	0.0270	0.0730	0.0270	0.0730	1.10	0.80	2.75	24.6
38.	Corrugated splash Sheets	i	0.0270	0.0730	0.0540	0.1461	0.68	0.79	1.90	8.0
39.	Corrugated splash Sheets	i	0.0540	0.1461	0.0270	0.0730	0.81	0.79	3.81	31.2
40.	Corrugated splash Sheets	i	0.0603	0.1778	0.0603	0.1778	0.53	0.71	2.71	10.8
41.	Corrugated splash Sheets	i	0.0270	0.0730	0.2220	0.0746	0.44	0.72	2.60	3.60

Table C.2 (a) Data for Cross flow Fills (Merkel's Theory)

Mass transfer per meter of fill of ATD, $h_d\, a_{fi}/G_w = \dfrac{h_d a_{fi}(ATD \times width)}{m_w} = a_d\,(G_w/G_a)^{-b_d}$

Loss coefficient per meter of fill height or ATD, $K_{fi1} = a_p\, G_w^{b_{pa}}\, G_a^{b_{pb}}$

Fig. 4.3.3	Description, Spacing (mm)	Airflow orientation	Fill con-Figuration	Size(s) tested, H×W×ADT [m]	a_d	$^b d$	$^a p$	$^b pa$	$^b pa$
a	Doron V-bar, 101.6×203.2	Parallel	Staggered	3.658×2.438×1.829 3.658×2.438×2.438	0.268	0.56	0.751	0.66	-0.73
a	Doron V-bar, 203.2×203.2	Parallel	In-line	3.658×2.438×1.829 3.658×2.438×2.438	0.239	0.38	0.985	0.72	-0.82
b	Ecodyne T-bar, 101.6×203.2	Parallel	Staggered	3.658×2.438×1.829 3.658×2.438×2.438	0.263	0.34	0.112	0.30	-0.22
b	Ecodyne T-bar, 203.2×203.2	Parallel	In-line	3.658×2.438×1.829 3.658×2.438×2.438	0.245	0.35	0.206	0.89	-0.069
c	Wood lath, 101.6×101.6	Parallel	Staggered	3.658×2.438×1.829 3.658×2.438×2.438	0.274	0.45	1.437	0.76	-0.80
c	Wood lath, 101.6×101.6	Perpendicular	Staggered	3.658×2.438×1.829 3.658×2.438×2.438	0.358	0.57	1.828	0.71	-0.59
d	Marley Alpha-bar, 101.6×406.4	Perpendicular	Staggered	3.658×2.438×1.829 3.658×2.438×2.438	0.307	0.052	1.816	0.71	-0.85

C.2 (b) Data for Counter flow Fills (Merkel's Theory)

Mass transfer per meter of fill of ATD, $h_d a_{fi}/G_w = h_d a_{fi}/G_w = a_d (G_w/G_a)^{b_{da}} ATD^{b_{db}}$									
Loss coefficient per meter of fill height or ATD, $K_{fi1} = a_p (G_w)^{b_{pa}} (G_a)^{b_{pb}} ADT^{b_{pc}}$									
Fig. 4.3.3	Description	Size(s) tested, H×W×ADT [m]	a_d	b_{da}	b_{db}	a_p	b_{pa}	b_{pb}	b_{pc}
e	American Tower Plastics Cool Drop	H×W=2.438×2.438 ADT=2.0,2.8 and 3.4	0.710	-0.42	-0.50	2.880	0.85	-0.600	0.17
f	Ecodyne Shape 10	H×W=2.438×2.438 ADT=1.829,2.438 And 3.353	0.605	-0.35	-0.42	1.103	1.10	-0.640	0.32
g	Toshi Fiber Cement (Dimpled and Unslotted)	H×W=2.438×2.438 ADT=1.22,1.62 and 2.03	1.169	-0.64	-0.51	0.621	0.99	-0.350	0.17
h	Munters 19060	H×W=2.438×2.438 ADT=0.609,0.914 and 1.524	1.597	-0.59	-0.19	6.875	0.31	-0.048	0.014
i	American Tower Plastics Cool Film	H×W=2.438×2.438 ADT=1.0, 1.5 and 2.0	2.138	-0.56	-0.38	7.821	0.23	-0.039	0.038
j	Marley MC67	H×W=2.438×2.438 ADT=0.914,1.219 and 1.524	1.495	-0.63	-0.35	7.089	0.27	-0.140	0.005
k	Brentwood Ind Accu-Pak CF1900	H×W=2.438×2.438 ADT=0.914,1.524 and 2.134	1.664	-0.62	-0.27	3.691	0.31	-0.099	0.45
l	Marley Alpha-bar, 101.6×406.4	H×W=2.438×2.438 ADT=0.914,1.554 and 2.144	1.41	-0.56	-0.38	7.821	0.23	-0.039	0.038
m	Brentwood Accu-Pak CF1900	H×W=2.438×2.438 ADT=0.914,1.554 and 2.154	1.732	-0.62	-0.27	3.691	0.31	-0.099	0.45

Combinations of different types of fill may be installed in a cooling tower to achieve a desired performance or to enhance the performance and reduce fouling in an existing tower.

Monjoie, Mortensen report certain fills tend to be more susceptible to fouling than others. When selecting a particular fill for a cooling system, it is important not only to consider initial performance characteristics and cost but also the long term structural performance and fouling characteristics. These can have significant cost implications on plant performance or output.