Indian Institute of Technology Delhi

Numerical Heat–Mass Transfer Modeling of a Baffle-Optimized Cooling Tower

Author: **Dhruvraj Gohil**
Department: Chemical Engineering

Abstract

This paper focuses on the detailed numerical analysis of a cooling tower by adding different baffles with different number and different baffle angle to increase the overall thermal efficiency of cooling tower. In the paper we discussed how baffle geometry parameters such as baffle angle and baffle number which affect the flow of air, temperature distribution, and evaporation rate of cooling tower we understand this with help of mass transport, heat transport phenomena within the cooling tower. mass and energy balance equations are computed at steady state and solved using numerical methods like Finite Deference Method(FDM). The analysis show how after adding baffles how an airflow ,heat and mass transfer rates are affected in cooling tower. The results show that the use of optimal number of baffles with optimal baffle angle improve the circulation between air and water streams which increase the turbulence levels and increase the overall heat and mass transfer coefficients so that it improve performance of cooling tower. It was observed that moderate baffle angles and an optimal number of baffles provide optimal thermal efficiency. Improperly angled baffles and large number of baffles increased airflow resistance so it decrease the thermal efficiency and cooling performance of cooling tower. Therefore, the study suggests that modifying the geometry of cooling tower particularly through baffle with optimal number and optimal angle can enhance the thermal efficiency and energy performance.

Contents

1 Introduction

A cooling tower is a heat removal device which is used to cool down water that has been heated during industrial processes or air-conditioning systems. These towers are used in thermal power plants, refineries and chemical industries. In the cooling tower the cooling process is done by evaporative cooling. in this process a fraction of the circulating hot water evaporates into the air stream. A wet cooling tower functions by spraying hot process water from the top of the tower through nozzles or spray systems onto the fill material. This fill material increases the surface area which is available for interaction between air and water droplets. As the air moves through the tower, it directly contacts with the falling water so both heat and mass transfer occurs During this process. also, the sensible heat is transferred from water to air and latent heat is removed through evaporation. Although these cooling towers work on a relatively simple principle, but the problem is uneven airflow and recirculation in towers. This happens particularly near the sidewalls or the fill section of tower. These irregular flow patterns cause uneven temperature distribution, uneven evaporation, and a decrease in overall thermal efficiency. The situation becomes stronger in large industrial towers. So, to overcome these problem engineers have introduced the use of baffles within cooling towers. Baffles increase the cooling performance and thermal efficiency of towers. They guide the incoming air uniformly through the fill section and prevent the formation of static region that reduce efficiency and performance. Basically, the baffles increase the contact time between air and water droplets so it promotes better evaporation and heat transfer. because of this we get more uniform temperature field within the tower and lead to higher cooling efficiency and improve heat and mass transfer.

In this work we do numerical analyses of the performance of a natural draft wet cooling tower by adding different number of baffles with different baffle angles. we study how the baffle angle and number affect the overall cooling efficiency by solving the governing equations of energy, and mass transport under steady-state conditions. At optimal angle and baffle number we get higher efficiency of cooling tower

2 Problem Background

2.1 Background theory

The cooling tower is model as a vertical cylindrical column where hot water enters at the top of the tower and flows downward through the fill section, while air enters at the bottom of the tower and rises upward due to natural buoyancy. Within this counterflow region, heat and mass transfer occur simultaneously between the air and water phases.

We want to find how adding baffles (with different angles, numbers) changes Air velocity distribution and Cooling efficiency

2.2 Assumption

The cooling tower operates under steady-state conditions, so all properties remain constant with time.

The flow of air and water is considered one-dimensional along the vertical direction (z-axis).

Radial and tangential variations of velocity, temperature, and humidity are neglected.

The physical properties of air and water (density, specific heat, latent heat) are assumed constant throughout the process.

The air–water mixture is dilute, meaning the vapor mass fraction is small and does not significantly affect air properties.

The tower is assumed adiabatic to the surroundings — no heat loss to the environment.

The cross-sectional area of the tower remains uniform along its height.

Axial heat conduction in both air and water is negligible compared to convection and evaporation.

Heat and mass transfer coefficients are determined using empirical correlations and are considered constant in each small section.

The vapor at the air–water interface is saturated and in equilibrium with the water temperature.

Radiation heat transfer between air and water is neglected.

Gravitational potential energy effects are negligible compared to the convective and evaporative heat transfer processes.

3 Numerical Methods

Three transport phenomena occur in the given problem: momentum, heat, and mass transport.

3.1 Transport Equations

Heat transport:

$$\rho c_p \frac{DT}{Dt} = \nabla \cdot (k \nabla T) + S_E \tag{1}$$

Mass transport:

$$\frac{\partial C}{\partial t} + \nabla \cdot (Cv) = \nabla \cdot (D \nabla C) + R \tag{2}$$

We model the tower along its vertical direction z (height).

3.2 Energy Balance on Water

Energy entering the water stream:

$$\dot{m}_w c_{p,w} T_w$$

Energy leaving the water stream:

$$\dot{m}_w c_{p,w} (T_w + dT_w)$$

The change in energy of water is:

$$\dot{m}_w c_{p,w} \, dT_w$$

Thus,

$$Q = \dot{m}_w c_{p,w} (T_{w,\text{in}} - T_{w,\text{out}}) \tag{3}$$

Over a small control volume, the heat transferred from water to air per unit height is given by Newton's law of cooling:

$$\delta Q = hA(z)(T_w - T_a) \, dz \tag{4}$$

For steady-state operation:

$$\text{Rate of energy loss by water} = \text{Rate of heat transfer to air}$$

Hence,

$$\dot{m}_w c_{p,w} (-dT_w) = hA(z)(T_w - T_a) \, dz \tag{5}$$

$$\frac{dT_w}{dz} = -\frac{hA(z)}{\dot{m}_w c_{p,w}} (T_w - T_a)$$

The negative sign is for heat loss and The term $\frac{hA(z)}{\dot{m}_w c_{p,w}}$ represents the rate of temperature decay per unit height. A higher h or A (due to baffles or better air–water contact) causes faster cooling.

3.3 Energy Balance on Air

Air enters at the bottom of the slice with enthalpy h_a, and the air leaves at the top with slightly higher enthalpy $h_a + dh_a$ because it gained heat from the water. The mass flow rate of air $\dot{m}_a$ is constant, so it is a steady-flow process with no accumulation. At steady state:

$$\text{Rate of energy in} + \text{Rate of heat added} = \text{Rate of energy out}$$

For air in this small section:

$$\text{Heat gained from water} = hA(T_w - T_a)\,dz$$

$$\dot{m}_a h_a + \big(hA(T_w - T_a)\,dz\big) = \dot{m}_a(h_a + dh_a)$$
$$hA(T_w - T_a)\,dz = \dot{m}_a(h_a + dh_a - h_a)$$
$$hA(T_w - T_a)\,dz = \dot{m}_a\,dh_a$$
$$\dot{m}_a\,\frac{dh_a}{dz} = hA(T_w - T_a)$$

The enthalpy of moist air is: Moist air = dry air + water vapor mixture. So enthalpy is:

$$h_a = h_{\text{dry air}} + h_{\text{water vapor (per kg dry air)}}$$

Here Y is the humidity ratio:

$$Y = \frac{m_{\text{vapor}}}{m_{\text{dry air}}}$$

For a gas, enthalpy = c_pT So, for dry air:

$$h_{\text{dry air}} = c_{p,a}T_a$$

where $c_{p,a} \approx 1.005\,\text{kJ/kg·K}$. To get 1 kg of water vapor (at temperature T_a) starting from liquid water at 0°C then we heat liquid water from 0°C to 100°C and enthalpy is $c_{p,l}(100-0)$ and then Evaporate it at 100°C and there the enthalpy is h_{fg} Then we heat the vapor from 100°C to T_a and there the enthalpy is $c_{p,v}(T_a - 100)$ these all respect to enthalpy at 0°C liquid water. Adding these three gives the total enthalpy of vapor (per kg of water vapor) relative to 0°C liquid water:

$$h_{\text{vapor}} = c_{p,l}(100-0) + h_{fg} + c_{p,v}(T_a - 100)$$

$c_{p,l}(100-0) - 100c_{p,v}$ is nearly constant (≈ 0).

$$h_{\text{vapor}} = h_{fg} + c_{p,v}T_a$$

Each kg of dry air carries Y kg of water vapor. So total vapor enthalpy per kg dry air is:

$$Y(h_{fg} + c_{p,v}T_a)$$

Total moist air enthalpy per kg dry air:

$$h_a = c_{p,a}T_a + Y(h_{fg} + c_{p,v}T_a)$$

Differentiating with respect to height:

$$\frac{dh_a}{dz} = (c_{p,a} + Yc_{p,v})\frac{dT_a}{dz} + (h_{fg} + c_{p,v}T_a)\frac{dY}{dz}$$

Substitute this into the previous equation:

$$\dot{m}_a[(c_{p,a} + Yc_{p,v})\frac{dT_a}{dz} + (h_{fg} + c_{p,v}T_a)\frac{dY}{dz}] = hA(T_w - T_a)$$

In cooling towers, the following approximations hold:

$$Y \ll 1, \quad c_{p,v} \approx c_{p,a}, \quad c_{p,v}T_a \ll h_{fg}$$

Thus,

$$(c_{p,a} + Yc_{p,v}) \approx c_{p,a}, \quad (h_{fg} + c_{p,v}T_a) \approx h_{fg}$$

Then,

$$\dot{m}_a[c_{p,a}\frac{dT_a}{dz} + h_{fg}\frac{dY}{dz}] = hA(T_w - T_a) \tag{6}$$

$$\dot{m}_a c_{p,a}\frac{dT_a}{dz} = hA(T_w - T_a) + \dot{m}_a h_{fg}\frac{dY}{dz}$$

3.4 Mass Transfer Equation (for Air)

Fick's law (1-D) gives the molar/mass flux of vapor relative to fixed coordinates:

$$j = -D_{AB}\frac{dc_v}{dz}$$

where j [kg/(m²·s)] is the mass flux of vapor into the gas and D_{AB} [m²/s] is the binary diffusion coefficient (vapor in air).

$$\frac{dc_v}{dz} \approx \frac{c_{v,s} - c_v}{\delta}$$

$$j = D_{AB}\frac{(c_{v,s} - c_v)}{\delta}$$

$c_{v,s}$ is vapor concentration at the surface (saturation at T_w); c_v is bulk vapor concentration. Humidity ratio Y is defined as mass of vapor per unit mass of dry air:

$$Y = \frac{m_v}{m_{da}}$$

The vapor mass concentration in the gas:

$$c_v = \frac{m_v}{V} = \frac{(m_v/m_{da})}{(V/m_{da})} = \frac{Y}{(V/m_{da})}$$

$$c_v \approx \rho_a Y$$

where ρ_a is the density of the air.

$$j = D_{AB}\frac{\rho_a(Y_s - Y)}{\delta}$$

$$k_c \equiv \frac{D_{AB}\rho_a}{\delta}$$

$$j = k_c(Y_s - Y)$$

The mass of vapor carried by air per unit time is:

$$\dot{m}_a Y$$

Consider the slice between z and $z + dz$.

$$\text{Increase in vapor flow through slice} = \dot{m}_a[Y(z + dz) - Y(z)] \approx \dot{m}_a \frac{dY}{dz}\, dz$$

Vapor enters the air by evaporation from water surfaces inside the slice. If $j(z)$ is the local mass flux of vapor into the air and $A(z)$ is the interfacial area in that slice, then total vapor added per second to the air in the slice is:

$$j(z)\, A(z)\, dz$$

Equate increase in vapor flow to vapor added:

$$\dot{m}_a \frac{dY}{dz}\, dz = j(z)\, A(z)\, dz$$

$$\dot{m}_a \frac{dY}{dz} = j(z)\, A(z)$$

$$j(z) = k_c[Y_s(z) - Y(z)]$$

$$\dot{m}_a \frac{dY}{dz} = k_c\, A(z)\, (Y_s - Y)$$

3.5 Vapor Pressure and Humidity Relations

The Tetens formula is:

$$p_{sat}(T_C) = 610.78 \exp\left(\frac{17.269 T_C}{T_C + 237.3}\right) \tag{7}$$

Relative humidity:

$$RH = \frac{e}{p_{sat}(T)} \quad \Rightarrow \quad e = RH\, p_{sat}(T)$$

Humidity ratio:

$$Y = 0.622 \frac{RH\, p_{sat}(T)}{p_{atm} - RH\, p_{sat}(T)} \tag{8}$$

At saturation ($RH = 1$):

$$Y_s(T_w) = 0.622 \frac{p_{sat}(T_w)}{p_{atm} - p_{sat}(T_w)} \tag{9}$$

3.6 Baffle Effects

As we add baffles it changes how air and water mix.

$$h = h_0(1 + \alpha_1 \sin\theta)(1 + \alpha_2 n - \alpha_3 n^2) \tag{10}$$

3.7 Equations (ODEs)

$$\frac{dT_w}{dz} = -\frac{hA(z)}{\dot{m}_w c_{p,w}}(T_w - T_a) \tag{11}$$

$$\dot{m}_a c_{p,a}\frac{dT_a}{dz} = hA(z)(T_w - T_a) + \dot{m}_a h_{fg}\frac{dY}{dz} \tag{12}$$

$$\frac{dY}{dz} = \frac{k_c A(z)}{\dot{m}_a}(Y_s(T_w) - Y) \tag{13}$$

3.8 Finite Difference Method (FDM)

We divide the height of tower(H) into N equal segments. For each segment i:

$$T_{w,i+1} = T_{w,i} - \frac{h_i A_i}{\dot{m}_w c_{p,w}}(T_{w,i} - T_{a,i})\,\Delta z \tag{14}$$

$$Y_{i+1} = Y_i + \frac{k_{c,i} A_i}{\dot{m}_a}(Y_{s,i} - Y_i)\,\Delta z \tag{15}$$

$$T_{a,i+1} = T_{a,i} + \frac{h_i A_i(T_{w,i} - T_{a,i}) + k_{c,i} A_i h_{fg}(Y_{i+1} - Y_i)}{\dot{m}_a c_{p,a}}\,\Delta z \tag{16}$$

3.9 Cooling Efficiency

$$\eta = \frac{T_{w,\text{in}} - T_{w,\text{out}}}{T_{w,\text{in}} - T_{a,\text{in}}} \times 100 \tag{17}$$

4 Method

Here we use the Finite Difference Method (FDM) to solve the system of ordinary differential equations (ODEs) by python code that describe the variation of water temperature $T_w(z)$, air temperature $T_a(z)$, and humidity ratio $Y(z)$ along the height of the cooling tower.

The cooling tower involves three coupled transport processes Heat transfer between water and air, Mass transfer of evaporated water vapor, and Momentum effects, which are simplified through the use of empirical coefficients h_{local} (heat transfer coefficient) and k_c (mass transfer coefficient).

To numerically solve these equations, the tower of total height H is divided into N segments of equal thickness, $\Delta z = H/N$.

The system of differential equations:

$$\frac{dT_w}{dz} = -\frac{hA}{\dot{m}_w c_{p,w}}(T_w - T_a)$$

$$\frac{dY}{dz} = \frac{k_c A}{\dot{m}_a}(Y_s - Y)$$

$$\frac{dT_a}{dz} = \frac{A}{\dot{m}_a c_{p,a}}[h(T_w - T_a) + k_c h_{fg}(Y_s - Y)]$$

where T_w and T_a are the water and air temperatures, respectively, Y is the humidity ratio, and Y_s is the saturation humidity ratio at the water surface temperature T_w. In the FDM we replace the derivatives with respect to height z by

$$\frac{dT}{dz} \approx \frac{T_{i+1} - T_i}{\Delta z}, \qquad \frac{dY}{dz} \approx \frac{Y_{i+1} - Y_i}{\Delta z}$$

This converts the continuous differential equations into iterative update equations that can be solved step by step, from the bottom of the tower ($i = 0$) to the top ($i = N$).

At each segment i, the temperature and humidity are updated as follows:

Water Energy Equation:

$$T_{w,i+1} = T_{w,i} - \frac{hA}{\dot{m}_w c_{p,w}}(T_{w,i} - T_{a,i})\,\Delta z$$

Air Humidity Equation:

$$Y_{i+1} = Y_i + \frac{k_c A}{\dot{m}_a}(Y_{s,i} - Y_i)\,\Delta z$$

Air Energy Equation:

$$T_{a,i+1} = T_{a,i} + \frac{A\,\Delta z}{\dot{m}_a c_{p,a}}[k_c h_{fg}(Y_{i+1} - Y_i) + h(T_{w,i} - T_{a,i})]$$

Here, $Y_{s,i}$ is the saturation humidity ratio evaluated at the local water temperature $T_{w,i}$.

The saturation humidity ratio is determined by:

$$Y_{sat}(T_w) = 0.622\,\frac{p_{sat}(T_w)}{p_{atm} - p_{sat}(T_w)}$$

where $p_{sat}(T_w)$ is the saturation vapor pressure at water temperature T_w, calculated using the Tetens empirical equation:

$$p_{sat}(T_w) = 610.78\,\exp\left(\frac{17.269\,T_w}{T_w + 237.3}\right)$$

Here, T_w is expressed in degrees Celsius.

5 Observation

Initial Parameter Values:

Parameter	Value
ρ_a	1.184 kg/m^3
$c_{p,a}$	1005 J/(kg $\cdot$ K)
$c_{p,w}$	4180 J/(kg $\cdot$ K)
h_{fg}	2.26×10^6 J/kg
P_{atm}	1005 J/(kg $\cdot$ K)
$\dot{m}_w$	1 kg/s
$\dot{m}_a$	2 kg/s
$T_{w,\text{in}}$	40°C
$T_{a,\text{in}}$	25°C
A	20 m^2
h_0	9 W/(m^2·K)
k_c	7.4×10^{-3} m/s
RH	0.5
H	20
N	10
α_1	0.7
α_2	0.15
α_3	0.01

Observation Table:

θ (°)	n	$T_{w,\text{out}}$ (°C)	$T_{a,\text{out}}$ (°C)	Q (kW)	$\dot{m}_{\text{evap}}$ (kg/s)	Efficiency (%)
0	0	36.400	36.608	63.493	0.02477	23.9998
0	1	36.343	36.680	62.986	0.02449	24.3803
30	1	36.290	36.706	61.943	0.02398	24.7340
45	1	36.289	36.686	61.646	0.02384	24.7418
60	1	36.291	36.668	61.458	0.02376	24.7286
90	1	36.293	36.653	61.321	0.02371	24.7111
0	2	36.315	36.707	62.615	0.02430	24.5688
30	2	36.289	36.686	61.651	0.02385	24.7420
45	2	36.292	36.661	61.388	0.02373	24.7205
60	2	36.296	36.641	61.224	0.02366	24.6950
90	2	36.299	36.626	61.106	0.02362	24.6718
0	3	36.301	36.714	62.346	0.02417	24.6610
30	3	36.291	36.668	61.451	0.02376	24.7279
45	3	36.296	36.640	61.214	0.02366	24.6932
60	3	36.300	36.621	61.068	0.02360	24.6634
90	3	36.304	36.606	60.963	0.02356	24.6387
0	4	36.294	36.713	62.154	0.02407	24.7051
30	4	36.293	36.652	61.314	0.02370	24.7100
45	4	36.300	36.625	61.096	0.02361	24.6697
60	4	36.304	36.606	60.962	0.02356	24.6386
90	4	36.308	36.592	60.867	0.02352	24.6139
0	5	36.291	36.709	62.023	0.02401	24.7254
30	5	36.296	36.641	61.223	0.02366	24.6948
45	5	36.302	36.614	61.018	0.02358	24.6521
60	5	36.307	36.596	60.893	0.02353	24.6208
90	5	36.311	36.582	60.804	0.02349	24.5965
0	6	36.290	36.705	61.941	0.02398	24.7342
30	6	36.297	36.634	61.167	0.02364	24.6843
45	6	36.304	36.607	60.971	0.02356	24.6406
60	6	36.309	36.589	60.851	0.02351	24.6095
90	6	36.312	36.576	60.765	0.02348	24.5855

Table 1: Simulation results for varying baffle numbers (n) and inclination angles (θ).

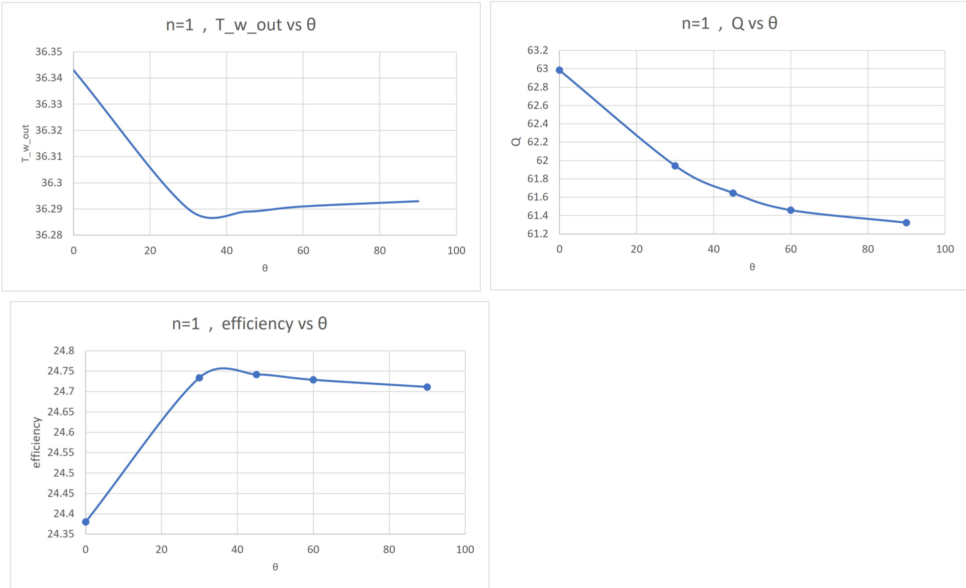

Figure 1: for n=1

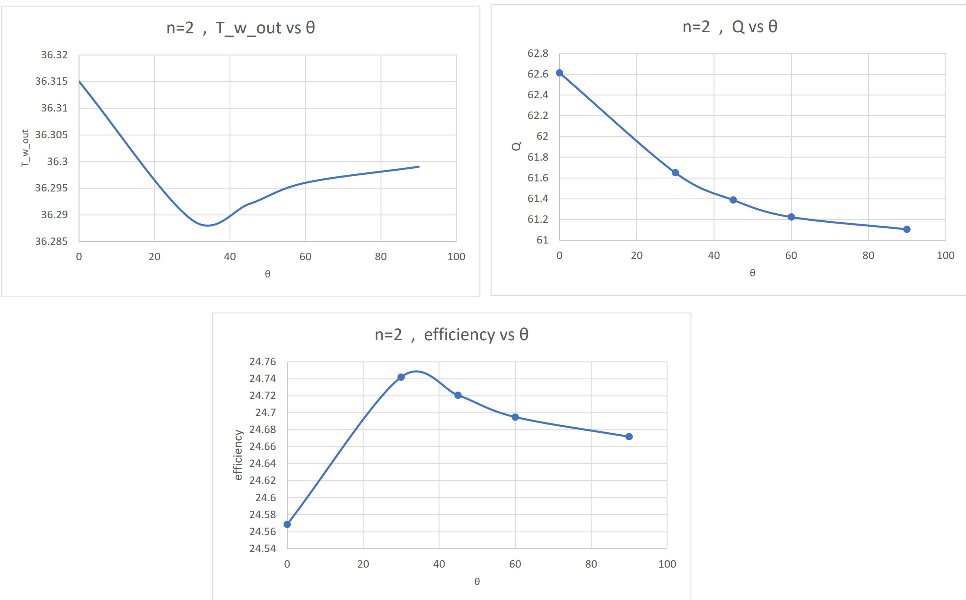

Figure 2: for n=2

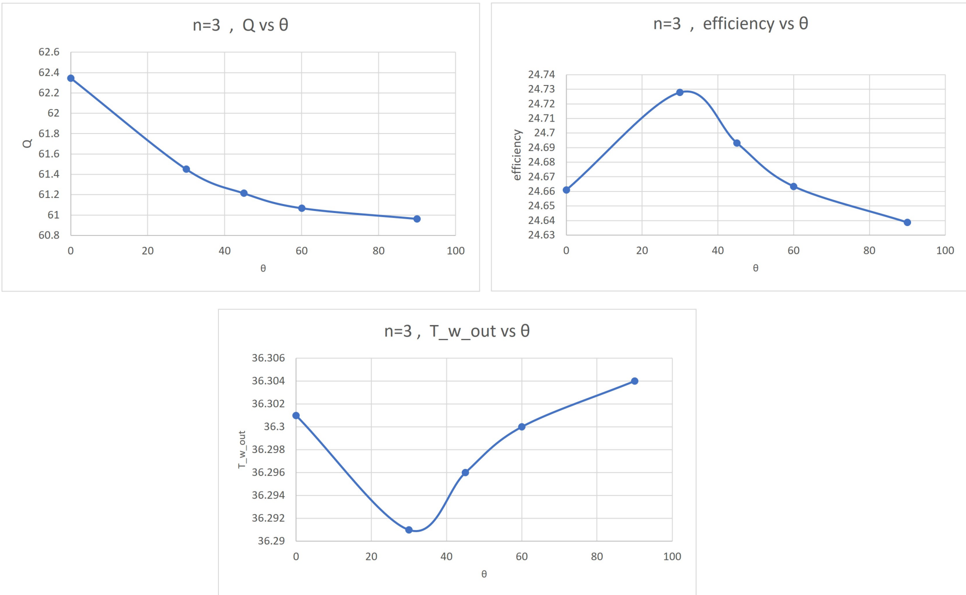

Figure 3: for n=3

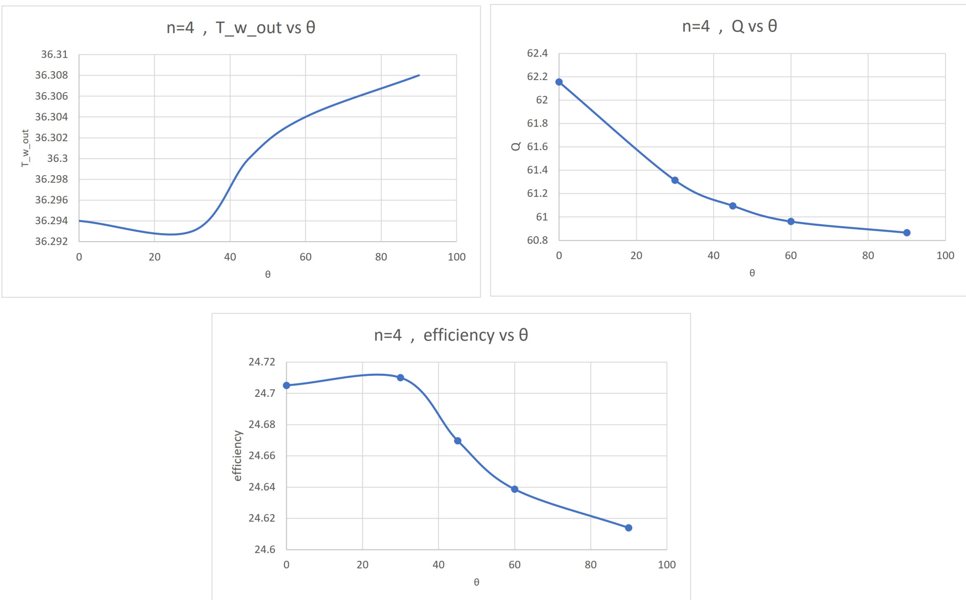

Figure 4: for n=4

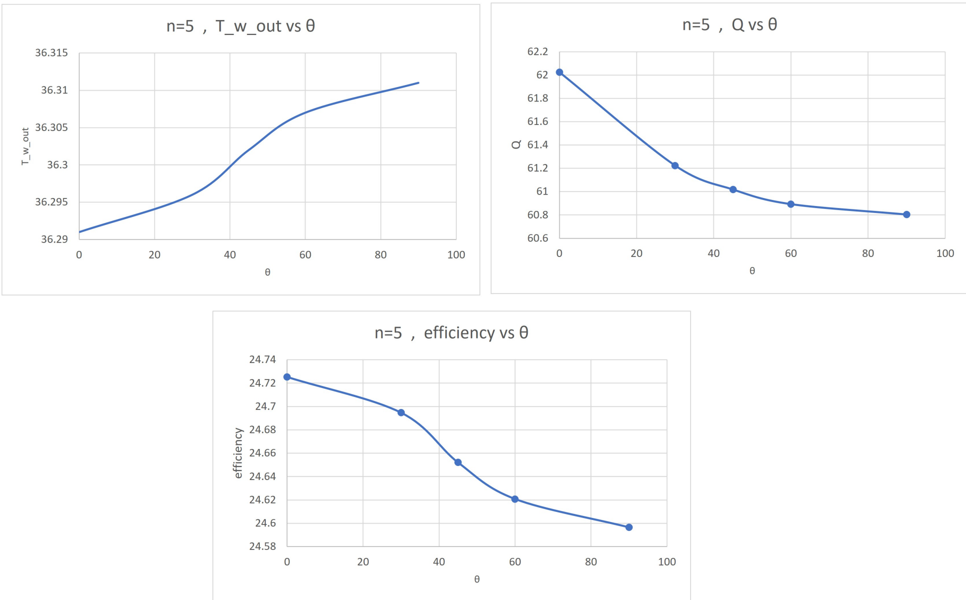

Figure 5: for n=5

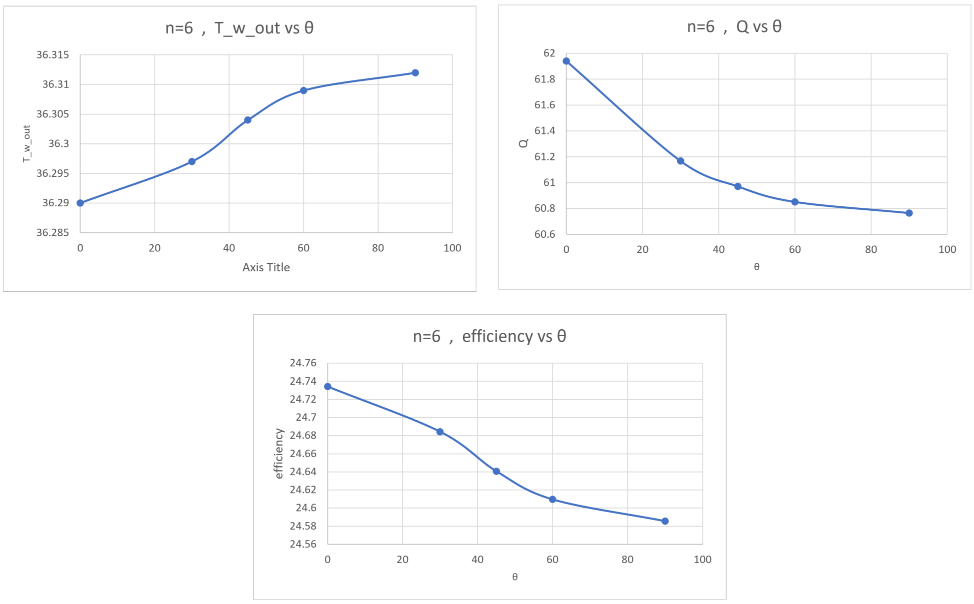

Figure 6: for n=6

6 Results and Discussion

The calculation results for varying baffle numbers ($n = 0$–6) and baffle inclination angles ($\theta = 0°$–$90°$) show how the arrangement and number of baffles affect the thermal performance, evaporation rate, and overall cooling efficiency of the cooling tower.

The outlet temperature of water ($T_{w,\text{out}}$) decreased when baffles were added compared to the case without baffles. Without baffles ($n = 0$), the outlet temperature $T_{w,\text{out}}$ is 36.40°C, and with 1–6 baffles, it gradually decreased to around 36.29–36.31°C. Similarly, the air outlet temperature ($T_{a,\text{out}}$) increased due to increased heat and mass transfer within the cooling tower. It is 36.60°C without baffles and ranges between 36.57–36.70°C with baffles. These small variations confirm that baffles improve air and water contact and enhance heat transfer efficiency.

The total heat transfer rate (Q) decreased slightly with an increasing number of baffles. For $n = 0$, $Q = 63.49$ kW, while for $n = 6$, it lies between 60.76–61.94 kW. This shows that although baffles increase turbulence and thermal contact between air and water, a larger number of baffles increases flow resistance, reducing the overall mass flow rate and thus the total heat transfer decrease.

The evaporative mass flow rate ($\dot{m}_{\text{evap}}$) followed a similar trend as the heat transfer rate. It reduced from 0.02477 kg/s without baffles to approximately 0.02348 kg/s with baffles. This indicates that while the air and water contact area improved, the increased air path restriction slightly decreased the total mass transfer rate, causing a small reduction in the overall evaporation performance.

The cooling efficiency initially increased with the addition of baffles and then decreased for higher baffle numbers. Cooling efficiency was 23.99% without baffles and increased to around 24.74% when the number of baffles was between 1 and 3. Then it slightly reduced to approximately 24.58–24.59% when the number of baffles reached 6. The best performance was observed when the number of baffles was between 2 and 3, indicating optimal efficiency. Adding a larger number of baffles increases airflow resistance, leading to a decrease in overall cooling performance.

The effect of baffle inclination angle (θ) was found to be less significant compared to the number of baffles. Both efficiency and Q showed a slight increase up to $\theta = 30°$–$45°$, after which the values decreased slightly. This indicates that moderate inclination improves fluid distribution inside the tower, whereas higher inclination angles cause air recirculation and increased resistance, reducing the overall performance and efficiency of the cooling tower.

7 Conclusions

After adding baffles to the cooling tower the mixing and contact time of air and water increases, so the baffles increase the overall heat and mass transfer and also increase the overall efficiency by increasing turbulence and increasing the contact between air and water.

As the baffle angle (θ) increases, the convective heat transfer coefficient h increases due to a greater flow disturbance.the improved mixing of air and water results in lower outlet water temperatures and higher cooling efficiency. However beyond an optimal inclination, the aerodynamic(air flow) resistance starts to dominate and cause a decrease in efficiency and an overall drop in performance of cooling towers.

Similarly, increasing the number of baffles(n) initially improves the heat transfer by increasing surface contact and interaction with water and air, but large baffles block airflow and cause recirculation of air. the observation table confirms that an optimal combination of baffle number and baffle angle yields good efficiency.

Overall, the observation table validates that modifying the geometry of cooling towers particularly through baffle with optimal number and optimal angle can enhance the overall efficiency and energy performance.

8 Appendices

8.1 Notations

Notations:

Symbol	Description
z	Vertical coordinate of tower height
H	Total height of the cooling tower
A	Heat and mass transfer area per unit height
$\dot{m}_a$	Mass flow rate of dry air
$\dot{m}_w$	Mass flow rate of circulating water
ρ	Density of air
T_a	Air temperature
T_w	Water temperature
Y	Humidity ratio (mass of vapor per mass of dry air)
Y^*	Humidity ratio of saturated air at water temperature
P	Static pressure of air
h	Convective heat transfer coefficient
j	mass flux of water vapor
k_c	Mass transfer coefficient
h_{fg}	Latent heat of vaporization of water
$c_{p,a}$	Specific heat of dry air
$c_{p,v}$	Specific heat of water vapor
$c_{p,w}$	Specific heat of water
g	Gravitational acceleration
μ	Dynamic viscosity of air
D_{AB}	Diffusion coefficient of water vapor in air
Q	Total rate of heat transfer between air and water
ΔP	Total pressure drop across the tower
η	Cooling tower efficiency
A_t	Cross-sectional area of the tower
ϵ	Surface roughness (baffle or packing)
θ	Baffle angle with respect to flow direction

References

1. Bird, R. B., Stewart, W. E., & Lightfoot, E. N. (2001). *Transport Phenomena* (2nd ed., Chapter 14: Interphase Transport in Nonisothermal Systems, p. 243). John Wiley & Sons.
2. Bird, R. B., Stewart, W. E., & Lightfoot, E. N. (2001). *Transport Phenomena* (2nd ed., Chapter 17: Diffusivity and the Mechanisms of Mass Transport, pp. 513–516). John Wiley & Sons.
3. Guo, Q., Qi, X., Zheng, W., & Sun, P. (2019). An analytical approach to wet cooling towers based on functional analysis. *Mathematical Problems in Engineering, 2019*, Article 8372093. `https://doi.org/10.1155/2019/8372093`
4. Çengel, Y. A., Boles, M. A., & Kanoğlu, M. (2019). Gas–vapor mixtures and air conditioning (Chapter 14, pp. 713–719). In *Thermodynamics: An Engineering Approach* (9th ed.). McGraw-Hill Education.