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| Research Article

Effect of Inlet Air Humidity on Thermal Behavior of Industrial Wet Cooling Towers: Comparative Modeling Using Poppe and Merkel Approaches

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Annotation
This study presents a detailed performance analysis of an induced draft counter flow wet cooling tower (IDCFWCT) at the Basrah Refinery using two established models: Poppe and Merkel. A comprehensive numerical simulation was conducted using MATLAB/Simulink, incorporating mass and energy conservation principles to predict outlet water temperature, heat rejection, and air exit conditions under varying relative humidity and airflow velocities. Experimental data were collected for validation. The results demonstrated the superior accuracy of the Poppe model, particularly under low humidity levels, due to its advanced treatment of evaporative processes. It was found that maintaining an inlet air velocity near 4 m/s offers optimal thermal performance. These insights are valuable for improving energy efficiency and water conservation in industrial cooling operations.

Keywords: Evaporative cooling, Numerical modeling, Heat rejection, Air-water interaction, Humidity impact, Induced draft.

1. Introduction

Wet cooling towers are considered essential units in industrial systems for getting rid of excess heat, especially in fields like power plants, oil refineries, and chemical industries. These towers mainly rely on evaporative cooling, where a part of the hot circulating water evaporates when it comes into contact with the incoming air, helping to cool the remaining water by removing latent heat. The performance of wet cooling towers largely depends on the surrounding air properties, especially the wet-bulb temperature and relative humidity, as these directly influence how well heat and mass are transferred between air and water [1,2]. Among the many tower types, the induced draft counter flow wet cooling towers (IDCFWCTs) stand out for their efficient thermal operation, which is often attributed to the design of their fill materials and the effectiveness of air-water interaction [3]. Still, predicting their thermal behavior accurately under changing environmental conditions can be quite difficult particularly when humidity is low and common modeling assumptions begin to lose their accuracy. Merkel's model, although widely applied for its simplicity, often struggles to give precise predictions in real-world scenarios [4].

Due to these challenges, more detailed approaches like the Poppe model have gained attention. This model accounts for both energy and mass balances in an integrated way, which improves prediction reliability under a range of operating conditions [5,6]. Evaluating and comparing such models plays an important role in improving the design and performance of cooling towers, especially in places with wide seasonal climate variations like southern Iraq.

2. LITERATURE REVIEW

Many researchers have studied counter flow cooling towers due to their significant importance in industrial applications and power plants. Among these researchers, **Jin et al., 2007** [7] present a simplified model for the study of CT performance. With an error rate of 5.6%, the findings revealed that the model could highly precisely forecast tower performance. **Qi et al., 2008** [8] created a better mathematical model for studying CT performance. Particularly in the analysis of water mass loss, the results revealed that the new model is more accurate than previous ones. **Ren et al., 2008** [9] investigated CT water evaporation. The results revealed that the model is sensitive to the saturation level of the air inlet; lower temperatures of the wet-bulb boost cooling capacity by 2.25%, while the temperature overall of water decrease diminishes with a higher water-to-air mass flow ratio by 3.5%. Developed a model for heat and mass transfer analysis in CTs under **Klimanek et al., 2009** [10] with less than 1% of errors, the model proved consistent with the Poppe model. **Costello et al., 2009** [11] examined CT performance under constrained running conditions. The optimal performance requires ratio of flow rate water to air (L/G) less than 1.0, according to the findings. Using Visual Studio. .NET, **Panjeshi et al., 2010** [12] developed a model for cooling tower design. Raising the wet-bulb air temperature increases the outlet water temperature, according to the findings. **Ragupathy et al., 2011** [13] investigated how well-expanded wire mesh packing cooled towers. The results revealed that vertical packing performs better than horizontal packing. **Rubio-Castro et al., 2011** [14] developed a Poppe model-based optimization method for CT design. Results showed that the evaporation rate using the Merkel procedure was 1.1561 kg/s When using Poppe 0.84252 kg/s, the evaporation rate decreased by 27% when using Poppe. Developed a technique using operational data for cooling tower performance analysis, **Pan et al. (2011)** [15]. Changing fan positions revealed that power output might rise by up to 260 kW. **Nasrabadi et al. (2014a)** [16] investigated low-temperature process cooling tower use. The model could forecast outlet water temperatures with an accuracy of 0.29°C for low-temperature processes and 0.570 °C for higher-temperatures operations, according to the findings. **Singh et al. (2016)** [17] investigated varying fill types' performance in cooling towers. At 25.9%, wire mesh packing offers the best efficiency according to the findings. **Anssam Dhaher Hussain,2008** [18] conducted both experimental and theoretical investigations to assess the performance of direct and indirect contact cooling towers. Mathematical models were developed to analyze the effect of operating conditions on heat and mass transfer efficiency. The results showed that the direct contact tower was approximately 20% more efficient than the indirect one. **Mohammed Faris et al., 2018 [19]** evaluated the performance of wet CTs using graphene-based Nano fluids. The findings showed a notable improvement in heat transfer efficiency and a reduction in water consumption by up to 22% compared to conventional water.

3. Mathematical Modeling Methodology

3.1 Model Assumptions

The mathematical model for both the Poppe and Merkel approaches was based on the following assumptions during its development:

1. Steady-state conditions are used to operate the cooling tower.

2. Heat and mass transfer between tower walls and the environment is negligible.
3. The air–water interface is one-dimensional along the tower height.
4. The assumption is that the specific heat of water and air is constant.
5. The cross-sectional area of the tower is uniform.
6. In the Merkel model, the Lewis factor is assumed to be unity (Lef=1).
7. In the Merkel model, evaporative water loss is considered negligible (dw=0).

To evaluate the thermal performance and predict the thermal behavior of the IDCFWCT at the Basra Refinery Plant, two widely recognized models were used: The Poppe model and the Merkel model. The models differ in their treatment of heat and mass transfer but share a common theoretical foundation based on conservation principles**.**

The mathematical formulation begins with the application of mass and energy balances over a differential control volume within the fill section of the cooling tower as illustrated in Figures 2 and 3.

The mass balance governs the rate of water evaporation, relating the change in air humidity to the evaporated water mass, while the energy balance expresses the enthalpy interaction between air and water due to both sensible and latent heat transfer. These fundamental balances serve as the basis for deriving the coupled differential equations used in the Poppe model, and the simplified analytical form in the Merkel model.

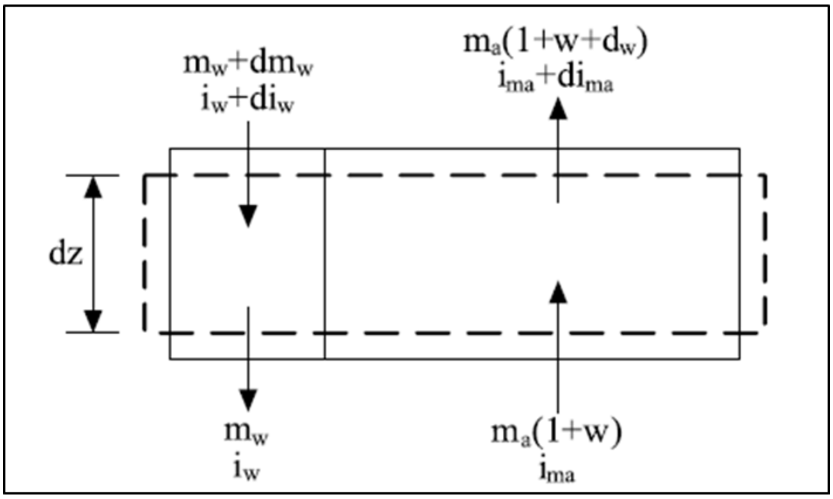

Figure 1. mass and energy flow representation for air-water [20]

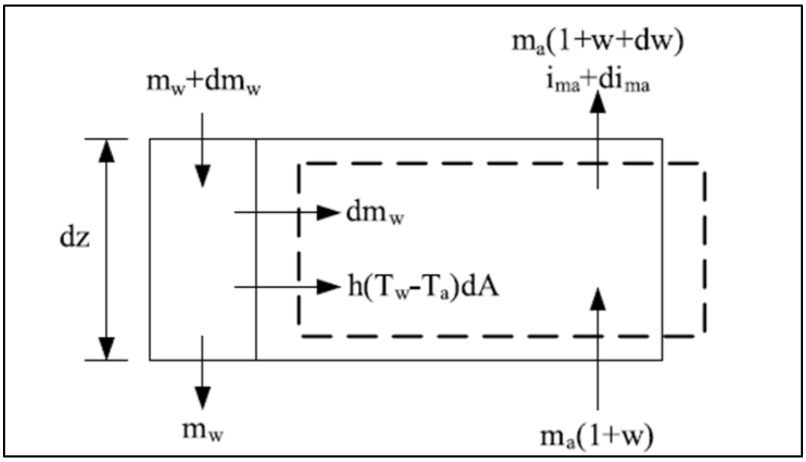

Figure 2. heat and mass transfer analysis across the interface layer [20]

3.2 Governing Equations and Simulation Basis of the Poppe Model

In counter flow cooling towers, air movement is produced mechanically using fans positioned at the base (forced draft) or at the top (induced draft) of the tower. The Poppe technique of analysis is derived from many investigations [21, 22, 23, 24]. The Poppe model provides a more rigorous representation by considering both mass balance and energy balances as shown in figures 2 and 3, including latent and sensible heat exchanges, evaporation losses, and the heat content (enthalpy) of air–water vapor mixtures. The key governing equations used in the Poppe model are summarized below:

$$\frac{dT_W}{dz} = \frac{m_a}{m_w}\left(\frac{1}{cp_W T_W}\frac{di_{ma}}{d_z} - \frac{1}{T_W}\frac{d_W}{d_Z}\right) \quad (1)$$

$$\frac{dw}{dz} = \frac{c_{pw}\frac{m_W}{m_a}(w_{sw}-w)}{i_{masw}-i_{ma}+(L_{ef}-1)[i_{masw}-i_{ma}-(w_{sw}-w)i_v]-(w_{sw}-w)\,c_{pw}T_W]}\left(\frac{dT_W}{dz}\right) \quad (2)$$

$$\frac{di_{ma}}{dz} =$$

$$c_{pw}\,\frac{m_W}{m_a}\left(1+\frac{c_{pw}\,T_W(w_{sw}-w)}{[(i_{masw}-i_{ma})+(L_{ef}-1)[i_{masw}-i_{ma}-(w_{sw}-w)i_v]-(w_{sw}-w)\,c_{pw}T_W]}\right)\left(\frac{dT_W}{dz}\right)$$

(3)

$$\frac{dMe_p}{dz} = \frac{c_{pw}}{i_{masw}-i_{ma}+(L_{ef}-1)[i_{masw}-i_{ma}-(w_{sw}-w)i_v]-(w_{sw}-w)\,cp_W T_W]}\left(\frac{dT_W}{dz}\right) \quad (4)$$

where $L_{ef} = \frac{h}{cp_{ma}h_d}$, This is referred to as the Lewis factor, and it is a measure of the relative rates of sensible and latent heat exchange that occur during an evaporative process. In order to precisely describe the Lewis factor for air-water-vapor systems, Bosnjakovic [25] created the equation that is shown below:

$$L_{ef} = 0.866^{0.667}\left(\frac{w_{sw}+0.622}{w+0.622} - 1\right)/ln\left(\frac{w_{sw}+0.622}{w+0.622}\right) \quad (5)$$

The Runge-Kutta method in MATLAB/Simulink was utilized to solve the coupled differential equations above, which enabled us to calculate the variations in water temperature, air humidity ratio, and enthalpy throughout the tower height.

The complete set of coupled differential equations derived from mass and energy balances, including expressions for Lewis factor, enthalpy gradients, and Merkel number integration, are detailed in Appendix A.

3.3 Mathematical Representation of the Merkel Model

The Merkel model simplifies the thermal analysis by assuming:

- ✓ Negligible evaporative losses (dw≈0)
- ✓ Lewis factor equals unity (Lef=1)

Under these assumptions, the energy balance reduces to the following two governing equations:

$$\frac{di_{ma}}{d_z} = \frac{h_d\,dA}{m_a}[(i_{masw} - i_{ma})] \quad (6)$$

$$\frac{dT_W}{d_z} = \frac{m_a}{m_W}\left(\frac{1}{cp_W}\frac{di_{ma}}{d_z}\right) \quad (7)$$

Combining equations (6) and (7) leads to the integral form of the Merkel number (Me_M)

$$Me_M = \frac{h_d\,A}{m_W} = \frac{h_d a_{fi} A_{fr} L_{fi}}{m_W} = \frac{h_d a_{fi} L_{fi}}{G_W} = \int_{T_{wo}}^{T_{wi}} \frac{cp_W}{(i_{masw} - i_{ma})} dT_w \quad (8)$$

Several studies and standards [26–29] have looked into numerical techniques for solving equation (8), with particular emphasis on the commonly applied 4-point Chebyshev integration method used to evaluate fill characteristics and cooling tower performance.

4. Results and Discussion

4.1 Validation of the Poppe and Merkel Model for Wet Cooling Tower Performance

The developed numerical model was validated by comparing its results with the theoretical data published by Rubio-Castro et al. (2011). In the first case analyzed, the model predicted an outlet water temperature of 25.85 °C, which is quite close to the 25.9 °C reported in the reference. Likewise, the calculated Merkel and Poppe numbers were 3.045 and 2.2991, respectively, compared to 3.083 and 2.3677 given in the Rubio-Castro study. The validation percentages for this case were 99.61% for the Poppe outlet temperature, 99.03% for the Merkel outlet temperature, 98.76% for the Merkel number, and 97.10% for the Poppe number. The test conditions included a water inlet temperature of 45 °C, an air inlet temperature of 22 °C, a mass flow rate ratio of water to air of 0.829, and a relative humidity of 0.0047. These close agreements confirm the robustness and accuracy of the proposed model in replicating the theoretical performance of wet cooling towers. Additional case results and validation percentages are summarized in Table 1.

Experimental Data(Rubio-Castro)	Case No.					
	1	2	3	4	5	6
Water Inlet Temperature (°C)	45	43	42	41	40	39
Water Outlet Temperature (°C)	26	25	24	23	22	21
Air Inlet Temperature(°C)	22	17	22	22	22	22
Mass flow rate of air (kg/s)	31.014	31.443	28.199	36.950	32.428	27.205
Mass flow rate of water (kg/s)	25.720	25.794	25.700	30.973	22.127	30.749
The mass flow rate of water to air	0.829	0.820	0.911	0.838	0.682	1.130
Relative Humidity (w)	0.0047	0.0067	0.0002	0.0047	0.0047	0.0047
Heat Rejection (kW)	3400	3400	3400	3400	3400	3400
Results Rubio-Castro Model						
Water Outlet Temperature (°C)(Poppe)	25.9	24.9	23.9	22.9	21.9	20.9
Water Outlet Temperature(°C)(Merkel)	26.3	25.28	24.26	23.27	22.31	21.32
Merkel number(Me)	3.083	3.055	2.466	2.293	7.335	1.858
Poppe number(Mep)	2.3677	1.6901	2.0671	2.3677	4.3938	1.4101
Results Predicted Model						
Water Outlet Temperature (°C)(Poppe)	25.85	24.85	23.85	22.85	21.85	20.87
Validation percentage(Tw, out) %	99.61	99.59	99.58	99.56	99.54	99.52
Water Outlet Temperature(°C)(Merkel)	26.25	25.23	24.21	23.26	22.24	21.30
Validation percentage(Tw, out) %	99.03	99.08	99.125	98.86	98.90	98.57
Merkel number(Me)	3.045	3.012	2.395	2.199	7.298	1.845
Validation percentage(Me)%	98.76	98.56	90.03	95.9	96.54	99.30
Poppe number(Mep)	2.2991	1.6581	2.0221	2.3599	4.2999	1.3645
Validation percentage(Mep) %	97.10	99.70	93.52	99.67	97.51	96.76

Table 1 presents CT Performance Comparison of Navarro Model and Predicted Model.

4.2 Operational and Environmental Conditions Affecting the Cooling Tower Performance

This section deals with investigation with and discussion of the performance of an IDCFWCT in B.R.P. the predicted model (MATLAB Simulink software) were utilized to conduct a specific evaluation of the tower's performance using the Merkel and Poppe models of analysis. This assessment is essential for understanding the tower's behavior under a variety of environmental and operational conditions and estimation of the efficiency for three cases than make a comparison of finding the percentage of validation.

1. **Case one 1**: is represented the results of experimental data.
2. **Case two 2**: the results by Poppe model assumptions.
3. **Case three 3**: the results by Merkel model assumptions.

The effect of inlet air temperatures for all cases can be investigated for four states as follows:

1. **State one 1**:is assumed the inlet air temperature is 12 ℃
2. **State two 2**: inlet air temperature is 22 ℃
3. **State three 3**: inlet air temperature is 32 ℃
4. **State four 4**: inlet air temperature is 42 ℃

4.2.1 Effect of Relative Humidity on Heat Rejection (Q) at Various Inlet Air Temperatures in a C.T

Figures 2 to 5 illustrate the effect of relative humidity on evaporative heat rejection in three cases, under constant conditions (temperature hot water: 45°C, flow rate of water: 2000 m³/h, and air velocity: 4 m/s), across varying inlet air temperatures (12–42°C). In Figures 2 and 3 (states 1 and 2), heat rejection consistently decreases with increasing relative humidity. For instance, at 12°C and 12% RH, heat rejection values were 19.85, 19.5, and 19 MW for cases 1, 2, and 3, respectively; these values dropped to 10.82, 10.5, and 10 MW at 85% RH. A similar trend is observed at 22°C, where heat rejection declined from 18.2 to 9.2 MW, 17.85 to 8.85 MW, and 17.35 to 8.33 MW as RH increased from 12% to 85%.

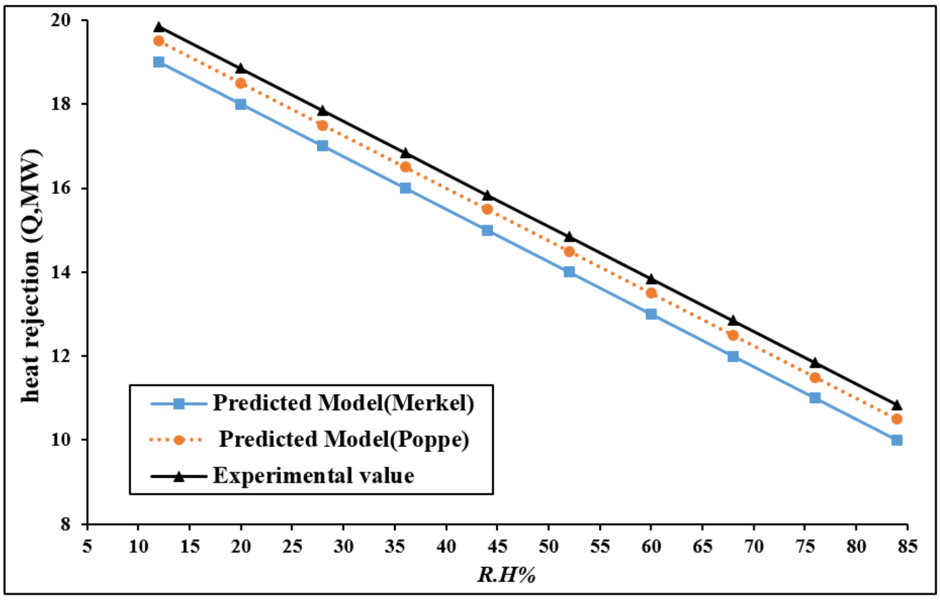

Figure 3. shows the impact of relative humidity on the rejected heat in a.

IDCFWCT at (case 1, 2, and 3, state 1)

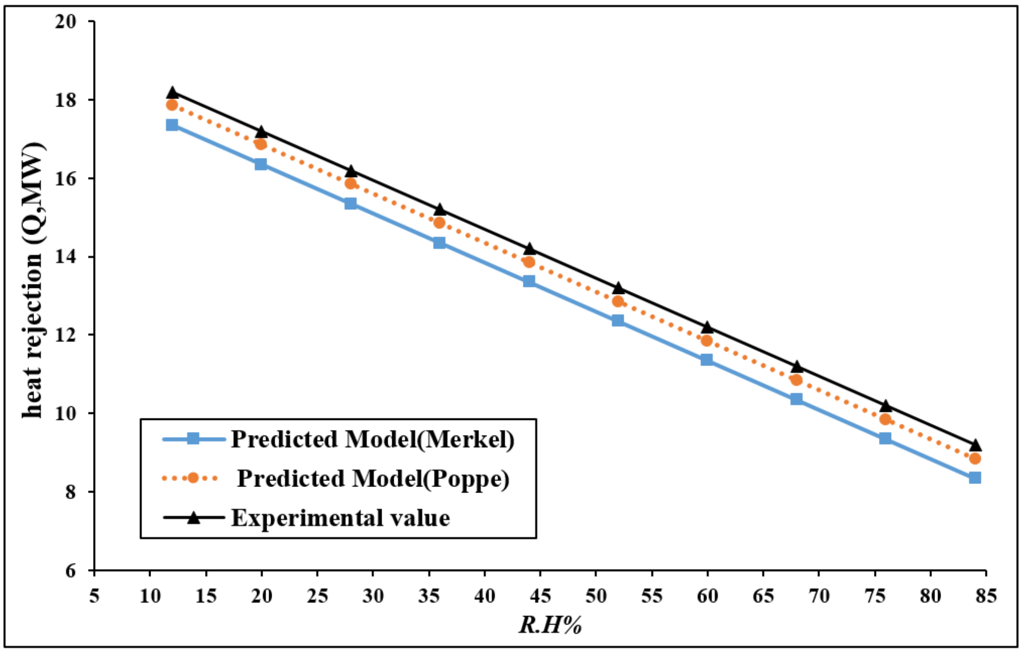

Figure 4. shows the impact of relative humidity on the rejected heat in an IDCFWCT at cases 1, 2, and 3, respectively.

Figures 4 and 5 and states 3 and 4 demonstrated a similar behavior between heat rejected and relative humidity. For state 3, when the inlet air temperature is 32°C and the relative humidity is 12%, the heat rejection value for cases 1, 2, and 3 is 16.7 MW, 16.31 MW, and 15.82 MW, respectively. As the humidity increased to 85%, heat rejection dropped for cases 1, 2, and 3 to 7.7 MW, 7.34 MW, and 6.85 MW, respectively. However, for state 4, when the air inlet temperature is 42°C, and relative humidity is 12%, the heat rejection for three cases is 15 MW, 14.65 MW, and 14.15 MW, respectively. In contrast, heat rejection values drop to 4.2 MW, 3.85 MW, and 3.32 MW when the relative humidity is changed to 85% for cases 1, 2, and 3. These behaviors confirm that the efficiency of CTs is strongly affected by both the air inlet temperatures and the relative humidity of the incoming air.

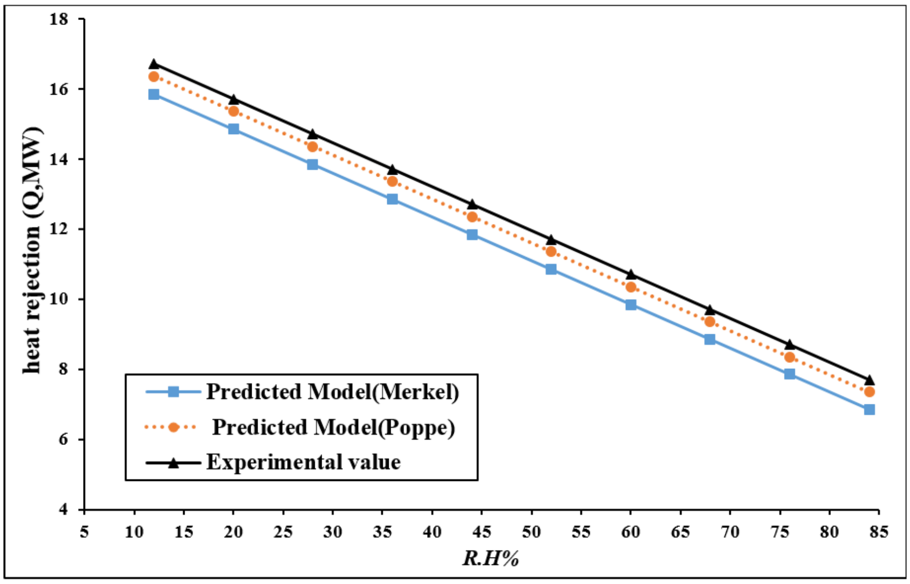

Figure 4 shows the impact of relative humidity on resisted heat in an IDCFWCT at (case 1, 2, and 3, state 3)

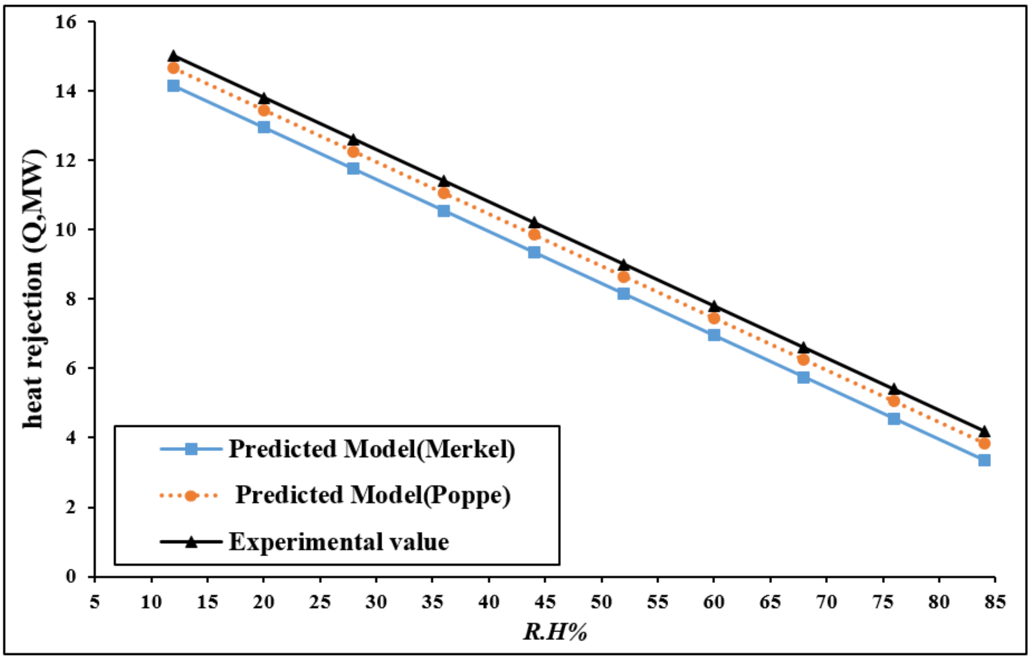

Figure 5 shows the impact of relative humidity on the rejected heat in an IDCFWCT at cases 1, 2, and 3, respectively.

These values of average verification percentage and the average relative error percentage of case 2 relative to case 1 for states 1, 2, 3, and 4 are (97.64% and 2.36%), (97.32% and 2.68%), (96.95% and 3.05%), and (95.74% and 4.26%), respectively. However, the average verification percentage and the average relative error percentage for case 3 relative to case 1 for states 1, 2, 3, and 4 are (94.24% and 5.74%), (93.5% and 6.5%), (92.6% and 7.4%), and (89.64% and 10.36%), respectively.

The previous results prove that the Poppe model is more accurate than the Merkel model. The cooling tower's heat transfer mechanism could explain the reduction in heat rejection values. When the inlet air is colder and drier, it could absorb more water vapor, resulting in greater evaporative cooling and higher heat rejection.

4.2.2 Effect of Relative Humidity on Outlet Water Temperature at Various Inlet Air Temperatures in a C.T

This section deals with the behavior of the outlet water temperature from the CT as a function of the relative humidity of the air inlet for three cases, where the flow rate of water is 2000 m³/hr, the air inlet velocity is 4 m/s, and the temperatures water inlet range is extended from 30°C to 50°C for all cases.

In Figures 6 and 7, states 1 and 2, the water outlet temperatures have increased when the relative humidity is increased for all cases. For state 1, It is demonstrated that the outlet water temperatures have increased for cases 1, 2, and 3 from 20°C to 24.5°C, 19.9 to 24.35°C, and 20.4 to 24.9°C when the relative humidity rises from 12% to 85%, respectively, when the water entering the CT is 30°C, the water flow rates is 2000 m³/hr, and the air inlet velocity is 4 m/s. While, for state 2, the outlet water temperatures have increased for three cases from 24°C to 28.5°C, 23.8 to 28.3°C, and 24.35 to 28.85°C when the relative humidity is increasing from 12% to 85%, respectively, under the conditions of an inlet water temperature is 35°C, the water flow rate is 2000 m³/hr, and the air inlet velocity is 4 m/s.

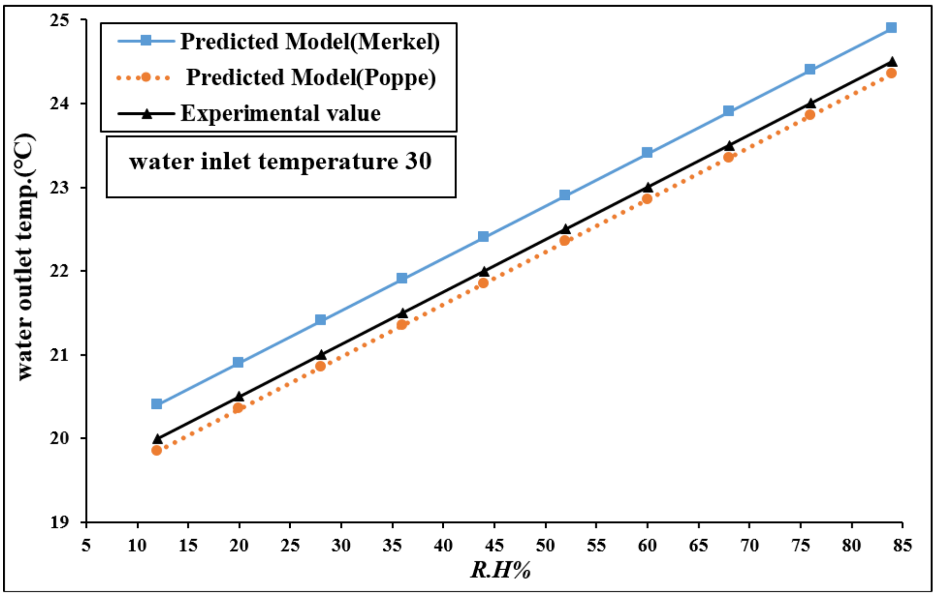

Figure 6 shows the impact of relative humidity on water outlet temperature in an IDCFWCT in case 1, 2, and 3 states.

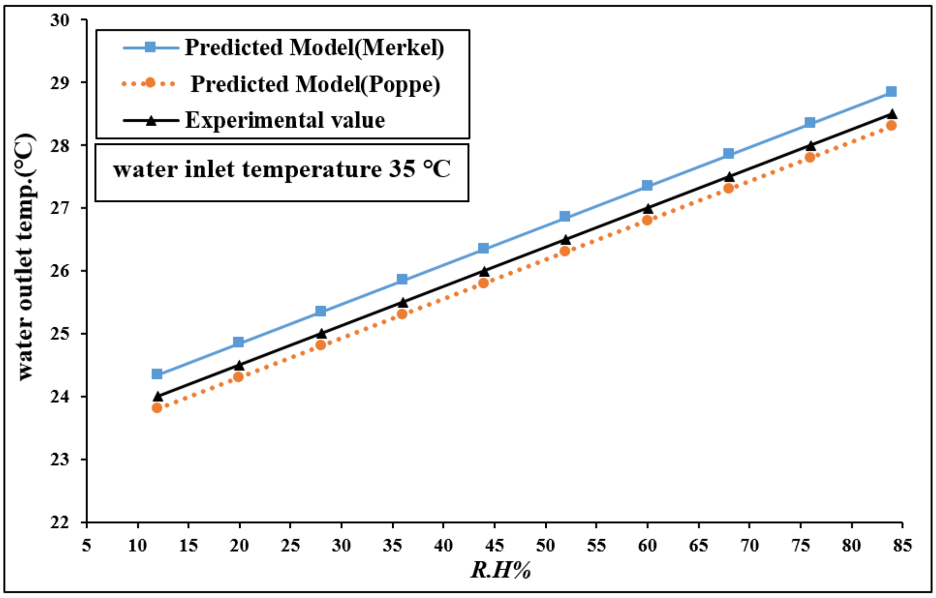

Figure 7 shows the impact of relative humidity on the water outlet temperature in an IDCFWCT at cases 1, 2, and 3.

Figures 8 and 9 state that states 3 and 4 deal with a slight increase in the outlet water temperature for three cases; for example, in state 3, as the air inlet temperature is 32℃ and the relative humidity has changed from 12% to 85%, the outlet water temperatures increase for cases 1, 2, and 3 from 30 to 34.5°C, 29.8 to 34.3°C, and 30.3 to 34.8°C, respectively, under the conditions of the water inflow to the cooling system is 40°C, the flow rate of water is 2000 m³/hr., and the air inlet velocity is 4 m/s. Similarly, in Figure 5-18, state 4 shows a sharp increase in outlet water temperature for three cases from 42°C to 43.8, 41.9 to 43.72, and 42.5 to 44.05 when the relative humidity has changed from 12% to 85%, respectively, as the water temperature inlet is 50°C, the flow rate of water is 2000 m³/hr., and the inlet air velocity is 4 m/s.

The average validation percentages and relative errors for case 2 compared to case 1 in states 1, 2, 3, and 4 are (99.32% and 0.676%), (99.23% and 0.764%), (99.69% and 0.31%), and (99.76% and

0.233%). For case 3 compared to case 1, the averages are (101.8% and 1.805%), (101.337% and 1.337%), (100.932% and 0.932%), and (100.582% and 0.582%).

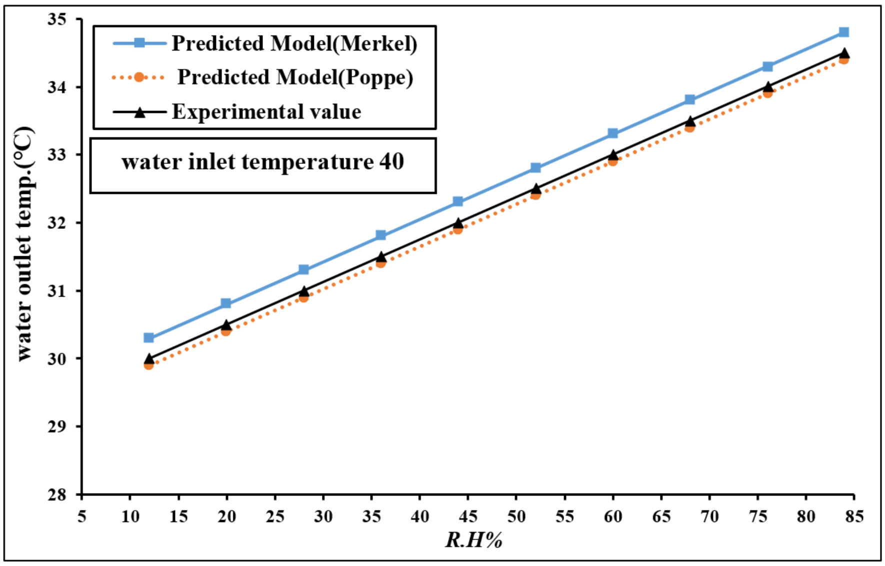

Figure 8 shows the impact of relative humidity on water outlet temperature in an IDCFWCT at (case 1, 2, and 3 states).

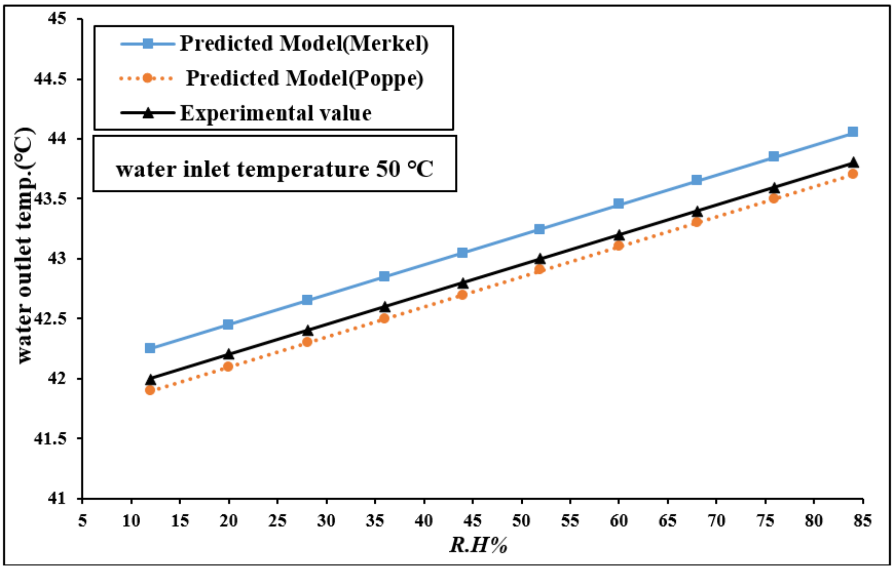

Figure 9 shows the impact of relative humidity on the water outlet temperature in an IDCFWCT at (case 1, 2, and 3 state 4).

The main reason that the Merkel model often gives higher temperatures for the air exiting the CT compared to the Poppe model is due to the model's assumptions and the physical representation of the thermal and moisture exchange process by the method. The analysis results indicate that the Poppe model demonstrates higher accuracy compared to the Merkel model, as it yields a higher average verification percentage and a lower relative error, indicating that its predictions are closer to the experimental values. This improvement can be attributed to the fact that the Poppe model is more sensitive to variations in specific humidity and latent heat content, as it accurately accounts for both thermal and moisture exchange process without assuming that the air is fully saturated, as the Merkel model does [30, 31]. In contrast, the Merkel model assumes that the air entering the CT is completely saturated with vapor water, an assumption that does not always hold true under real operating conditions, especially when the relative humidity is low [32,33]. This leads to a form of overestimation in predicting the outlet air or cold water temperature, thereby increasing the relative error [34]. Notably, the Poppe model shows consistent performance across various levels of relative humidity.

The increase in outlet water temperature under various values of the relative humidity, as shown in Figures 6, 7, 8, and 9, is due to many parameters, such as the temperature and relative humidity of the incoming air, which have a large effect on the performance of cooling inside the cooling towers. When the air entering the tower is colder, the cooling process is becoming more effective, and then the rate of heat and mass transfer is improved. Meanwhile, when the air entering the tower is hot, the cooling efficiency decreases, resulting in higher outlet water temperatures.

The relative humidity also has a significant effect on evaporative cooling efficiency. When the incoming air is drier (lower relative humidity), it can absorb more water vapor produced by the evaporation of hot water. The increase in the evaporation rate improves the heat transfer from the water to the air streams, which consequently lowers the temperature of the water exiting those streams. Vice versa, when relative humidity is high, the air is partially saturated with water vapor. Therefore, a reduction in the evaporation rate lowers the heat transfer efficiency, resulting in a higher outlet water temperature.

CONCLUSIONS

The findings of this research clearly indicate that the Poppe model offers better accuracy than the Merkel model when it comes to evaluating the performance of wet cooling towers, especially in cases where the relative humidity is low. One of the reasons for this is that the Poppe model gives a more realistic account of how heat and mass transfer happen together, which made its results much closer to what was observed experimentally. In the simulations carried out here, the air velocity was set at 4 m/s, which is the design value and generally considered optimal for tower performance. It turned out that this speed gave the best cooling results. On the other hand, increasing the air speed led to a drop in performance—probably because the contact time between air and water was reduced. Simulation tools like MATLAB/Simulink also proved to be very useful for analyzing how the tower behaves under different environmental conditions and helped in predicting performance in advance.

RECOMMENDATIONS

The study suggests using the Poppe model as a main approach for designing and improving the performance of induced draft counter flow wet cooling towers, especially in places where the relative humidity tends to change frequently. This recommendation is based on the model’s accuracy in describing both heat and mass transfer processes. It also turned out that keeping the air velocity close to the design value of 4 m/s gave the best thermal results. In terms of simulation, MATLAB/Simulink proved to be a reliable environment for carrying out numerical modeling and forecasting system behavior. To maintain steady operating conditions and improve heat exchange, it’s also suggested to consider structural enhancements like airflow guides or wind deflectors. Lastly, the study highlights the need for long-term seasonal evaluations to make sure the models stay accurate and to help guide flexible operation strategies that support better use of water and energy in industrial systems.

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Appendix

Detailed Derivation of the Poppe and Merkel Models

Appendix

Detailed Derivation of the Poppe and Merkel Models

The derivation of the governing equations begins with applying the mass balance on a differential control volume, as shown in Figure 3:

$$\dot{m}_a(1+w)+(\dot{m}_w+d\dot{m}_w)=[1+(w+d_w)]+\dot{m}_w \quad (A.1)$$

Thus:

$$dm_w = m_a d_w$$

The energy balance for the same control volume in the fill section yields:

$$m_a di_{ma} - m_w di_w - i_w dm_w = 0 \quad (A.2)$$

Where:

$$di_w = c_{pw} dT_w \quad (A.3)$$

Substituting Eq. (A.1) and (A.3) into Eq. (A.2) gives:

$$dT_w = \frac{m_a}{m_w}\left(\frac{1}{c p_w} di_{ma} - T_w d_w\right) \quad (A.4)$$

Rewriting Eq. (4.4) in terms of dw/dT_w:

$$\frac{d_w}{dT_w} = \frac{di_{ma}}{T_w di_w} - \frac{1}{T_w}\frac{m_w}{m_a} \quad (A.5)$$

To express the change in T_w with respect to height z of the tower:

$$\frac{dT_w}{d_z} = \frac{m_a}{m_w}\left(\frac{1}{c p_w T_w}\frac{di_{ma}}{d_z} - \frac{1}{T_w}\frac{d_w}{d_z}\right) \quad (A.6)$$

The enthalpy of moist air is given by:

$$i_{ma} = c_{pa}T_a + w\,(i_{fgw0} + c_{pv}T_a\,) \qquad (A.7)$$

The rate of heat transfer at the interface consists of:

$$dQ = dQ_m + dQ_c \quad (A.8)$$

Where mass transfer rate is:

$$dm_w = h_d(w_{sw} - w)dA \qquad (A.9)$$

Thus, latent heat transfer becomes:

$$dQ_m = i_v dm_w = i_v\, h_d(w_{sw} - w)dA \quad (A.10)$$

With:

$$i_v = i_{fgwo} + c_{pv}T_w \quad (A.11)$$

And sensible heat transfer is:

$$dQ_c = h(T_w - T_a)dA \quad (A.12)$$

Now, using enthalpy definitions:

$$i_{masw} = c_{pa}T_w + w_{sw}\left(i_{fgw0} + c_{pv}T_w\right) \text{ (A.13)}$$

Subtracting Eq. (4.7) from (4.13):

$$i_{masw} - i_{ma} = (c_{pa} + wc_{pv})(T_w - T_a) + (w_{sw} - w)i_v$$

$$= C_{pma}(T_w - T_a) + (w_{sw} - w)i_v \text{ (A.14)}$$

Solving Eq. (4.14) for (T_w - T_a):

$$T_w - T_a = \frac{(i_{masw}-i_{ma})-(w_{sw}-w)i_v}{c_{pma}} \text{ (A.15)}$$

Substituting into Eq. (A.12) and combining with (A.10) and (A.8), total enthalpy transfer becomes:

$$dQ = h_d\, dA\left[\frac{h}{c_{pma}h_d}(i_{masw} - i_{ma}) + \left(1 - \frac{h}{c_{pma}h_d}\right)(w_{sw} - w)i_v\right] \text{ (A.16)}$$

with:

$$L_{ef} = \frac{h}{cp_{ma}h_d}$$

Finally, Merkel's approximation (dw ≈ 0 and L_{ef} = 1) simplifies the energy equation into:

$$\frac{di_{ma}}{d_z} = \frac{h_d a_{fi} A_{fr}}{m_a}\left[L_{ef}(i_{masw} - i_{ma}) + (1 - L_{ef})(w_{sw} - w)i_v\right] \text{ (A.17)}$$

$$\frac{dw}{dz} = \frac{c_{pw}\frac{m_w}{m_a}(w_{sw}-w)}{i_{masw}-i_{ma}+(L_{ef}-1)[i_{masw}-i_{ma}-(w_{sw}-w)i_v]-(w_{sw}-w)\, c_{pw}T_w]}\left(\frac{dT_w}{dz}\right) \text{ (A.18)}$$

and the Merkel number:

$$Me_M = \int_{T_{wo}}^{T_{wi}} \frac{cp_w}{(i_{masw}-i_{ma})}\, dT_w \text{ (A.18)}$$