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DEVELOPMENT OF PERFORMANCE MODEL FOR INDUSTRIAL MECHANICAL COUNTER FLOW *COOLING TOWER*

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ABSTRACTS

A performance model of an industrial cooling tower has been developed. The cooling tower performance model was developed from first principle by taking mass and energy balance on the water entering the cooling tower and the air leaving the cooling tower. The system of ordinary differential equations arrived at were solved using the forth order Runge Kutta algorithm adapted to a Visual Basic 6.0 program compiler. The results obtained were validated using plant data obtained from the Port Harcourt Refining Company of Nigeria. The results match plant data with a deviation of 2% for the cooling water outlet temperature, and 1.4% deviation for the outlet wet bulb temperature. Simulation results show that for the proper functioning of the cooling tower, the water-to-air loading ratio should be greater than unity, so that there will be sufficient air to extract the heat from the water.
Keywords: Modelling, Counter-Flow, Industrial Cooling Tower, Wet bulb Temperature

INTRODUCTION

Cooling towers are heat rejection devices, which extracts waste heat to the atmosphere through the cooling of a water stream to a lower temperature. A cooling tower relies on the latent heat of water vapouration to exchange heat between the process and the air passing through the tower. In a cooling tower, warmer water is brought into direct contact with the cooler air. When the air enters the cooling tower, its moisture content is generally less than saturation. When the air exits, it emerges at a higher temperature and with moisture content at or near saturation. Even at saturation, cooling can take place since a temperature increase result in an increase in heat capacity which allows more sensible heat (Young and Ciammarchella, 2008). Usually a fan pushes or draws air from the atmosphere into the tower to cool recirculation water. The warm water enters through the top of the tower. As the water falls through the tower, fresh air is forced through it. This fresh air cools the water. The basic theory of cooling tower operation was first proposed by Walker *et al.* (1923), who developed the basic equations for total mass and energy transfer, and considered each process separately. Merkel (1925) combined the coefficient of sensible heat and mass transfer into a single overall coefficient based upon the enthalpy potential as a driving force. The theory proposed by Merkel requires a few simplifying assumptions, which have been almost universally adopted for the calculation of cooling tower performance. There is no mathematical model that is capable of simulating every detail of the simultaneous heat and mass transfer process occurring within the tower. Consequently, simplifying assumptions must be made for the analysis. Eight numerical models are available in the literature. These include: the ESC code based on the classical Merkel model for counter flow (Baker, 1984) and on the Zivi and Brand model cross flow towers (Zivi and Brand, 1957), FACTS developed by Benton (Benton, 1983), VERA2D which treats the flow of water in the cooling tower as one-dimensional and the flow of air as two-dimensional and steady (Majumdar *et al.,* 1983), STAR model developed by Caytan (Caytan, 1982), Sutherland's model (Sutherland, 1983), Fujita and Tezuka model (Fujita and Tezuka, 1986), Webb's model (Webb, 1988), and the Jaber and Webb model. It is difficult to draw general conclusions concerning the comparative merits of the correlations, or of the codes, yet an effort was made by Mohiuddin and Kant (1991) to compare the various models from the viewpoint of design, computational error, computational time, simplicity of usage and practicability. Heat and mass transfer are the core principle in the analysis of cooling tower performance. A basic disadvantage of the Merkel equation is the assumption that evapouration of water flow is neglected in the energy balance and that saturated air was at the exit. Zubair et al. (2003) investigated the performance characteristics through the counter flow cooling tower, the result showed that heat transfer is a major mode of heat transfer. For this reason evapouration term was included in models developed by Osterle (1991), Kloppers and Kroger (2005) and Smrekar et al. (2006).The main objective of this work is to contribute to the research for developing mathematical models for industrial counter flow cooling tower and validating the results with industrial plant data.

MATERIALS AND METHOD

Model Development

In the counter flow cooling tower, water flows downwards while air flows upwards. It is assumed that the conditions of water and air vary only with vertical position in the tower. The following assumptions were used in the development of the model: Heat and mass transfer through the tower wall to the environment is negligible, Heat transfer from the tower fan to air and water is negligible, Water and dry air specific heats are constant, Heat and mass transfer coefficient throughout the tower are constant, Heat and mass transfer is in a direction normal to the flow, The system operates at steady state, Water loss by drift is negligible, Temperatures of water and air at any cross section are uniform.

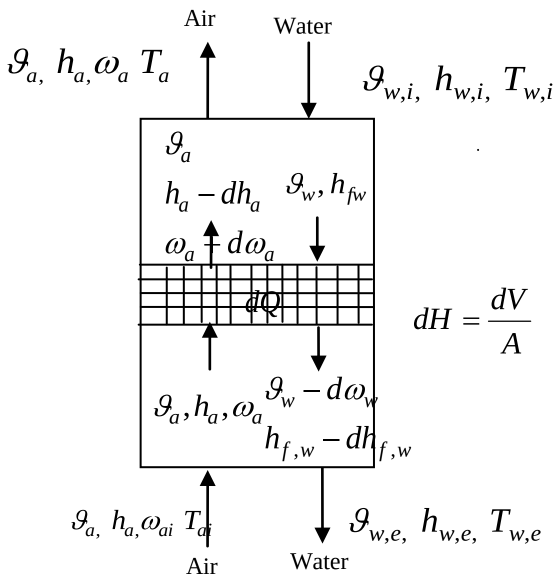

Fig.1 Schematics of mass and energy balance of a counter flow wet cooling tower.

where ϑ_a, ϑ_w are the volumetric flow rate of air and water respectively, ω is the humidity ratio, h_a is the enthalpy of air, $h_{f,w}$ is the enthalpy of saturated water vapour evaluated at the water temperature, T_a and T_w is the temperature of air and water respectively, H is the tower height, Q is the quantity of heat rejected by the system, i and e are subscripts for inlet and exit stream respectively.

Mass Balance Equation

For a steady state model without reaction taking place in the system, The Amount of Evaporated Water into the Air is derived as:

$$d\vartheta_w = \vartheta_a d\omega_a \tag{1}$$

Energy Balance Equation

The change of water temperature along the tower height is derived as:

$$\frac{dT_w}{dH} = \frac{\vartheta_a}{\vartheta_w Cp_w}\left(\frac{dh_a}{dH} - h_{f,w}\frac{d\omega_a}{dH}\right) \tag{2}$$

Where the values of dha and $d\omega a$ with respect to the tower height is evaluated as:

$$\frac{dh_a}{dH} = \frac{KaA}{dH}\left(Le_f Cp_a\left(T_w - T\right) + h_{g,w}\left(\omega_{s,w} - \omega\right)\right) \tag{3}$$

And

$$\frac{d\omega_a}{dH} = \frac{KaA}{\vartheta_a}\left(\omega_{s,w} - \omega\right) \tag{4}$$

Where $\omega_{s,w}$ the saturated humidity ratio of water is evaluated the water temperature T_w, ω is the humidity ratio, and Ka is the tower characteristic. The energy balance equation can is written based on the concept of enthalpy potential in terms of heat and mass transfer coefficients h_C and $\mathrm{h_d}$ respectively. The convective mass transfer is established from Osterle (1991) as:

$$\vartheta_a d\omega = h_d \rho_a a(\omega_{s,w} - \omega) dV \quad (5)$$

Where $\mathrm{h_d}$ is the mass transfer coefficient, $\omega_{s,w}$ is the saturated humidity ratio of water evaluated the water temperature T_w, ω is the humidity ratio.

Also for air energy balance where the phase change temperature of water, $h_{fg,w}$ is approximate to the enthalpy of saturated water vapour, $h_{g,w}$, that is $h_{fg,w} \approx h_{g,w}$ (Zubair and Qureshi, 2006).

$$\vartheta_a dh_a = h_C\ a(T_w - T) dV + h_d \rho_a a\ h_{g,w}(\omega_{s,w} - \omega) dV \quad (6)$$

$$Cp_a \vartheta_a dh_a = h_C\ a\ Cp_a(T_w - T) dV + h_d Cp_a \rho_a a\ h_{g,w}(\omega_{s,w} - \omega) dV \quad (7)$$

The resulting model for the heat transfer of air with respect to the tower heights is evaluated as:

$$\frac{dh_a}{dH} = \frac{KaA}{dH}\left(Le_f Cp_a(T_w - T) + h_{g,w}(\omega_{s,w} - \omega)\right) \quad (8)$$

Where

$$Le_f = \frac{h_C}{h_d \rho_a Cp_a} \quad \text{(Lewis factor)}$$

$$Ka = h_d \rho_a a \quad \text{(Tower characteristic)}$$

The model equation for humidity ratio with respect to the tower height is evaluated as:

$$\frac{d\omega}{dH} = \frac{KaA}{\vartheta_a}(\omega_{s,w} - \omega) \quad (9)$$

DETERMINATION OF PARAMETERS

The Tower Characteristic Ratio

The general accepted concept of cooling tower performance is in terms of the tower characteristic ratio. KaV/ϑ_w .For a counter flow tower of unit plane area through which an amount of air ϑ_a is flowing upwards and an amount of water ϑ_w is flowing downwards. The counter flow can be resolved into a one dimensional problem (Baker and Shryock, 1961), with the assumption that the flow pattern is vertical, with the water flowing downwards through the tower and the air flowing upwards. Both processes where combined by Merkel into a single equation (Mohiuddin and Kant 1995).

$$\vartheta_w Cp_w dT_w = Ka \vartheta_a (h_{g,w} - h_a) dV \quad (10)$$

Equation 10 upon integration yield:

$$\frac{KaV}{\vartheta_w} = \int_{T_{w0}}^{T_{wI}} \frac{Cp_w dT_w}{h_{g,w} - h_a} \qquad (11)$$

$$\frac{KaV}{\vartheta_a} = \int_{h_{a1}}^{h_{a2}} \frac{dh_a}{h_{g,w} - h_a} \qquad (12)$$

where $T_{wo,}T_{w1}$ are the temperatures of inlet and outlet water respectively, $h_{a1,,}h_{a2}$ are the specific enthalpy of inlet moist air and specific enthalpy of saturated air leaving the tower. Equations 11 and 12 can be calculated using Tchebysher's method as suggested by Fujita and Tezuka (1986), Hallett (1975).

MOIST AIR PROPERTIES

Moist air properties are calculated by using the following equation (ASHRAE, 1991).
The Specific Enthalpy of moist Air (h_a) is calculated from the formula:

$$h_a = \mathrm{Cp_a} T_{db} + \omega\left(2501 - 1.805 T_{db}\right) \qquad (13)$$

Where Cp_a is the specific heat of dry air, T_{db} is the dry bulb temperature of moist air.
The Specific humidity of moist air (ω) is calculated using the formula:

$$\omega = \left(\frac{\left(2501 - 2.381\, T_{wb}\right)\omega_{s,w} - \left(T_{db} - T_{dw}\right)}{2501 + 1.805\, T_{wb} - 4.186 T_{db}}\right) \qquad (14)$$

Where T_{wb} is the moist air wet bulb temperature, $\omega_{s,w}$ is the specific humidity of moist air saturated at T_{wb} .

The specific humidity of moist air saturated at T_{db} can be calculated in terms of the partial pressure of water vapour, using:

$$\omega_{s,w} = 0.62198\, \frac{P_{ws}}{P_t - P_{ws}} \qquad (15)$$

Where P_t is the total pressure, P_{ws} is the partial vapour pressure at saturation.

The water vapour saturation pressure P_{ws} in equation (15) can be calculated using one of the following two equations:

The saturation pressure over ice for the temperature range of – 100^{O}C to 0^{O}C is given by:

$$\ln\left(10^5 * P_{ws}\right) = C_1 T + C_2 + C_3 T + C_4 T^2 + C_5 T^3 + C_6 T^4 + C_7 \ln(T) \qquad (16)$$

The saturation pressure over liquid water for the temperature range of 0 – 200^{O}C is given by;

$$\ln\left(10^5 * P_{ws}\right) = C_8 T + C_9 + C_{10} T + C_{11} T^2 + C_{12} T^3 + C_{13} \ln(T) \qquad (17)$$

Where $C_1, C_2 \cdots C_{13}$ are constants, and T is the absolute temperature.

The density of moist air mixture ρ_a, is calculated using the following equation:

$$\rho_a = \frac{P_t}{R_a T(1+1.6078\,\omega)} \tag{18}$$

Where R_a is the gas constant for dry air?

The specific enthalpy of saturated air leaving the tower h_{a2} at T_{wb2} is calculated using the following equations obtained by fitting a polynomial to the enthalpy data (Mohiuddin and Kant, 1995).

For the temperature range 10 – 35 ^{O}C, the specific enthalpy of the saturated air is given by:

$$h_{a2} = C_{14}T_{wb2} + C_{15} + C_{16}T_{wb2} + C_{17}T_{wb2}^2 + C_{18}T_{wb2}^3 + C_{19}T_{wb2}^4 \tag{19}$$

For the temperature range 35 – 60^{O}C, the specific enthalpy of the saturated air a given by:

$$h_{a2} = C_{20}T_{wb2} + C_{21} + C_{22}T_{wb2} + C_{23}T_{wb2}^2 + C_{24}T_{wb2}^3 + C_{25}T_{wb2}^4 \tag{20}$$

Where $C_{14}, C_{15} \cdots C_{25}$ are constants and T_{wb2} is the wet bulb temperature of the saturated air leaving the tower.In specific case where the temperature of the saturated air leaving the tower T_{wb2} is not known to the designer, it can be approximated by the average of the inlet and outlet water temperatures. This is substantiated by Mohiuddin and Kant (1991), the data of Hutchison and Spivey (1942) and Mekelvery and Brooke (1959).

MATERIALS

The data for the calculation of the thermodynamic properties of air and water is shown in Table 1. The data for the operating condition of the cooling tower is shown in Table 2. The constants for calculating the vapour saturation pressures, and specific enthalpies of saturated air leaving the tower were obtained from Mohiuddin and Kant 1995 and ASHRAE 1991.

Table 1 – Value for Thermodynamic Properties (Ashrae 1991, Mohiuddin and Kant 1998)

Properties	Value
Specific heat capacity of dry air (Kjkg^{-1} K^{-1}).	1.004
Specific heat capacity of water (Kjkg^{-1} K^{-1})	4.187
Gas rate constant for dry air (Kj Kg^{-1} K^{-1})	0.287055
Absolute Temperature (K)	273.16

Table 2 constants for calculating the water vapour saturation pressure (ASHRAE 1991).

Constants	Value
C1	- 5674.5359
C2	6.3925249
C3	$-0.9677843 \times 10^{-2}$
C4	0.6221570×10^{-6}
C5	$0.20747825 \times 10^{-8}$
C6	$-0.9484204 \times 10^{-12}$
C7	-4.1635019
C8	-5800.2206
C9	1.3914993
C10	-0.048640239
C11	$0.41764768 \times 10^{-4}$
C12	$-0.14452093 \times 10^{-7}$
C13	6.5459673

Table 3 Constants for calculating the specific enthalpy of saturated air learning the tower (Mohiuddin & Kant 1995).

Constants	Value
C14	- 53.52515
C15	- 3.517334
C16	2.809715
C17	0.1669836×10^{-1}
C18	0.924304×10^{-3}
C19	0.9275973×10^{-5}
C20	- 26325.88
C21	3174.125
C22	-151.3213
C23	3.762619
C24	-0.04617667
C25	0.2495032×10^{-3}

Table 4 Operating condition of the cooling tower (NPHRC, 1987)

Parameter	Value
Water Flowrate (m^3/h)	13500
Inlet temperature (^OC)	42.6
Outlet temperature (^OC)	32
Wet-bulb temperature (^OC).	28
Design calorific capacity (Kcal/h)	143 x 10^6
Length of cooling tower (m)	66m
Width of cooling tower (m)	10.5m
Height of cooling tower(m)	7.8m
Fan driver speed	90kw/1500 rpm.
Fan rotation (rpm)	208

SOLUTION TECHNIQUE

The sets of Ordinary Differential Equation (ODE) obtained from the model were solved numerically using the fourth order Runge Kutta method adapted to a Visual Basic 6.0 program for simulation of the models. For the water temperature profile along the cooling tower height, equations (3) and (4) which predicts the change in humidity and enthalpy of dry air along the cooling tower respectively, were solved first and substituted into equation (2) to obtain the water temperature profile along the cooling tower height. Calculations for obtaining the parameters were also incorporated into computer programming code.

RESULTS AND DISCUSSION

The model result for the performance model is shown in Table 5. The model predicts the outlet temperature of the cooling water, the enthalpy of the dry air leaving the tower and the specific humidity along the tower height. The outlet temperature of the cooling water and the outlet wet bulb temperatures match the plant data very well with a percentage deviation of 2% and 1.42% respectively. This shows that the model can be used to monitor the performance of the cooling tower.

Table 5: Comparison between Plant and Predicted Outlet Temperature of Cooling Water

Parameter	Plant data	Model prediction	Percentage deviation
Water Outlet temperature(^0C)	32	32.64	2
Air Outlet Wet Bulb Temperature(^0C)	28	28.4	1.43

Figure 2 shows the relationship between the cooling water outlet temperature and the enthalpy of the air leaving the cooling tower along the tower height.

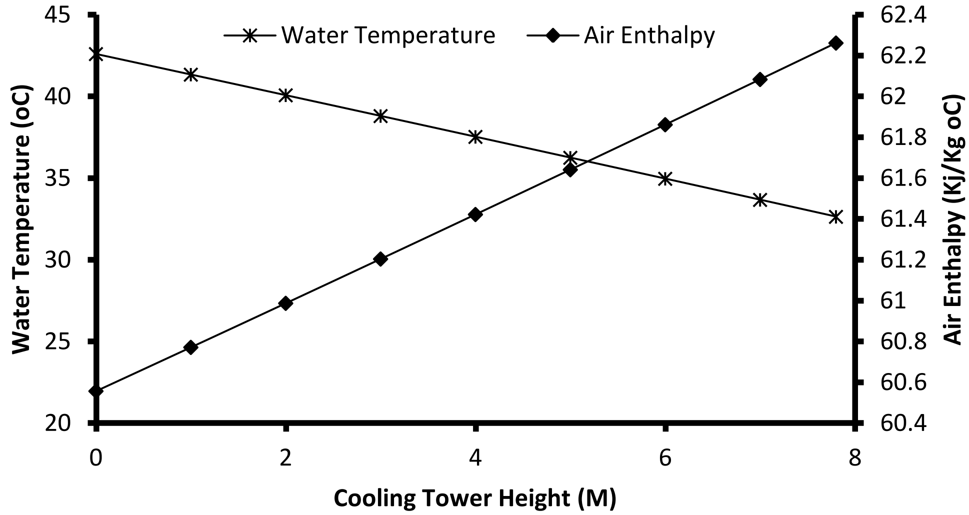

Figure 2: Profile of Cooling Water Temperature and Enthalpy of Air along the Cooling Tower Height.

It can be seen from Figure 2 that, the water is supplied at a temperature of 42.6^0C and leaves at a temperature of 32.64^0C while the heat content or enthalpy of the air increases from $60.557 Kj\ Kg^{-1}\ {}^0C^{-1}$ and leaves at an enthalpy of $62.2619\ Kj\ Kg^{-1}\ {}^0C^{-1}$. This follows the expected trend of the operation of a counterflow cooling tower. As the water enters from the top it gets in contact with the cool air coming from the bottom of the cooling tower. There is heat transfer between the water and air. The air absorbs the heat from the water thus increasing its enthalpy. Since enthalpy and temperature has a direct proportional relationship, increase in enthalpy translate into increase in the temperature of the air stream leaving the cooling tower. The relationship between the water temperature and the enthalpy of the air is clearly shown in Figure 3.

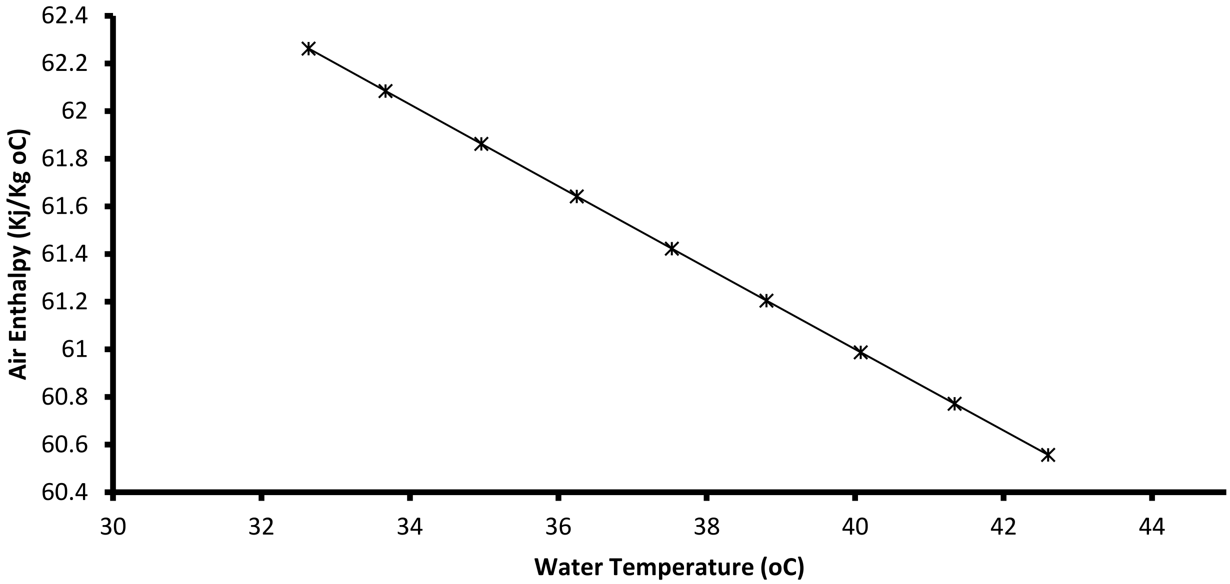

Figure 3: Relationship between Air Enthalpy and Water Temperature

It can be seen from Figure 3 that, the higher the air enthalpy the lower the water temperature. This is because the enthalpy and by extension the temperature of the air increases as more heat is absorbed from the water. Also as the water and air moves along the cooling tower height, the relative humidity also changes. The change in relative humidity along the cooling tower height is shown in figure 4.

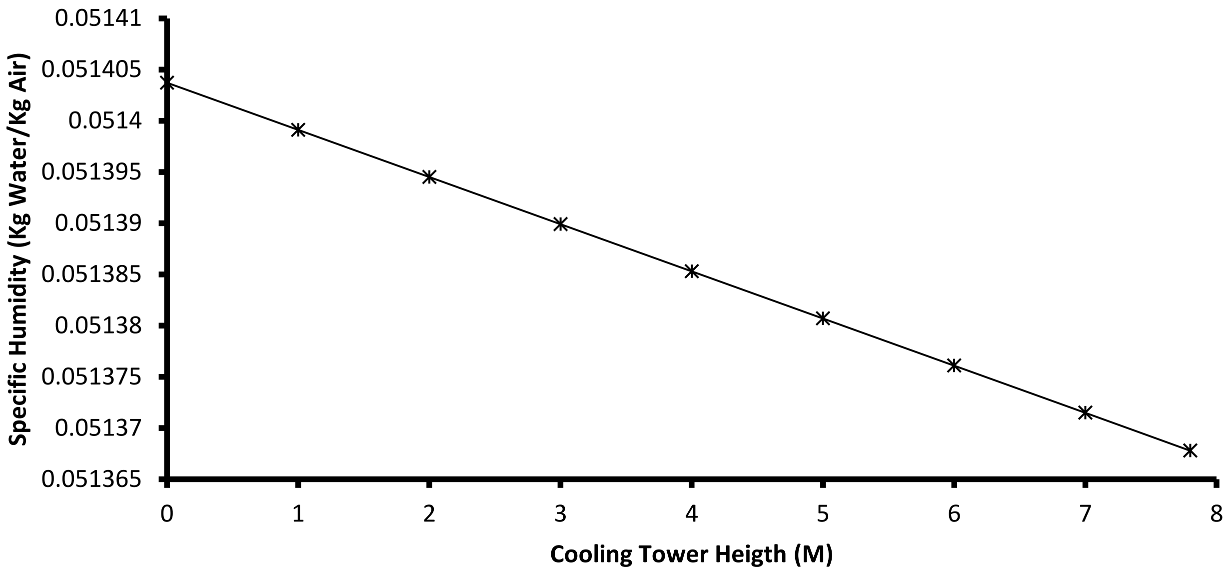

Figure 4: Profile of Specific Humidity Ratio along the Cooling Tower Height.

The effect of evaporation in the cooling of the water can be indicated in term of the air humidity ratio. The humidity ratio is the specific humidity of water per kilogram of air. It is a known fact that cooler air contains more moisture than warmer air. Thus the humidity of the hot dry air leaving the cooling tower will be less than the humidity of the cold wet air entering the cooling tower. The graph in Figure 4 follows that trend. The relative humidity at inlet is 0.0514 and that at the outlet is 0.513. The cooling range determines how much moisture is extracted from the water.

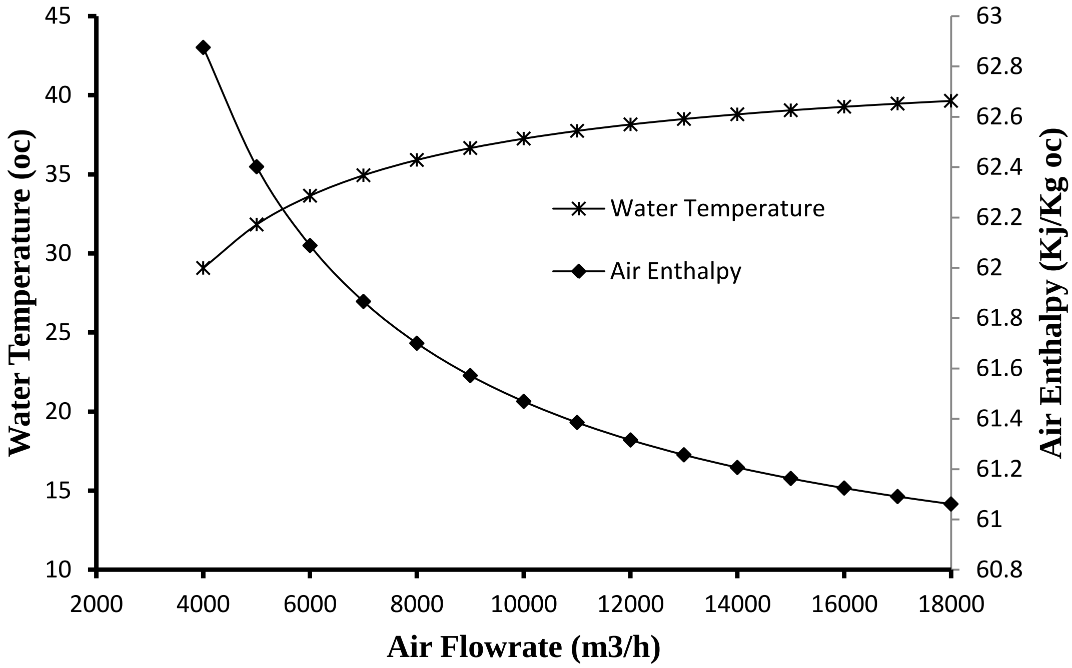

Figure 5: Variation of Water Temperature and Air Enthalpy with Air Flowrate

Figure 5 shows the effect of varying the water-to-air ratio on the performance of the cooling tower. A low flowrate of water of water means a high Water-to-Air loading ratio and vice versa since the flowrate of the water is kept constant. At an air flowrate of 4000 m^3/h, the loading ratio is 1:3.38, the outlet temperature of the cooling water is 29.1^{0}C and the enthalpy of the oulet air is 62.87 KJKg^{-1} ^{0}C^{-1}, while at an air flowrate of 18000 m^3/h, the loading ratio is 1:0.75, the outlet temperature of the cooling water is 39.64^{0}C and the enthalpy of the oulet air is 61.06 KJKg^{-1} ^{0}C^{-1}. From the above, it can be seen that at a low loading ratio, much cooling is not achieved. This is because the contact time between the water and the air is reduced. The air spends less time in the cooling tower. Since the flowrate is in volumetric flowrate, a low water-to-air loading ratio implies that the volume of air in the system is less than the volume of water. This reduces the performance of the cooling tower since it is the air that does the cooling. Figure 6 shows the relationship between the humidity ratio and the water-to-air loading rate. From Figure 6 it can be seen that the relative humidity is low at high water-to-air loading ratio. This is because more moisture is absorbed from the water, the enthalpy of the air increases and the air becomes dryer. From the foregoing, it is seen that for greater and effective performance, the water-to-air loading ratio should be such that there is sufficient air circulation in the system to cool the water.

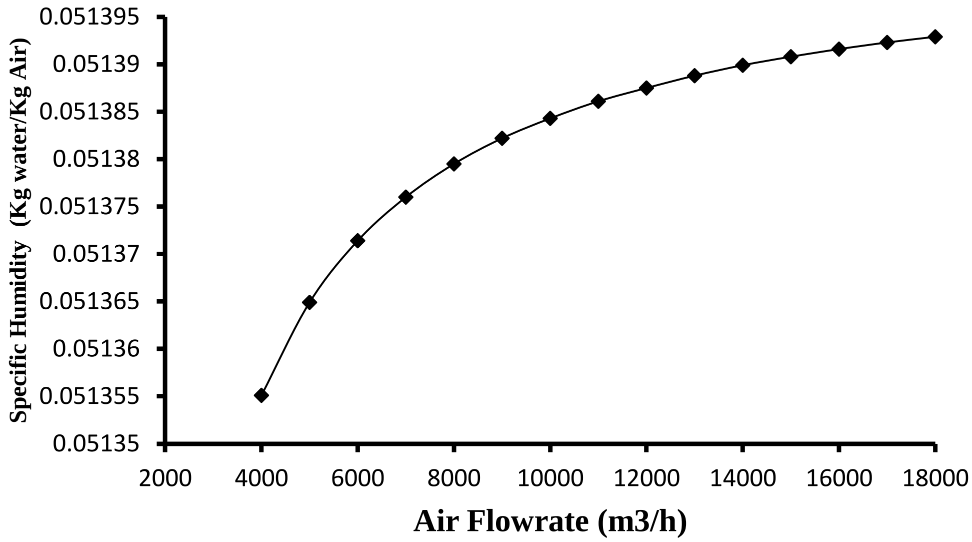

Figure 6: Relative Humidity with Air Flowrat

CONCLUSIONS

Cooling towers are very important in the normal operation of the refinery and other process industry where it is used to extract heat from the process fluid especially water before it is recycled. Since its operation is very crucial to other process equipment especially heat exchangers, it becomes imperative that its performance and operation be monitored.In this work a performance model for the cooling tower has been developed. The result obtained was validated using plant data from the Port Harcourt Refining Company of Nigeria. The results show good agreement with plant data with a deviation of 2% for the cooling water outlet temperature and 1.4% for the outlet air wet bulb temperature. Simulation results show that for the proper functioning of the cooling tower, the Water-to Air loading ratio should be greater than unity so that there will be sufficient air to extract the heat from the water.

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Nomenclature

Symbols	**Meaning**	**Unit**
a	air/water interfacial area per unit volume of tower	m^2/m^3
A	tower cross-sectional area	m^2
Cp_a	Specific heat of dry air at constant pressure	kJ/kgK
Cp_w	Specific heat of dry air at constant pressure	kJ/kgK
h	enthalpy	kJ/kg
h_c	Heat transfer coefficient of air	kW/m^2K
h_d	Mass transfer coefficient of water	Kg/m^2K
$h_{f,w}$	Enthalpy of saturated air evaluated at T_w	kJ/kg
$h_{fg,w}$	Enthalpy phase change	kJ/kg
$h_{g,w}$	Enthalpy of saturated liquid water evaluated at T_w	kJ/kg
H	Tower height	m
K_a	Tower characteristic	kg/m^3s
Le_f	Lewis factor $\left(Le_f = \dfrac{h_c}{h_d \rho_a Cp_a}\right)$	
P	Pressure	Kpa
R	Gas constant	kJ/kg/K
T	Temperature	^0C

T_{db}	Dry bulb temperature	^0C
T_{wb}	Wet bulb temperature	^0C
V	Volume of tower	m^3

Greek Symbols

ρ_a	Air density	kg/m^3
ω	Humidity ratio	kgw/kga
$\omega_{s,w}$	Saturated humidity ratio evaluated at T_w	kgw/kga
ϑ_a	Dry air volumetric flow rate	m^3/s
ϑ_w	Water volumetric flow rate	m^3/s

Subscripts

a	dry air
e	exit
i	inlet
w	water
s	saturate