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CFD Prediction of Forced Draft Counter-Flow Cooling Tower Performance

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Abstract

Numerical and experimental studies were conducted for open type forced draft water cooling tower. The numerical part includes a three dimensional computational solution of air and water simultaneous equations which represents the fluid flow, heat transfer and mass transfer. Finite volume method with staggered grid and k-ε turbulent model was used. Experimentally, mechanical forced draft counter-flow cooling tower was used to validate the numerical results. The agreement seems acceptable between the numerical and experimental results.

تنبأ CFD لاداء برج التبريد من نوع الجريان القسري المعاكس

الخلاصة

اجريت دراسة عددية وعملية لبرج تبريد مائي من النوع المفعوح ذو الجريان القسري . شملت الدراسة العددية الحسابات الثلاثية البعد للهواء والماء بصورة آنية والتي تجعل جريان المائع وانتقال الحرارة والكعلة .ا ستعلمت طريقة الحجوم المحددة مع نقاط شبكة مخالفة وموديل (K-2) الاضطرابي –عمليا استعمل برج تبريد مائي من النوع المفعوح ذو الجريان القسري لتقييم نفاذية النتائج العددية . التوافق بين النتائج العددية والعملية يبدو مقبولا .

1. Introduction

Cooling towers are commonly used to dissipate heat from water needed for condenser, heat exchanger, and other process equipment. A cooling tower cools water by a combination of heat and mass transfer. The hot water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill, which exposes very large water surface area to atmospheric air. A portion of the water absorbs heat and it is changed to a vapor at constant pressure. This latent heat has been long used to transfer heat from water to the atmosphere.

Robinson [1] was the first who considered the problem of cooling tower in 1922, others Walker, Lewis and McAdams [2], they developed the basic equations for heat and mass transfer by consider them separately. Majumdar, Spalding and Singhal [3] studied numerically the performance of natural and forced cooling tower in two-dimension. Abdullah [4] studied numerically the open type forced draft water cooling towers in two-dimension. Al-Saghar [5] conducted two-dimension study of numerical and experimental forced draft water cooling tower. Recently more studies were carried for simulation of cooling

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tower by CFD, Meroney [6], Wang [7], Williamson [8], Fisenko [9], Rafat [10].

Apparently, it seems more numerical simulation is needed especially in three-dimensional. In this work, a three-dimensional numerical simulation was conducted in addition to experimental studies to verify the numerical results.

2. Numerical Investigation

In order to predict the thermal performance of the cooling tower, it is required to build a computational simulation system that needs:

a) Physical model to express resistance to air flows and interfaces heat and mass transfer.

b) Mathematical model which provides an accurate solution of the conservation equations of mass, momentum and energy.

2.1 Physical Model

In cooling tower, water is cooled by evaporation of part of the water into air. This cooling effect is either assisted or obated by simultaneously convective heat transfer between water and air. In a counterflow cooling tower water flows downwards and air streams upward. A simplifying approximation of Merkel's total heat theory has been almost universally adopted for the calculation of tower performance. Merkel's theory states that all of the heat transfer taking place at any position in the cooling tower is proportional to the difference between the total heat of the air at that point in the tower, and the total heat of air saturated at the temperature of the water at that point in the tower.

As an equation, the above statement would be written:

$$q^{.///} = Ka\left(h_{sw} - h_a\right) \quad .....(1)$$

Where Ka is an empirical mass transfer coefficient and can be determined from experimental work, and (h_{sw} – h_a) is the difference between the enthalpies of the saturated air and dry air. An expression of evaporation rate, $m_v^{.///}$ consistent with Equation 1 is:

$$m_V^{.///} = Ka\left(w_{SW} - w_a\right) \quad .....(2)$$

The flow resistance offered by various solid obstacles and water flow within the tower are expressed for each control cell in the following integrated form:

$$\int f_x dV = N_V \frac{ru^2}{2} \Delta V \quad .....(3)$$

$$\int f_y dV = N_V \frac{rv^2}{2} \Delta V \quad ....(4)$$

$$\int f_z dV = N_V \frac{rw^2}{2} \Delta V \quad .....(5)$$

Where ΔV is the volume, and N_V is the total number of velocity heads lost in the louver and eliminator.

2.2 Mathematical Model

The present model treats airflow to be steady, three dimensional, turbulent and incompressible flows, while the water flow is considered to be one-dimensional. The cooling tower is a forced draft counterblow type, in which air passes upward through a falling spray of water. Figure 1 shows the geometric shape of the tower. The governing equations are:

1. Continuity equation (Mass of air)

$$\frac{\partial}{\partial x}(ru) + \frac{\partial}{\partial y}(rv) + \frac{\partial}{\partial z}(rw) = m_v^{///} \ldots(6)$$

2. Continuity equation (Mass of water)

$$\frac{\partial}{\partial y}(r_F u_F) = m_v^{.///} \qquad ...(7)$$

3. X-Direction air momentum equation
4.

$$\frac{\partial}{\partial x}(ru^2)+\frac{\partial}{\partial y}(rvu)+\frac{\partial}{\partial z}(rwu)=2\{\frac{\partial}{\partial x}(m_{eff}\frac{\partial u}{\partial x})\}+\frac{\partial}{\partial y}(m_{eff}\frac{\partial u}{\partial y})+\frac{\partial}{\partial z}(m_{eff}\frac{\partial u}{\partial z})+$$
$$\frac{\partial}{\partial y}(m_{eff}\frac{\partial v}{\partial x})+\frac{\partial}{\partial z}(m_{eff}\frac{\partial w}{\partial x})-\frac{\partial p}{\partial x}-f_x ..(8)$$

4. Y-Direction air momentum equation

$$\frac{\partial}{\partial x}(ruv)+\frac{\partial}{\partial y}(rv^2)+\frac{\partial}{\partial z}(rwv)=\frac{\partial}{\partial x}(m_{eff}\frac{\partial v}{\partial x})+2\{\frac{\partial}{\partial y}(m_{eff}\frac{\partial v}{\partial y})\}+\frac{\partial}{\partial z}(m_{eff}\frac{\partial v}{\partial z})+$$
$$\frac{\partial}{\partial x}(m_{eff}\frac{\partial u}{\partial y})+\frac{\partial}{\partial z}(m_{eff}\frac{\partial w}{\partial y})-\frac{\partial p}{\partial y}-f_y$$
$$-(r-r_{amb})g \qquad ....(9)$$

5. Z-Direction air momentum equation

$$\frac{\partial}{\partial x}(ruw)+\frac{\partial}{\partial y}(rvw)+\frac{\partial}{\partial z}(rw^2)=\frac{\partial}{\partial x}$$
$$(m_{eff}\frac{\partial w}{\partial x})+\frac{\partial}{\partial y}(m_{eff}\frac{\partial w}{\partial y})+2\{\frac{\partial}{\partial z}(m_{eff}\frac{\partial w}{\partial z})\}+$$
$$\frac{\partial}{\partial x}(m_{eff}\frac{\partial u}{\partial z})+\frac{\partial}{\partial y}(m_{eff}\frac{\partial v}{\partial z})-\frac{\partial p}{\partial z}-f_z ...(10)$$

6. Air enthalpy equation

$$\frac{\partial}{\partial x}(ruh_a)+\frac{\partial}{\partial y}(rvh_a)+\frac{\partial}{\partial z}(rwh_a)-\frac{\partial}{\partial x}(\Gamma_{eff}$$
$$\frac{\partial h_a}{\partial x})-\frac{\partial}{\partial y}(\Gamma_{eff}\frac{\partial h_a}{\partial y})-$$
$$\frac{\partial}{\partial z}(\Gamma_{eff}\frac{\partial h_a}{\partial z})=q^{.///} \qquad ....(11)$$

7. Moisture fraction of air equation

$$\frac{\partial}{\partial x}(ruw_a)+\frac{\partial}{\partial y}(rvw_a)+\frac{\partial}{\partial z}(rww_a)-\frac{\partial}{\partial x}$$
$$(\Gamma_{eff}\frac{\partial w_a}{\partial x})-\frac{\partial}{\partial y}(\Gamma_{eff}\frac{\partial w_a}{\partial y})-$$
$$\frac{\partial}{\partial z}(\Gamma_{eff}\frac{\partial w_a}{\partial z})=m_v^{.///} \qquad ....(12)$$

8. Water enthalpy equation

$$\frac{\partial}{\partial y}(r_F u_F h_W) = -q^{.///} ....(13)$$

Equation of state should be used because the air density various along the cooling tower

$$r = \frac{p.w_w}{R(t_{adb}+273)} \qquad ...(14)$$

$$m_{eff} = r\, c_m \frac{k^2}{e} \qquad ...(15)$$

$$\Gamma_{eff} = m_{eff} / s_{eff} \quad ..(16)$$

Where σ_{eff} is the effective Prandtl number and has been assumed to be unity. The standard model uses the following transport equations used for k and ε.

$$\frac{\partial}{\partial x}(ruk)+\frac{\partial}{\partial y}(rvk)+\frac{\partial}{\partial z}(rwk)=\frac{\partial}{\partial x}(\Gamma_k\frac{\partial k}{\partial x})+\frac{\partial}{\partial y}(\Gamma_k\frac{\partial k}{\partial y})+\frac{\partial}{\partial z}(\Gamma_k\frac{\partial k}{\partial z})+m_t\begin{bmatrix}2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+2(\frac{\partial w}{\partial z})^2+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2\\+(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x})^2+(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y})^2\end{bmatrix}-re \quad (17)$$

$$\frac{\partial}{\partial x}(rue)+\frac{\partial}{\partial y}(rve)+\frac{\partial}{\partial z}(rwe)=\frac{\partial}{\partial x}(\Gamma_e\frac{\partial e}{\partial x})+\frac{\partial}{\partial y}(\Gamma_e\frac{\partial e}{\partial y})+\frac{\partial}{\partial z}(\Gamma_e\frac{\partial e}{\partial z})+c_{1e}\frac{e}{k}m_t\begin{bmatrix}2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+2(\frac{\partial w}{\partial z})^2\\+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2\\+(\frac{\partial u}{\partial z}+\frac{\partial u}{\partial x})^2+(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y})^2\end{bmatrix}-c_{2e}r\frac{e^2}{k} \quad (18)$$

The recommended values of the empirical constants and functions are given in Table 1. These values represent what is considered the standard (k - ε) model.

The general form of the governing equations.

$$\frac{\partial}{\partial x}(ruf)+\frac{\partial}{\partial y}(rvf)+\frac{\partial}{\partial z}(rwf)=\frac{\partial}{\partial x}(\Gamma_f\frac{\partial f}{\partial x})+\frac{\partial}{\partial y}(\Gamma_f\frac{\partial f}{\partial y})+\frac{\partial}{\partial z}(\Gamma_f\frac{\partial f}{\partial z})+S_f \quad ....(19)$$

Where ϕ is the dependent variable which may be a vector quantity such as velocity components (u,v,w) or scalar quantity such as temperature, air moisture fraction (ω_a), or air enthalpy (h_a). The source term (S_ϕ) means the source of heat transfer, mass transfer or pressure variation that allows fluid to flow, Γ_ϕ is the diffusion coefficient which is a dynamic viscosity (μ_{eff}) in momentum equation but effective exchange coefficient (Γ_{eff}) in enthalpy and moisture fraction equations.

Most investigators used the hybrid method[6] (central / upwind differencing), for solving the transport equation, SIMPLE algorithms has been used in the present study. Staggard grids are used. For full details of descritization of governing equations refer to Mehdi [7].

The following variable quantities are required as initial and boundary condition.

1. The velocity of inlet mass flow rate of water ($\rho_F u_F$).
2. The inlet velocity of air in x-direction (u).
3. Inlet water temperature.
4. Inlet air wet and dry bulb temperatures.

3. Experimental Investigation

The experimental work was carried out using available mechanical forced draft counterblow cooling tower (Hilton water cooling tower), shown in Figure 2. The tower was equipped with several measuring devices to express the condition of water and air at inlet, outlet and other five stations along the tower. The tower was equipped also with four heaters (2.5 kW each) to heat the water and that represent the load on it.

A minimum flow of 350 kg/h of clean water at main temperature is required to operate the tower. Air inlet and outlet dry and wet bulb temperatures are measured respectively by means of thermometers. Air dry and wet bulb temperatures at five stations along the tower are measured by means of psychometric gun.

The mass flow rate of air is measured by using an orifice plate and associated ducting and manometer. The air blower is used to draft air inside the tower. The thermometers, which used in the experimental work, are high accuracy quartz thermometers.

The Merkel equation can be written in empirical from,

$$\frac{KaV}{\dot{m}_w} = l\,(\frac{\dot{m}_w}{\dot{m}_a})^{-n} \qquad ...(20)$$

The constants λ and n depend on the packing design. Table 2 shows the numerical value of (λ and n) with progressive height of packing at each measuring station.

4. Results And Discussion

4.1 Experimental Results

Numerous experiments are made to show the effect of the following factors on the tower characteristic or the number of transfer units (NTU)

1. Mass flow rate of water.
2. Mass flow rate of air.
3. Cooling range.
4. Inlet air wet bulb temperature and tower approach.
5. Packing height (cooling tower volume).

The thermal capability of any cooling tower may be defined by some parameters; one of these parameters is water flow rate. The latent heat of vaporization has long been used to transfer heat to the atmosphere.

The falling water could be made to splash into droplets to increase the surface area exposed to the air. New water was added to replace that lost to evaporation, the water was continuously recirculated over the surface.

Many laboratory experiments are carried out to show the effect of mass flow rate of water on the tower characteristic (NTU). Figure 3 shows experimentally that ($KaV/\dot{m}_w$) is decreased with the increment of ($\dot{m}_w/\dot{m}_a$) value under different packing height.

Figure 4 illustrates the relation of the volumetric mass transfer coefficient (Ka) against ($\dot{m}_w/\dot{m}_a$) at constant air mass flow rate. It is clear that increasing water mass flow rate increases the volumetric mass tower characteristics and this lead to decrease the value of tower characteristics.

The influence of air mass flow rate on volumetric mass transfer coefficient (Ka) at constant water mass flow rate ($\dot{m}_w$), is depicted in Figure 5. It can be observed that (Ka) increases with increasing air mass flow rate, which means increased evaporative from water into air stream, and this leads to increase the value of tower characteristics. In other words, the decrease of ($\dot{m}_w/\dot{m}_a$) for the same water flow rate means the decrease of enthalpy in the air-side and a value of $1/(h_{sw} - h_a)$ is consequently decreased as shown in Figure 6. It can be observed from Figure 7 that high air flow rate gives low approach which leads to increase the NTU.

Tower approach means the difference between the outlet water temperature and inlet air wet bulb temperature ($t_{wo} - t_{awbi}$). Figure 8 shows the variation of tower approach with the number of transfer units (NTU's) of the tower under the same design conditions. The approach of the tower is found to increase with decrease in NTU.

Figure 9 illustrates the predicted performance curves for the Hilton water-cooling tower. It is shown that cold water temperature increases with the increment in air wet bulb temperature, and decreases with the increment of volume flow rate of air.

4.2 Validation of the Code

In order to verify the computer program results, which represent a numerical simulation for a counter-flow cooling tower, a comparison between the numerical and experimental results are made for different properties as shown in Figure 10. Figure 10 compares between the experimental and numerical air enthalpy along the tower height for different volumetric mass transfer coefficient. The air enthalpy increases gradually along the tower height. This increment is due to the heat transfer from warm water to the bulk air. acceptable agreement was observed.

4.3 Numerical Results

In this section, the numerical results are presented for flow field in the cooling tower and other properties in three dimensions for the following two cases

a) "t_{adb} = 44°C, t_{awb} = 26°C , t_{wi} =50°C, Ka =0.322"
b) "t_{adb} = 42°C, t_{awb} = 23°C , t_{wi} = 46.5°C, Ka =0.416"

Figures 11 to 13 show the distribution of different parameters through the cooling tower starting from flow field, air enthalpy, air specific humidity. Also the results of using two different types of packing on the properties of air and water, also the rate of heat and mass transfer are presented.

Figures 14 and 15 illustrate the variation of air enthalpy and air moisture content respectively along the tower height for two types of packing. The first type is aluminum fill. The characteristic equation is,

$$\frac{KaV}{\dot{m}_w} = 0.371\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^{-0.626}$$

The second type is ceramic fill. The characteristic equation is [8],

$$\frac{KaV}{\dot{m}_w} = 0.199\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^{-0.592}$$

The numerical result shows that the aluminum fills is more effective than ceramic fill in heat and mass transfer between the water and the bulk air. The value of air enthalpy is increased (14.5%) when aluminum fill is used; also the value of air moisture content

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NOTATIONS

$C_{\mu}, C_{1\varepsilon}, C_{2\varepsilon}$	Turbulent empirical constants	
f_x, f_y, f_z	Resistance to air flow in x-,y-,and z-direction respectively	(N/m^3)
h_a	Enthalpy of air-water vapour mixture at wet bulb temperature	(kJ/kg_{da})
h_w	Specific enthalpy of water	(kJ/kg)
k	Turbulent kinetic energy	(m^2/s^2)
Ka	Volumetric mass transfer coefficient	($kg/m^3.s$)
$\frac{KaV}{\dot{m}_w}$	Tower characteristics based on $\dot{m}_w$	-
$\frac{KaV}{\dot{m}_a}$	Tower characteristics based on $\dot{m}_a$	-
$\dot{m}_w$	Mass flow of water per unit plan area of packing	($kg/m^2.s$)
$\dot{m}_a$	Mass flow of water per unit plan area of packing	($kg/m^2.s$)
m'''_v	Rate of mass transfer per unit volume	($kg/m^3.s$)
n	Constant depend on the packing design	
N_v	Total number of velocity heads lost in the louver and eliminator	
NTU	Number of transfer units	
P	Pressure	(kN/m^2)
$\dot{q}'''$	Rate of heat transfer per unit volume	(W/m^3)
R	Universal gas constant	($J/k_{mol}.k$)

T	Time	(s)
t_{wi}	Hot water temperature	(^oC)
t_{wo}	Cold water temperature	(^oC)
t_{awb}	Air dry bulb temperature	(^oC)
t_{adb}	Air dry bulb temperature	(^oC)
u_F	Water velocity	(m/s)
u,v,w	Air velocity component in x-,y-,and z-direction respectively	(m/s)
V	Active cooling volume per unit plan area of packing	(m^3/m^2)
ω_{sw}	Moisture fraction of saturated moist air	(kg/k_{gda})
ω_a	Moisture fraction of saturated moist air	(kg/k_{gda})
Z	Packing height	(m)

Subscripts

P	Control point
x, y, z	Direction

Greek symbols

ρ	Density of moist air	(kg/m^3)
ρ_F	Density of moist water	(kg/m^3)
ρ_{amb}	Density of moist air	(kg/m^3)
$\sigma_k, \sigma_\varepsilon$	Constants for the $k-\varepsilon$ model	-
ε	Turbulent energy dissipation rate	(m^2/s^3)
Γ_ε	Diffusion coefficient for dissipation rate equation	(kg/m.s)
Γ_k	Diffusion coefficient for dissipation rate equation	(kg/m.s)
Γ_{eff}	Effective exchange coefficient	(kg/m.s)
ϕ	Dependent variable	

μ	Dynamic viscosity	(kg/m.s)
μ_t	Turbulent dynamic viscosity	(kg/m.s)
μ_{eff}	Effective viscosity	(kg/m.s)
σ_{eff}	Effective Prandtl number	-
λ	Constant depend on the packing design	-

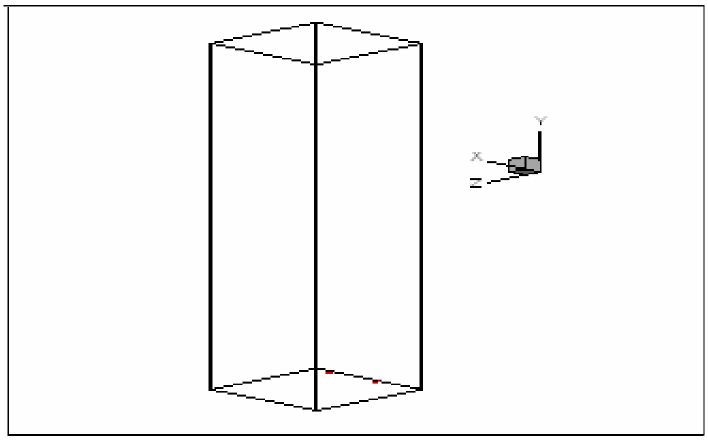


Figure (1) Geometric Shap0e of the Tower

Figure (2) Schematic layout of Hilton forced draft water cooling tower

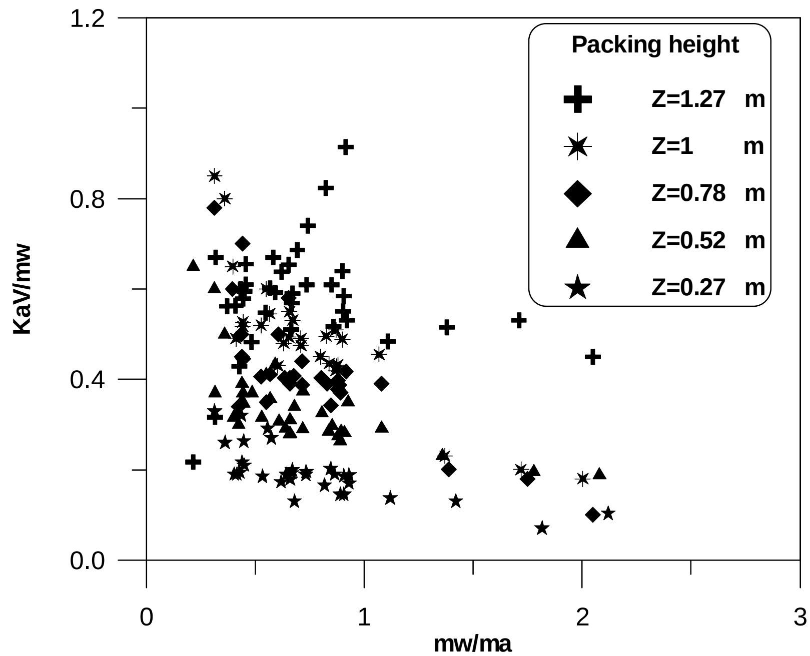


Figure (3) Tower characteristic curves

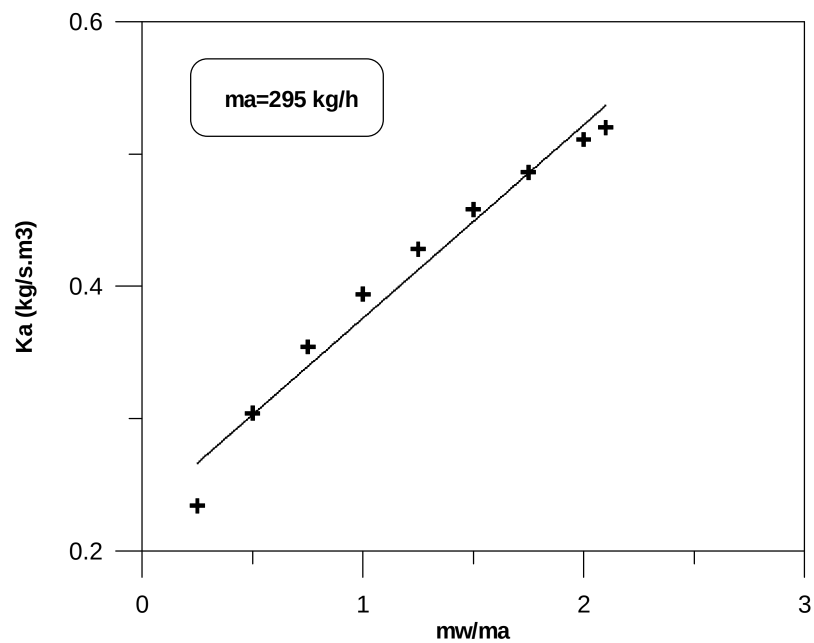


Figure (4) Variation of volumetric mass transfer coefficient with mw/ma

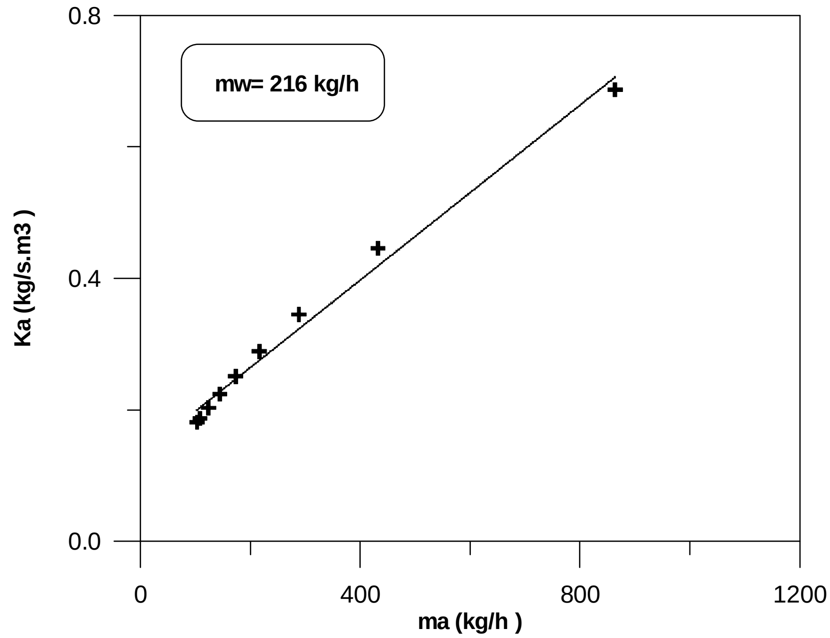


Figure (5) Variation of volumetric mass transfer coefficient with air massflowrate

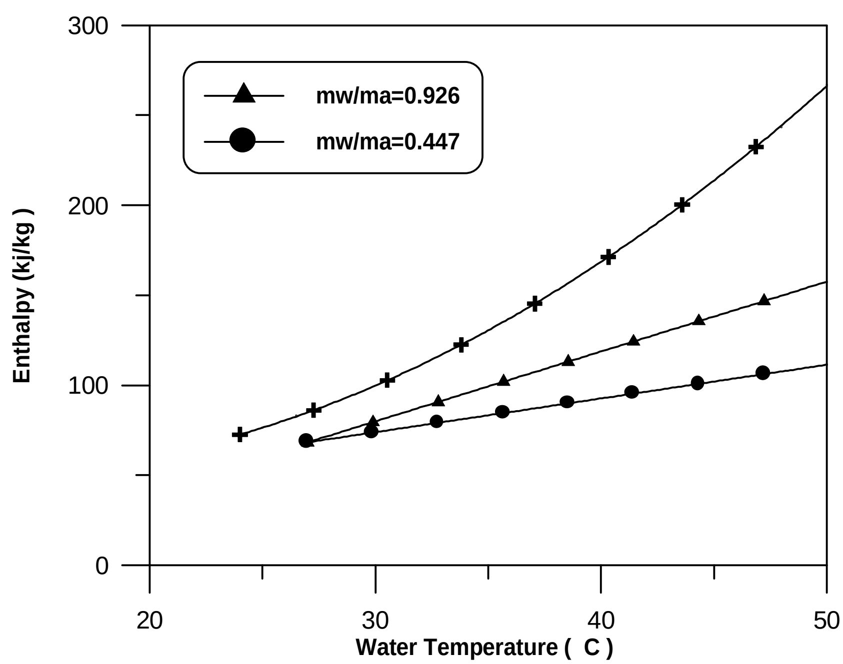


Figure (6) Effect of (mw/ma)variation on thermal process

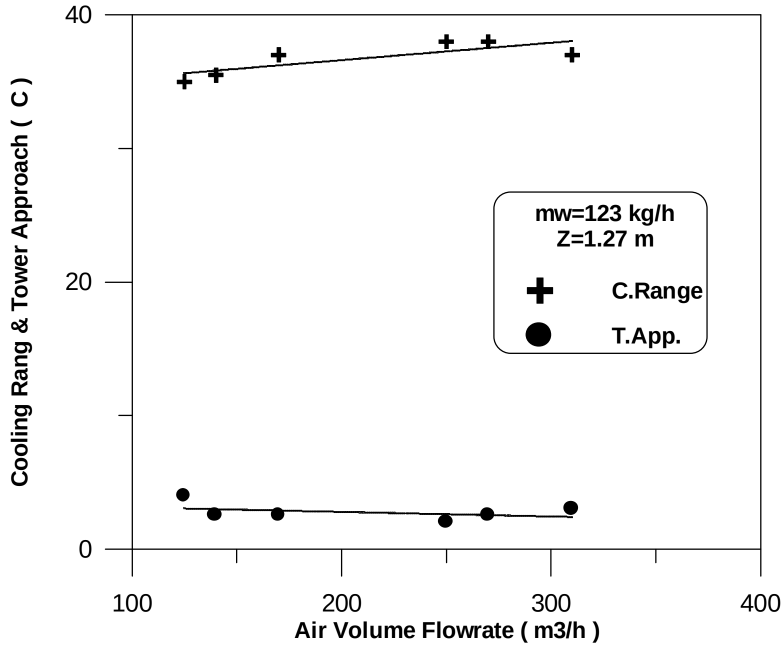


Figure (7) Variation of cooling range and tower approach with different volume flow rate of air

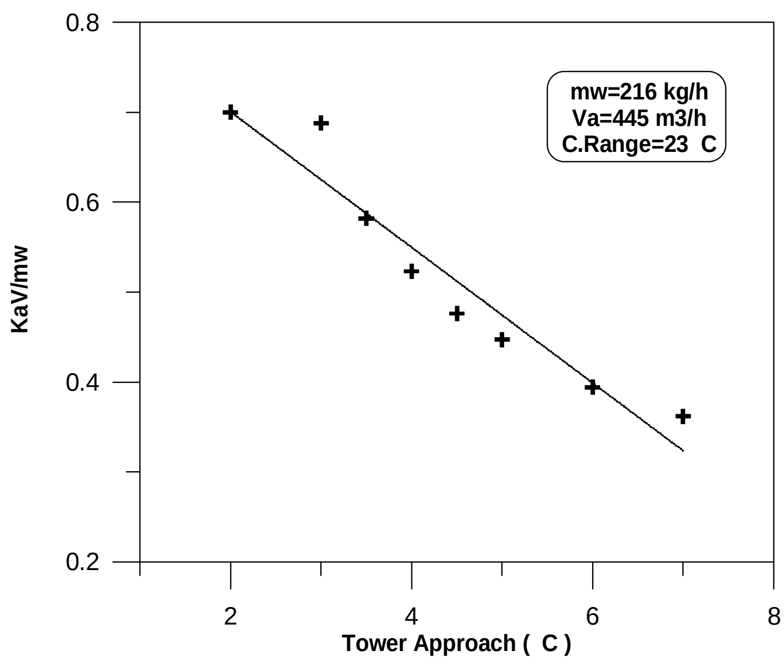


Figure (8) Variation of NTU with tower approach

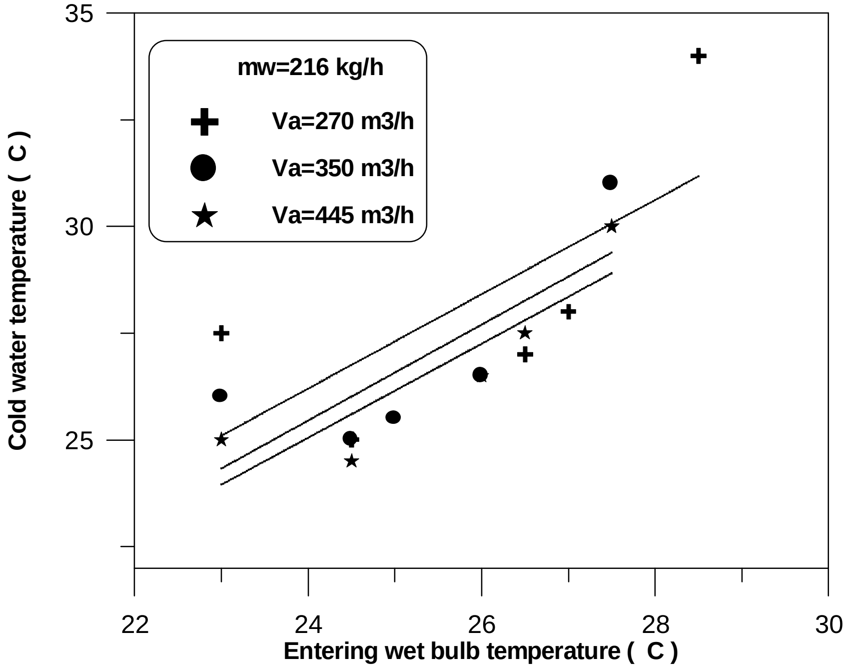
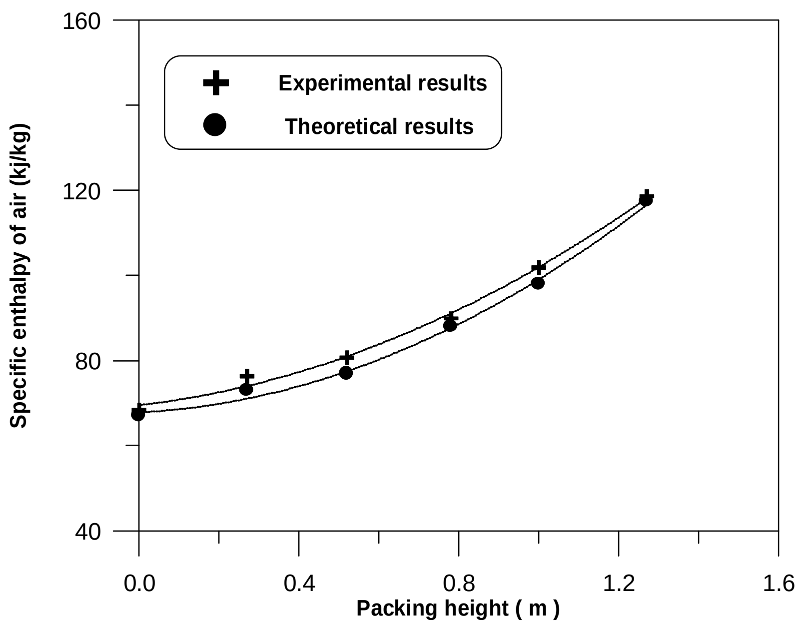
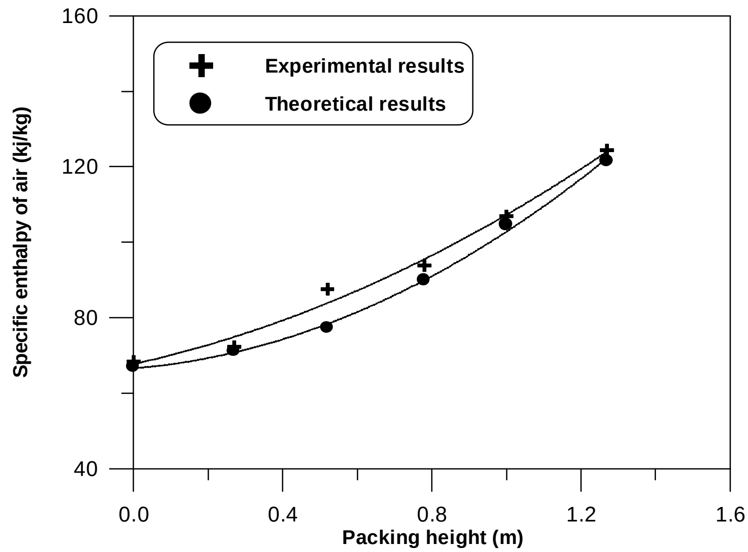


Figure (9) Performance curves for a Hilton water cooling tower

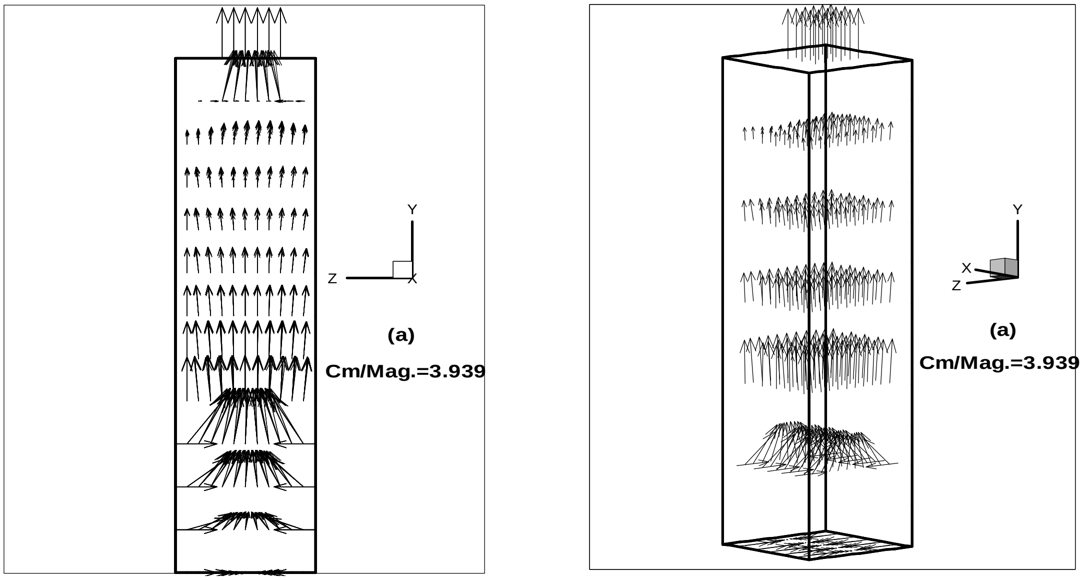


(a) t_{adb}=40.5 ^{o}C,t_{awb}=23^{o}C,t_{wi}=45.5^{o}C,Ka=0.459

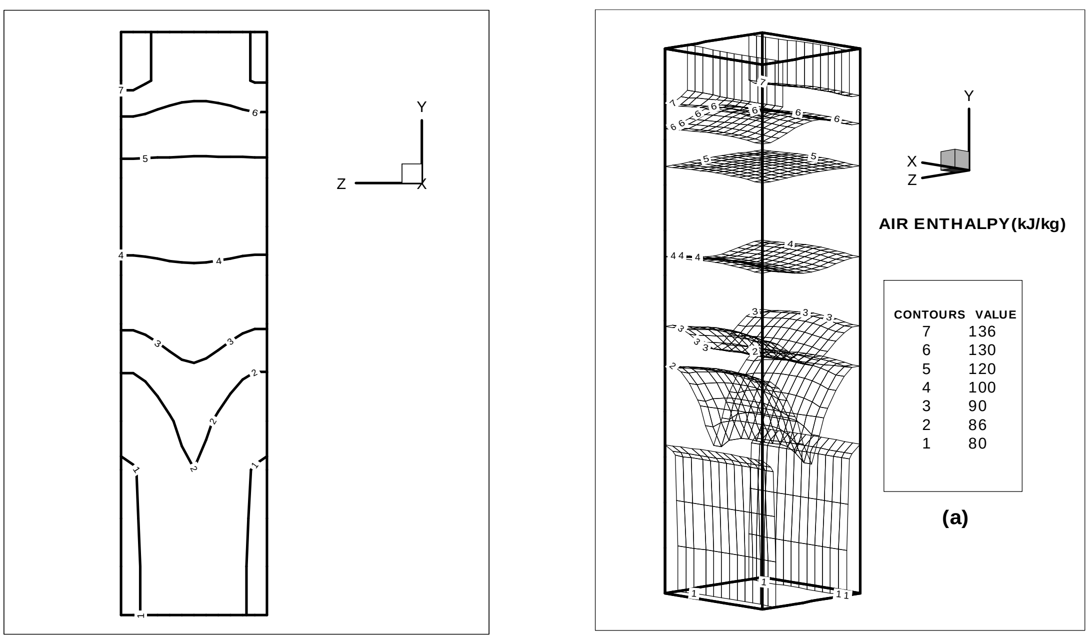


(b) t_{adb}=44.5 ^{o}C,t_{awb}=23^{o}C,t_{wi}=48^{o}C,Ka=0.407

Figure (10) The Theoretical and Experimental Variation of air Enthalpy Along Cooling Tower height

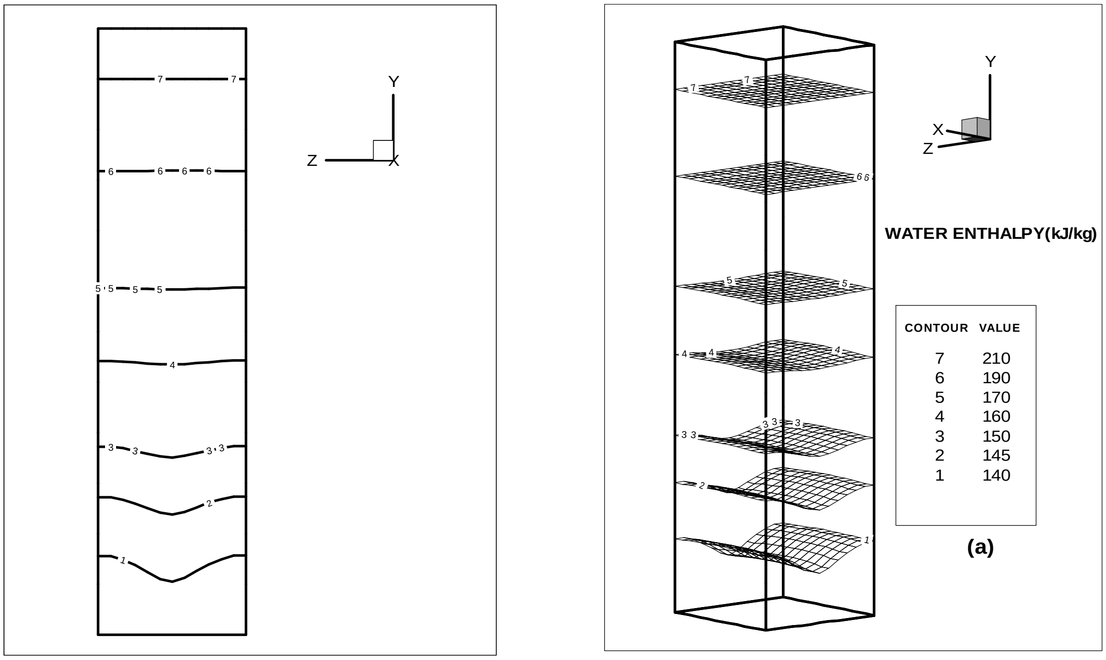


a) t_{adb}=44^{o}C,t_{awb}=26^{o}C,t_{wi}=50^{o}C,Ka=0.322

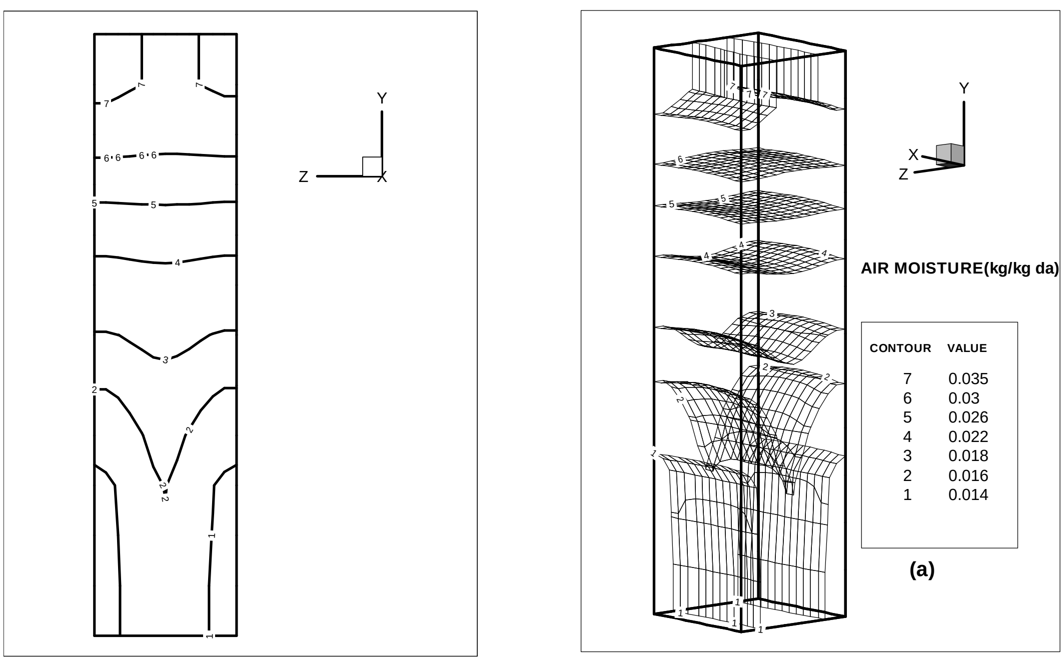


b t_{adb}=42^{o}C,t_{awb}=23^{o}C,t_{wi}=46.5^{o}C,Ka=0.416

Figure (11) Air velocity vectors in a mechanical forced draft counterflow tower

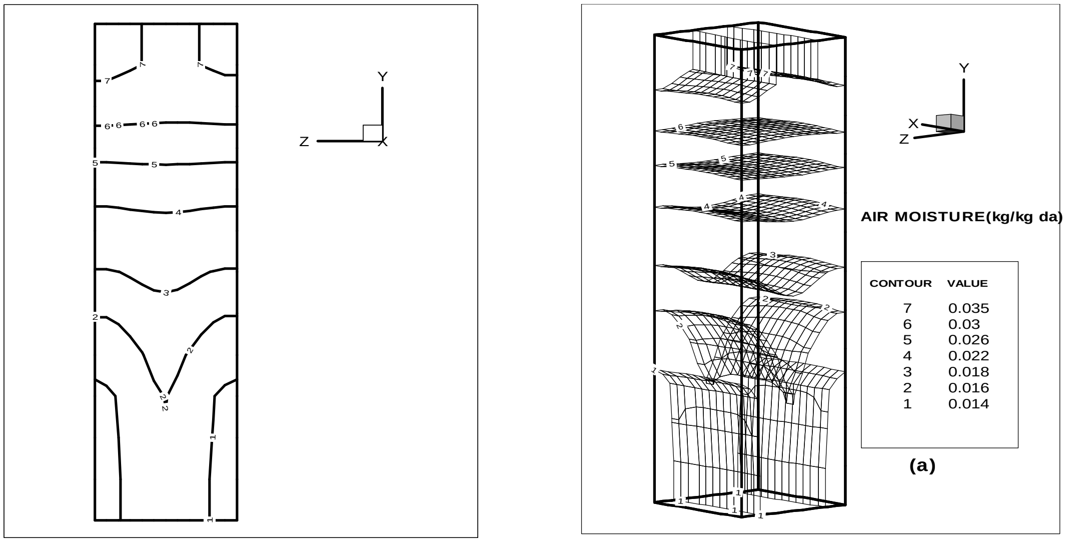


a) $t_{adb}=44^{o}C, t_{awb}=26^{o}C, t_{wi}=50^{o}C, Ka=0.322$

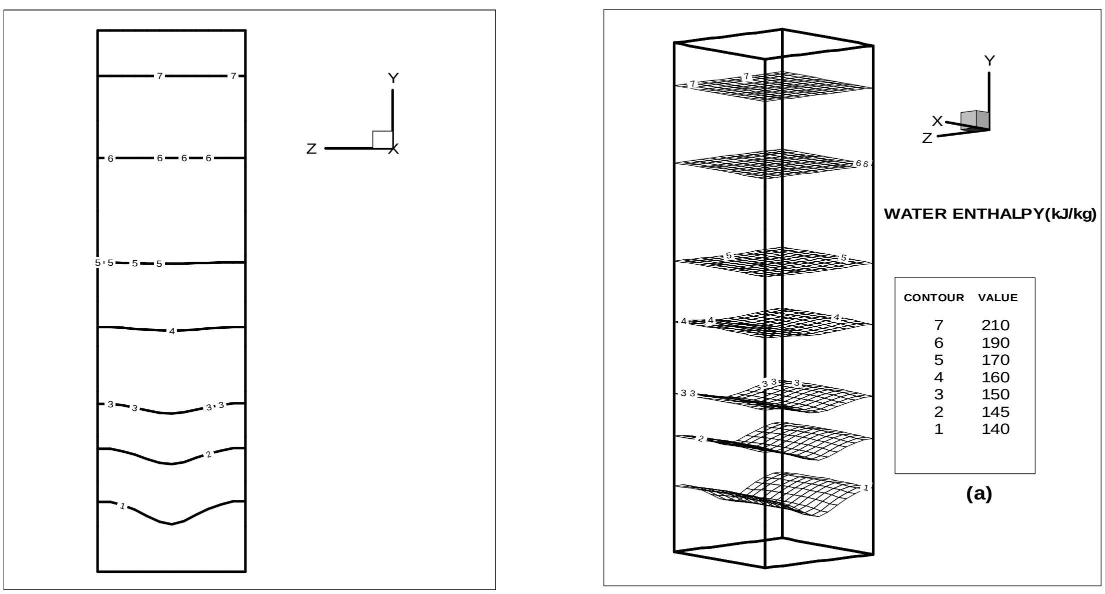


b) $t_{adb}=42^{o}C, t_{awb}=23^{o}C, t_{wi}=46.5^{o}C, Ka=0.416$

Figure 12 Air enthalpy contours in a mechanical forced draft counterflow tower



a) t_{adb}=44°C,t_{awb}=26°C,t_{wi}=50°C,Ka=0.322



b) t_{adb}=42°C,t_{awb}=23°C,t_{wi}=46.5°C,Ka=0.416

Figure (13) Air specific humidity contours in a mechanical forced draft counterflow tower

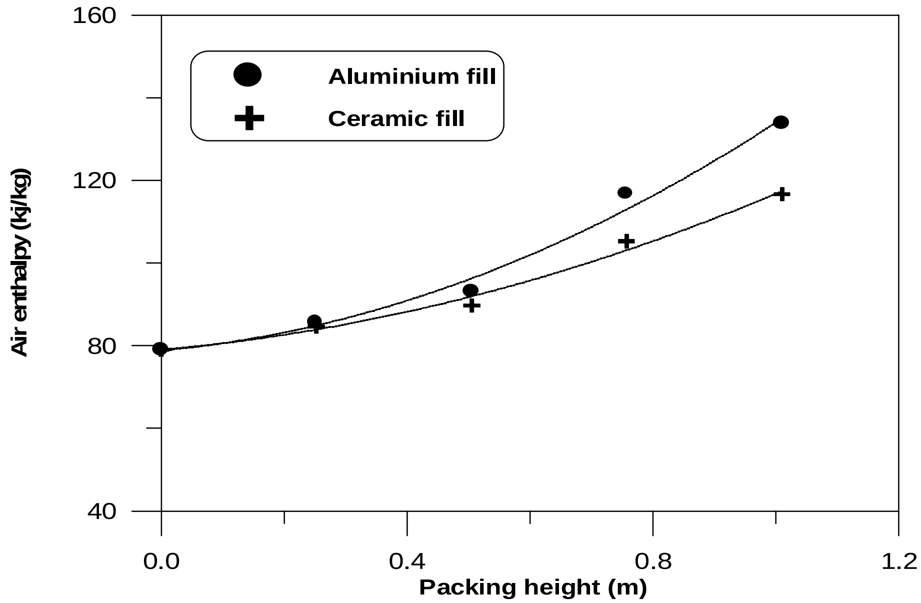


Figure (14) Variation of air enthalpy at different types of packing along cooling

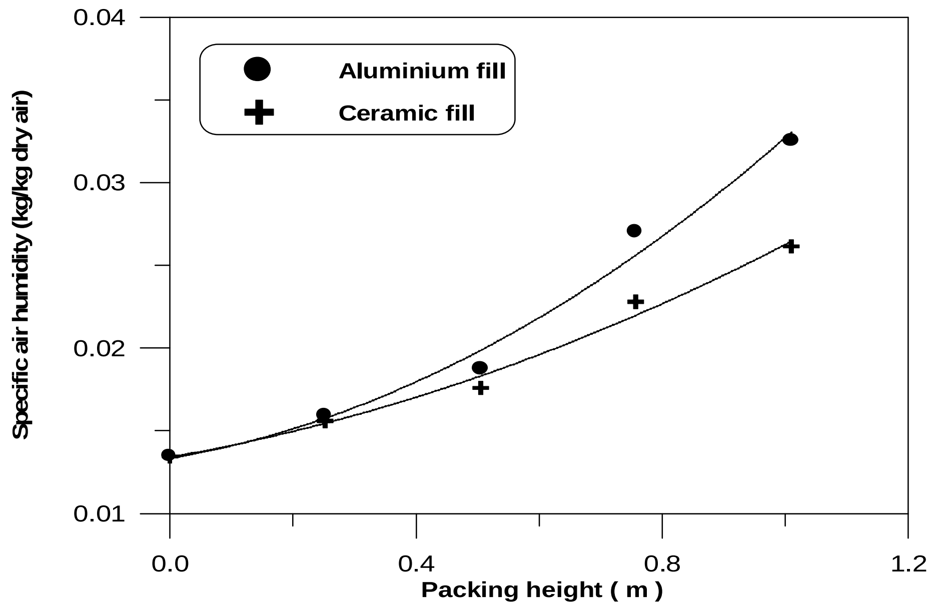


Figure (15) Variation of air moisture content at different types of packing tower along