Performance of Cooling Tower with Honeycomb Packing

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Abstract

The objective of this study is to investigate experimentally and theoretically heat and mass transfer characteristics of the cooling tower. The investigation carried out at a mechanical forced direct cooling tower. A column-packing unit is made of a new type of packing named honeycomb (plastic). Air and water are used as fluids and the runs are done at the air and water mass flow rates ranging between 0.05 and 0.15 kg/s, and between 0.1 and 0.25 kg/s, respectively. The inlet water temperatures ranging between 35 and 50 ^{o}C. A mathematical model based on the equations of mass transfer and energy is used and solved to determine the characteristics of cooling tower, pressure drop, temperature ratio, and tower effectiveness. There is reasonable agreement from the comparison between the calculated and measured data.

Keywords: tower effectiveness, honeycomb, evaporative cooling, heat, mass

أداء برج التبريد باستخدام حشوة خلية النحل

الخلاصة

الهدف من هذا البحث هو دراسة انتقال المادة والحرارة عمليا ونظريا في برج تبريد تبخيري الحصول على مجموعة من النتائج العملية والنظرية لمعاملات انتقال المادة و الحرارة لبرج تبريد. والتجارب تمت باستعمال برج تبريد ميكانيكي و باستعمال نوع جديد من الحشوة يطلق عليه تسمية مستنبطة من شكله و يدعى حشوة قرص (خلية) النحل.الموائع المستخدمة في التجارب هما الماء و الهواء و بمعدلات جريان تتراوح بين 0.15 - 0.05 كلغم \ ثا و 0.1- 0.25 كلغم \ثا على التوالي. أما درجات الحرارة التي استخدمت بالتجارب فتراوحت بين 50-35 درجة مئوية .تم استخدام موديل رياضي يعتمد معادلات انتقال المادة والطاقة لحل و أيجاد معاملات الخاصة ببرج التبريد و كذلك لحساب هبوط الضغط و نسبة التغير بالحرارة و فعالية البرج , و قد بينت النتائج المحسوبة من الموديل توافقا" كبيرا" مع النتائج المقاسة .

Introduction

Cooling tower have been introduced as of the most of the direct contact heat exchangers and the most used widely in several heat transfer and mass transfer applications, for example chemical process, petrochemical process, power generation units, and air conditioning processes. Fisenko et al. [1], developed a mathematical model for predicting the performance of a nature draft cooling tower. The calculated results were validated by the measured data. Prasad [2] Applied the novel numerical and experimental techniques to determine the performance of the multi-cell cross flow evaporative cooling tower. Fisenko et al. [3] Developed the mathematical model of a mechanical draft cooling tower performance. The model represented a boundary-value problem for a system of ordinary differential equations.

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Khan et al.[4], presented mathematical modeling of cooling towers incorporating fouling growth model , besides considered effect of pressure and fouling on thermal cooling tower performance. Karami and Heidarinejad [5], developed heat and mass transfer characteristic of wet counter-flow cooling tower. They presented by increasing in mass flow ratio, tower effectiveness is increased but temperature ratio is decreased. Poppe and Rogener [6], developed a new model for cooling towers which did not use the simplifying assumptions made by Merkel, in their study different packing are studying. De Villiers and Kroger [7],developed relations for various geometries and configurations and explained that the mass transfer relation could be calculate an effective drop diameter, a diameter that would have the same effect as the actual ensemble of drops in the tower. Kloppers ana Kroger [8], investigated the effect of the Lewis factor, or Lewis relation, on the performance prediction of natural draft and mechanical draft wet-cooling towers. They found that if the same definition of Lewis factor is employed in the fill test analysis and in the subsequent cooling tower performance analysis, the water outlet temperature would be accurately predicted. Yan [9], investigated the rate of vaporization or condensation of the water vapor on the wetted channel walls in laminar mixed flows under the simultaneous of combined bouncy effects of thermal and mass diffusion. Lemouari et al. [10], study the performance of a forced counter-current flow cooling tower with grid type packing ,the effect of air and water flow rates on the water temperature range was studied. Lijuan[11] developed a new model based on the double film theory for air-cooling towers thermodynamic calculation. Rafat[12],Investigated numerically the effect of wind break walls on the thermal performance of natural draft wet cooling tower (NDWCT)under crosswind. Gharagheizi et al. [13] presented an experimental and comparative study on the performance of mechanical cooling tower with two types of film packing; they used vertical and horizontal corrugated packing and reported that the performance of the cooling tower is affected by air/water mass flow ratio. Boumaza et al. [14] Used vertical grid apparatus type of packing in an evaporative cooling system to study its thermal and hydraulic performances. This type of packing consists of vertical grids disposed between walls in the form of zigzag. El-Dessouky [15] studied the thermal and hydraulic performances of three-phase fluidized bed cooling tower .He used spongy rubber ball with density of 375 kg/m^3 as a packing and developed a correlation for calculation tower characteristic. Bedekar et al.[16] studied experimentally the performance of counter flow packed bed mechanical cooling tower, using a film type packing, they concluded that the tower performance decrease with an increase in the L/G ratio. Bender [17], investigated the effect of crosswinds on a double-cell mechanical induced cooling tower. They examined the flow over a prototype mechanical induced cooling tower. Merkel [18], simplified the complexity of simultaneous heat and mass transfer by assumed that Lewis factor equal to unity. This assumption has been generally accepted in theoretical analyses and cooling tower design. Kloppers [19] developed a model for counter flow wet-cooling towers using new assumption. Maclaine[20], developed an analytical for wet surface heat exchangers by analogy from conventional solutions for dry surface heat exchangers. Jalal and Waheed [21] studied the theoretical and experimental conducted on forced draft water cooling tower. In such towers, the heat and mass transfer take place from the hot water to the bulk air, which passes through the tower. The theoretical study includes two parts, the first part describes the numerical solution for the water cooling tower governing equations, a two dimension air momentum equation (Navier-Stocks equations) and air enthalpy equation (energy equation), moisture content and water enthalpy equation.

Experimental installation

The parts of the experimental installation have shown in figure(1),are numerated from 1 to15. The basis of the installation is the cooling tower(1),1.5 m height and 35 x 35 cm outside cross section. The tower construction structure is made of plastic, and the front is made of plexiglass plates 5 mm thick, the front plexiglass plate is removable, so the easy access to interior of tower is able in order to replace packing or from maintenance, and to enable the access of various measuring probes. Heating of water up to the wanted temperature has been carried out by means of five electrical heaters (2), each 2.5 kw of power, the temperature of water controlled by regular (11). The heated water is pumped by water pump (Marqus) (3) to the vessel (4) making the uniform water temperature, and then the water distributed by means a perforated plate (5), show in figure (2.a). The water is distributed in the form of falling films over the packing. The volumetric flow rate is measured by standard rot meter (6), the water flow rate is regulated by the water valve (P.V.C) (7). The pressure drop is measured by inclined U-manometer (10). The relative humidity of air at tower inlet is measured by psychrometer measuring both the dry bulb and wet bulb temperatures. The airflow into the tower was measured using airflow meter (9). In figure (1) , the inlet and outlet air and water temperature numerated by (13),(12),(15),and (14) respectively. In the research a new type of packing is used, named honeycomb (8) the name of this packing coming from its shape, which it is look like honey-comb (see figure (2.b)). The temperature is measured by the calibrated thermocouples (Four thermocouples are measuring the air temperature and other six measuring the water temperature).

Mathematical modeling

The heat and mass transfer characteristics of the evaporative cooling system can be determined by the conservation equations of heat and mass. The assumption of this model;

1. Adiabatic system.
2. Lewis number is not change through the tower.
3. Heat and mass transfer coefficients are constant.
4. The temperature profile through the cross section is uniform.

5. The air-water interface is saturated vapor through the interface.
6. Constant cross sectional area .

By considering control volume of each sector as shown in figure (2), the energy balance will be as follows[2,3,4,5];

$$G.di = L.di_W + G.i_w.dH \quad ......(1)$$

The energy balance in the liquid side can be written in terms of convective heat and mass transfer coefficients as follows[2,3,4,5];

$$L.di_w = h\ .a.dV.(T_W - T) + K_G.a.dV.i_{fg,W}.(H_{sat,w} - H) \quad ....(2)$$

The mass balance of the water and vapor over the control volume for each sector can be written as;

$$G.dH = K_G.a\ .dV.(H_{sat,w} - H) \quad .....(3)$$

The equation of Lewis number[1,2,3,4,5]:

$$Le = (h / K_G Cp)) \quad ....(4)$$

Substituting equation (4), into equation (2) gives:

$$L.di_w = Le.K_G.Cp.a.dV.(T_W - T) + K_G.a.dV.i_{fg,W}.(H_{sat,w} - H) \quad ......(5)$$

Substituting equation (1) into equation (5), we get

$$G.di - G.i_w.dH = K_G.a.dV.(Le.Cp.(T_W - T) + i_{fg,W}.(H_{sat,w} - H)) \quad ...(6)$$

Combining equation (6) and equation (3), we get

$$\frac{di}{dH} = i_w + \frac{Le.Cp.(T_W - T) + i_{fg,W}.(H_{sat,w} - H)}{(H_{sat,w} - H)} \quad ....(7)$$

By re-arranging, we get

$$\frac{di}{dH} = Le.Cp.\frac{(T_W - T)}{(H_{sat} - H)} + i_{g,w} \quad .....(8)$$

Where $(i_{g,w} = i_w + i_{fg,w})$

The specific heat of air:

$$Cp = \frac{(i_{sat,W} - i) - i_g^0 (H_{sat,w} - H)}{(T_W - T)} \quad ....(9)$$

By substituting equation (9) in equation (8) we get;

$$\frac{di}{dH} = Le.\frac{(i_{sat,W} - i)}{(H_{sat,w} - H)} + (i_{g,w} - i_g^0.Le) \quad ......(10)$$

By rearranging, we get

$$\frac{di}{dH} + \frac{Le}{(H_{sat,w} - H)} i = (i_{g,w} - i_g^0.Le) + \frac{Le.i_{sat,W}}{(H_{sat,w} - H)} \quad .....(11)$$

Assume y = i

x = H

C_1 = Le

$C_2 = H_{sat,w}$

$C_3 = (i_{g,w} - i_g^0.Le)$

$C_4 = Le.i_{sat,W}$

Equation (11) become as follows;

$$\frac{dy}{dx} + \frac{C_1}{C_2 - x} y = C_3 + \frac{C_4}{C_2 - x} \quad .....(12)$$

Let

$$Q_{(X)} = C_3 + \frac{C_4}{C_2 - x}$$

So the equation (12) can be solve as follows

$$I_{(X)} = e^{\int \frac{C_1.dx}{C_2 - x}} = e^{-C_1.\ln(C_2 - x)} = (c_2 - x)^{-C_1} \quad .... (13)$$

$$I_{(X)}.y = \int I_{(X)}.Q_{(X)}.dx \quad ......(14)$$

By substituting equation (13) in equation (14) we get;

$$(c_2-x)^{-C_1}.y=\int(c_2-x)^{-C_1}.(C_3.\frac{C_2-x}{C_2-x}+\frac{C_4}{C_2-x}).dx$$

(15)

$$(c_2-x)^{-C_1}.y=\int\frac{c_3dx}{(c_2-x)^{c_1}}+\int\frac{c_4(c_2-x)^{-c_1}}{(c_2-x)}dx$$

(16)

(In this stage let C_1=1)

$$(c_2-x).y=(C_3.C_2.x-C_3.\frac{x^2}{2}+C_4.x)$$

(17)

$$y=\frac{(C_3.C_2.x-C_3.\frac{x^2}{2}+C_4.x)}{(C_2-x)}$$

(18)

$$i=-(i_{g,w}-i_g^o.Le)(H_{sat,w}-H).\ln(H_{sat,w}-H)+Le.i_{sat,w}$$

(19)

The water temperature distribution can be calculated from

$$\Delta T_W=-(\frac{G}{L})(\Delta h-H.i_w)$$ …..(20)

Results and Discussion

From figures (3.a) and (3.b), one can see that the variation of outlet air temperature with air flow rate at different inlet water temperature tends to decrease with increasing air flow rate. However, for high air flow rate region, decreasing rate of outlet air temperature decreases. At specific air and water flow rates, and inlet air temperature, effect of inlet water temperature on the outlet air temperature is very small. The reasonable agreement is obtained from the comparison between the predicted results and the present experimental data.

Figures (4.a) and (4.b) shows the variation of outlet water temperatures with air flow rate, it can be seen that the outlet water temperature decreases as air flow rate increases. The decrease of the outlet water temperature caused by the increase of air flow rate results in an increase in heat transfer rate. The results we got from the model are reasonable agreement with the experimental data.

The relation between the pressure drop across the cooling tower and the air flow rate is shows in figure (5), it can be seen that the pressure drop tends to increase as the air flow rate increases. The pressure drop slightly increases at small value of air flow rate and rapidly increases at high value of air flow rate.

Figure (6), shows the comparison between the data points of the outlet air temperatures obtained from the experiment and those obtained from the model .It can be seen that the majority of the data fall within ± 4% of the model ,which prove that the model give high accuracy relatively with real results(experiment results). Figure (7), shows the comparison between the data points of the outlet water temperatures obtained from the experiment those obtained from model. One can see that the majority of the data fall within ± 6 % of the model.

In figure (8), one can see the variation of temperature ratio with flow rate. The temperature ratio can be calculated from the following equation;

$$TR=\frac{T_{W,in}-T_{W,out}}{T_{W.in}-T_{a,wb,in}}$$

……(21)[4, 5]

From figure (8), one can notice that the temperature ratio increases with increasing air flow rate at a given air and water temperatures. This increasing can be explained by equation (21) in which the outlet water temperature decreases as air flow rate increases (the denominator of this equation not change because that the inlet water and inlet air wet bulb temperature are constant). For a given air flow rate, the inlet water temperatures have significant effect on the decrease of temperature ratio as shown in figure (8).

The effect of water flow rate on the temperature ratio is show in figure (9). It can be see that increase of temperature ratio becomes relative less as water flow increases; this happed because the outlet water temperature increases with increasing flow rate.

In figure (10), one can see the relation between the temperature ratio and the tower effectiveness, which can be calculated by equation (22) [3, 5]. The tower effectiveness increased with increased temperature ratio.

$$\varepsilon = \frac{i_{air,out} - i_{air,in}}{i_w - i_{air,in}} \qquad (22)$$

Tower characteristics .In other meanings, the value of specifies the size of equipment necessary to achieve maximum possible effectiveness.

$$NTU = \int_{H_i}^{H_O} \frac{dH}{H_{Sat} - H} \qquad (23)$$

The number of transfer unit increases as the temperature ratio increases at L/G = 2.5, 1.8, and 1.1. However, trends of curves become cajole as L/G decreases.

Conclusions

1. The performance of using Honeycomb packing is very good in using and it can be using in Iraq because of its high efficiency and its production in Iraq is very easy (simple shape).
2. By comparison, between the results obtained from the experiment and those obtained from the model. It can be seeing that the model give high accuracy relatively with real results (experiment results).
3. The tower effectiveness increased with the temperature ratio because the outlet water temperature decreases as airflow rate increases.
4. The number of transfer unit increases as the temperature ratio increases at different L/G. However, trends of curves become cajole as L/G decreases.
5. The pressure drop increasing with increasing airflow rate at different temperature.
6. Variation of outlet air temperature with airflow rate at different inlet water temperature tends to decrease with increasing airflow rate.

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Nomenclature

a Surface area of packing per unit volume, m^2/m^3

Cp Specific heat, $kJ/kg.^oC$

G Air flow rate, kg/s

H Humidity ratio, kg of moisture / kg of dry air

$H_{sat,w}$ Saturated humidity ratio of air-water vapor at T_w , kg of moisture / kg of dry air

h Heat transfer coefficient , $W/m^2.^oC$

i_w Enthalpy of water, kJ/kg

i_g^o Enthalpy of water vapor at zero ^oC, kJ/kg

$i_{sat,w}$ Saturated enthalpy of air at T_w, kJ/kg

i Enthalpy of air , kJ/kg

$i_{fg,w}$ Phase change enthalpy , kJ/kg

Kg Mass transfer coefficient, $kg/m^2.s$

Le Lewis number (heat transfer coefficient of air /mass transfer coefficient multiply by heat of moist air).

L Water flow rate, kg/s

NTU Number of transfer unit

V Total volume of the tower, m^3

T Air temperature, ^oC

T_w Water temperature, ^oC

TR Temperature ratio

T_w Temperature of water , ^oC

ε Tower effectiveness

***Subscripts*:**

in Inlet

sat Saturation

w Water

out Outlet

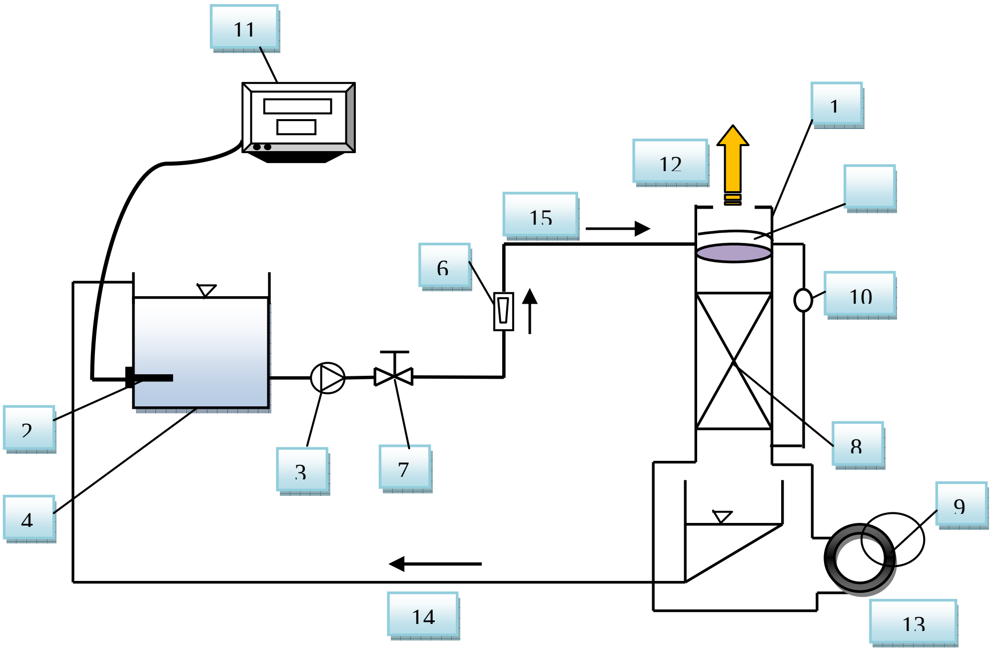


Figure (1): Layout of experimental apparatus

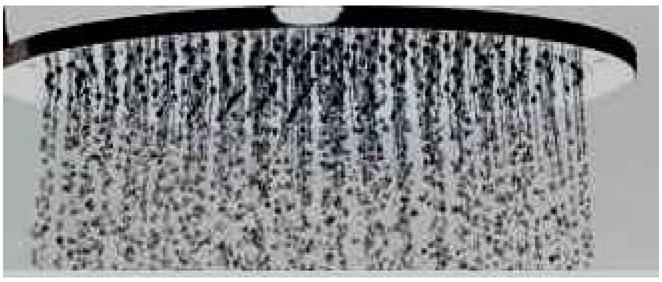


Figure (2.a): Water distribution system

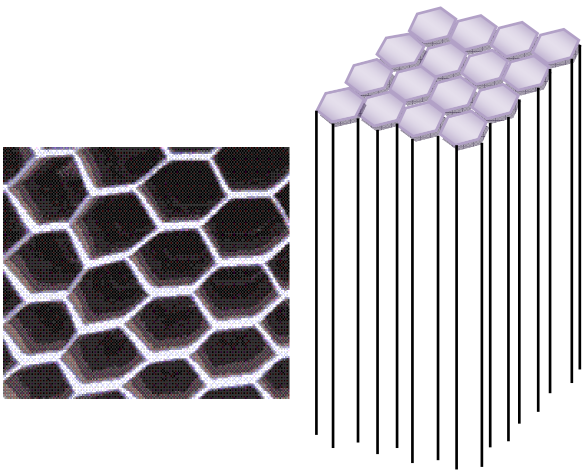
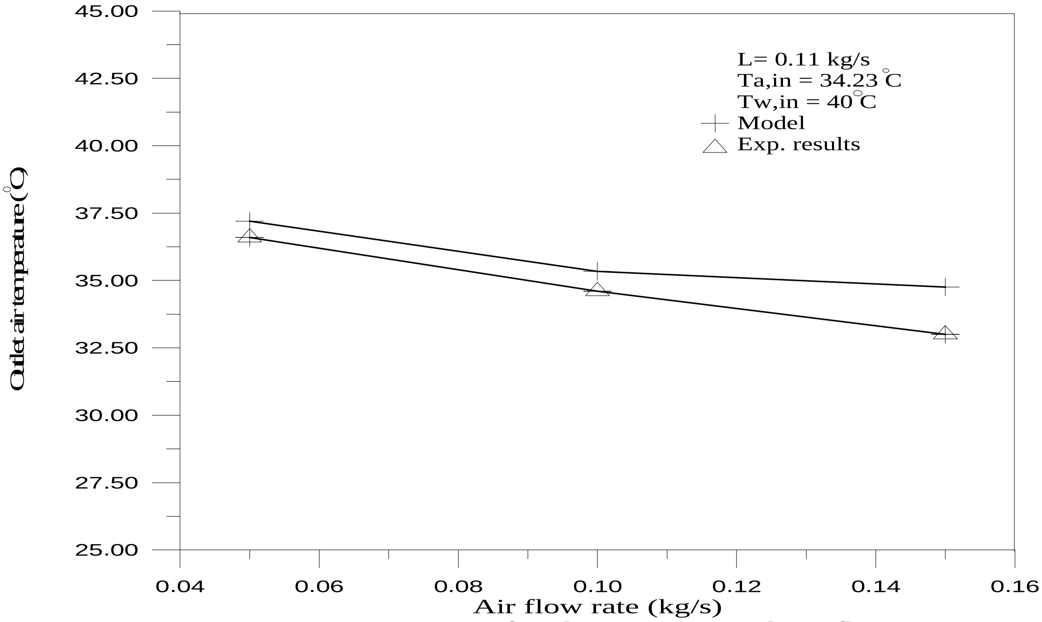
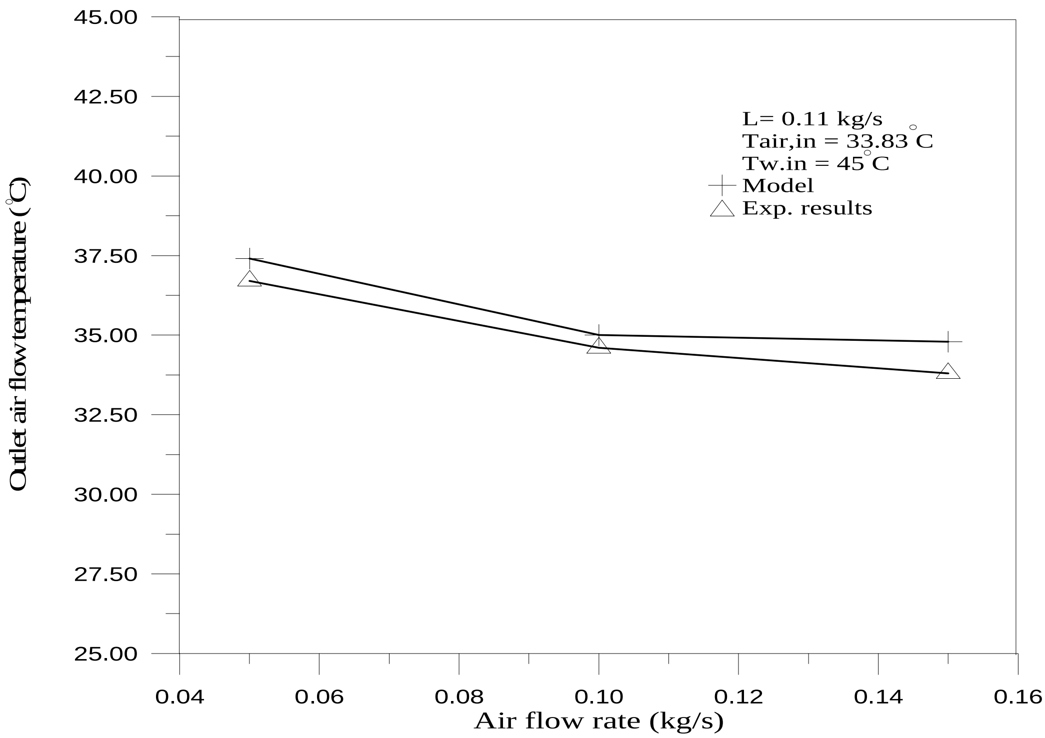


Figure (2.b): Fill type Honeycomb



Fig(3.a):Variation of outlet air temp. with air flow rate



Fig(3.b):Variation of outlet temp. with air flow rate

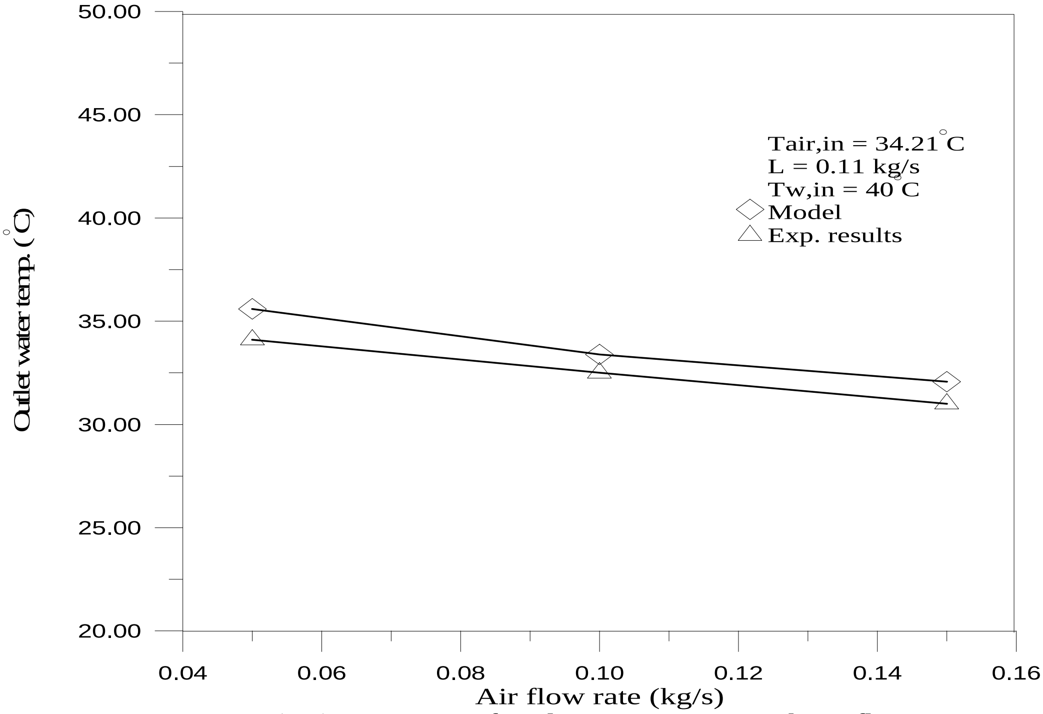


Fig.(4.a):Variation of outlet water temp. with air flow rate.

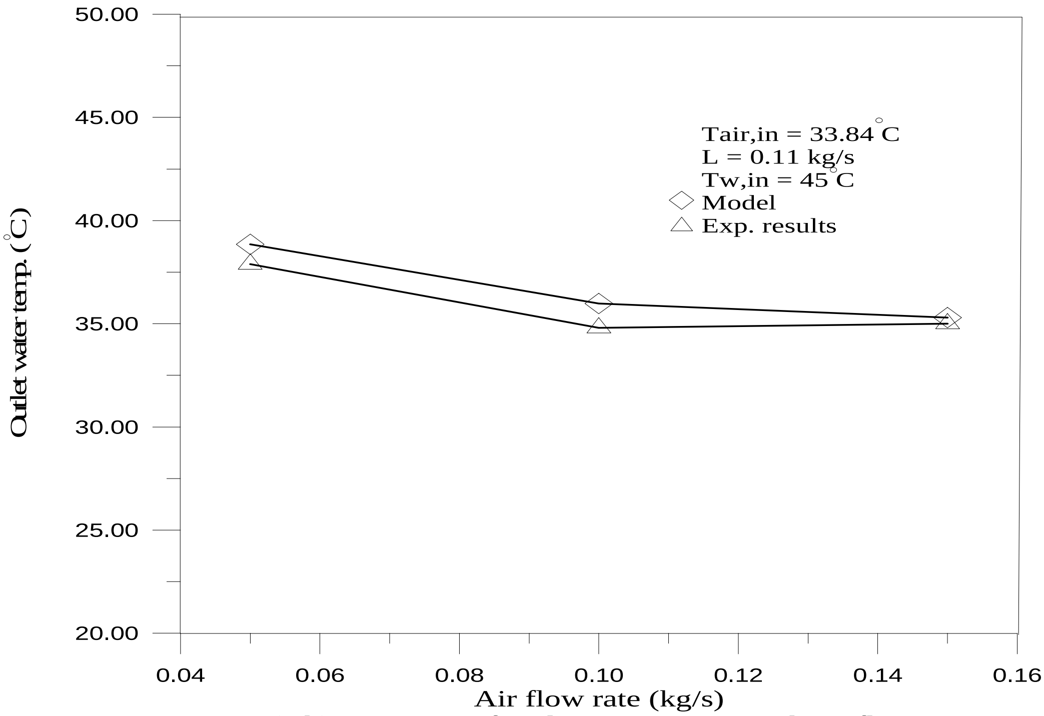


Fig.(4.b):Variation of outlet water temp. with air flow rate.

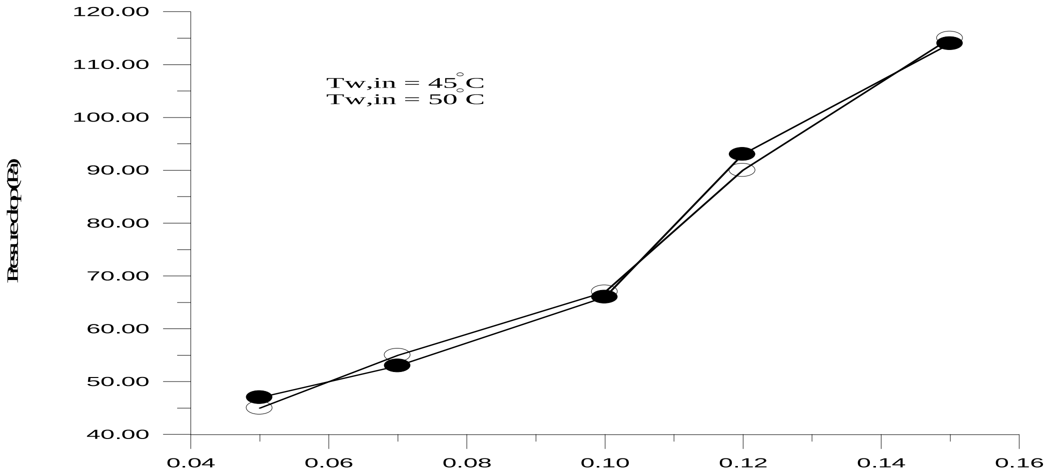


Fig.(5):The effect of air flow rate on pressure drop

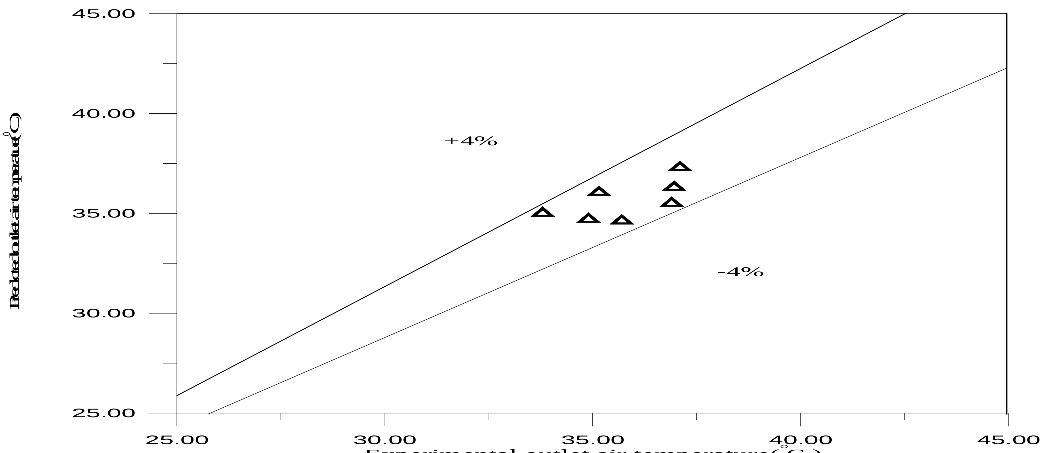


Fig.(6):Comparison between predicated and experimental outlet air temp.

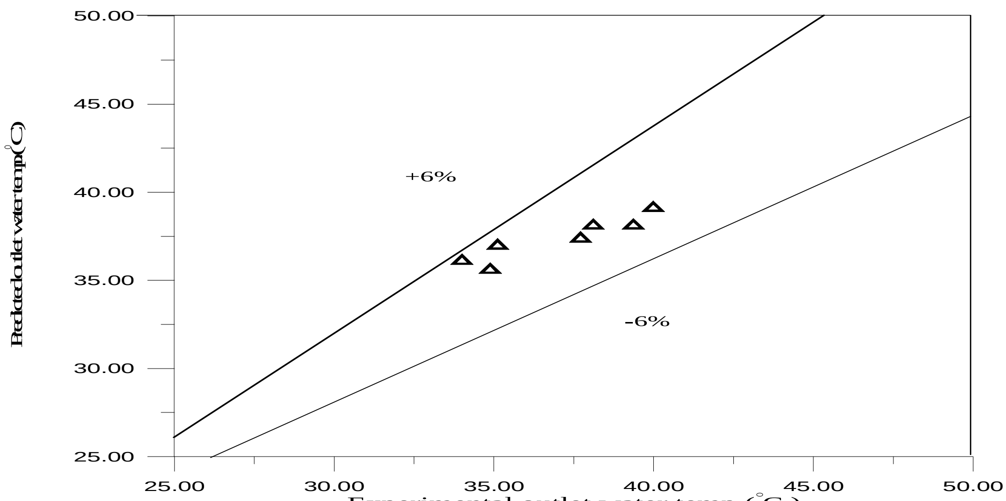
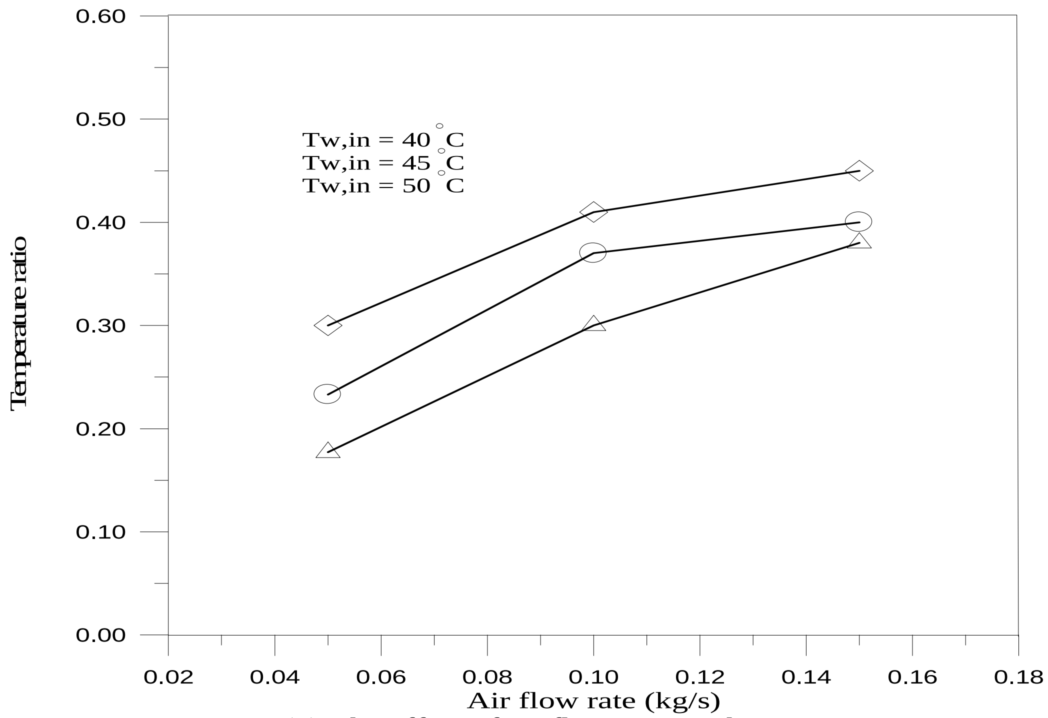


Fig.(7):Comparison between predicated and experimental outlet water temp.



Figure(8):The effect of air flow rate on the temperature ratio

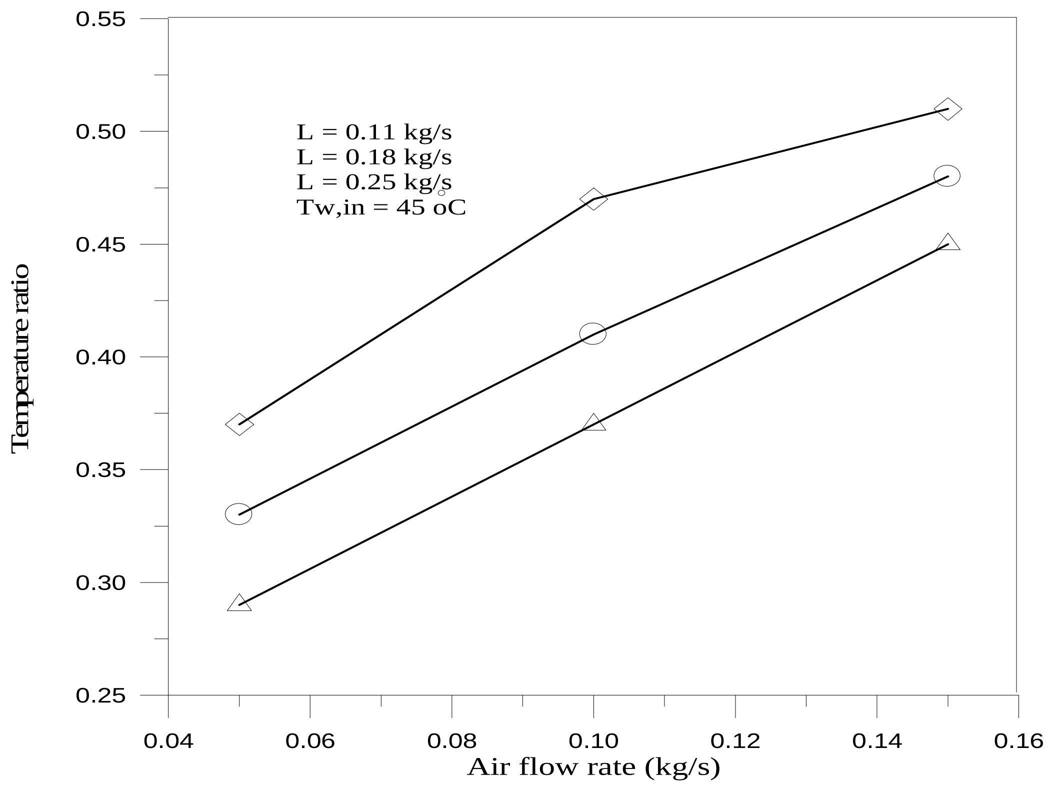
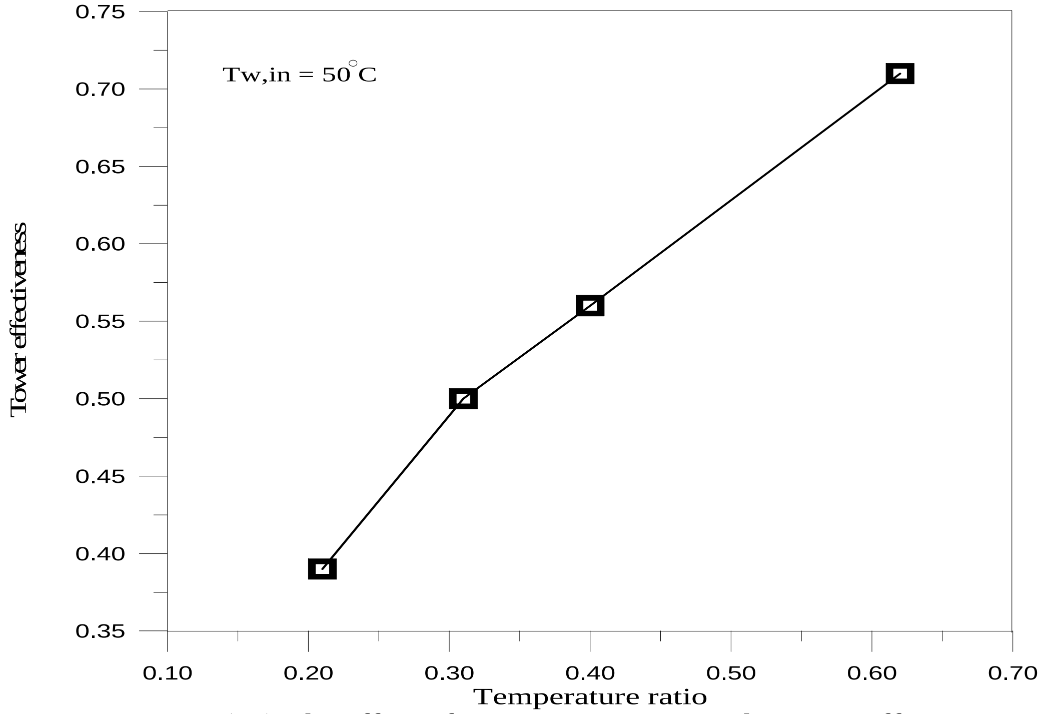
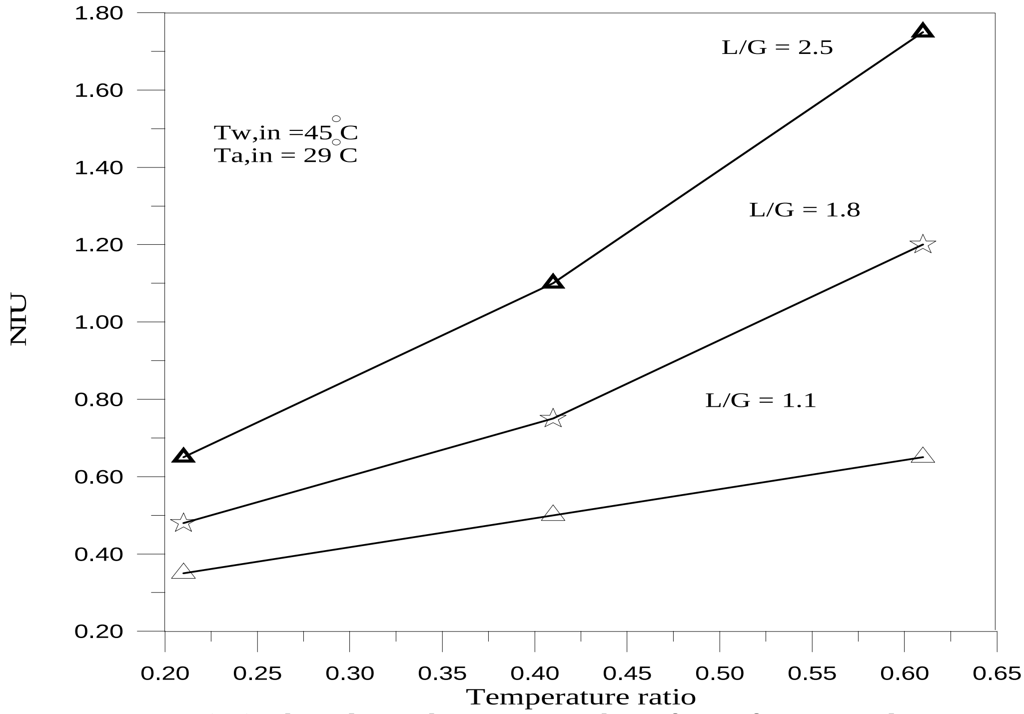


Fig.(9):The effect of liquid folw rate on the temp. ratio.



Figure(10):The effect of temperature ratio on the tower effectiveness.



Figure(11):The relation between number of transfer unit and temp. ratio