Study the Performance of Different Packing in Open Cooling Towers

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Abstract

*The effect of different shapes (corrugate, and grid) of packing on the performance in air - water-cooling tower were addressed in this research. A mechanical forced draught counter current flow-cooling tower of (40 * 40) cm cross-sectional area and 1.5 m high was constructed. Airflow rates of 0.4,1.4, and 2 kg/m².s were used in conjunction with water flow rates of 1.5, 2, and 2.5 kg/m².s, and water temperature of 40, 45, 50 ^0C. The tower characteristics (K_G a Z/L) over all volumetric mass transfer coefficient(K_Ga) volumetric heat transfer coefficient in gas phase (h_Ga) and volumetric heat transfer coefficient in liquid phase (h_la) are found to be a function of the water and airflow rates simultaneously. The numerical analysis was done by preparing computer program to reduce each set of data obtain form the plant and to evaluate the performance coefficient, volumetric owns transfer coefficient, volumetric heat transfer coefficient.*

The work was extended to include the longitudinal and transverse temperature profiles within the tower for different parameters. By employing packed heights of 100, 75, 50 and 25 cm, end effects were studied, and the tower characteristics (K_G a Z/L) corrected from these effects. Least square method was used to correlate the experimental results for (K_G a Z/L) in terms of air flux G and water flux L.

دراسة أداء حشوات مختلفـة الإشكـال على ابراج تبريد الماء

الخلاصة

يعنى البحث بدراسة أداء حشوات مختلفـة الإشكال (Corrugated and Grid) ومن مادة Steel ,على برج تبريد الماء بجريان متعاكس ذي مقطع(40 ×40) سم وارتفاع 1.5 م. درس الاداء عند ثلاثة معدلات لجريان الماء(2.5, 2, and1.5)

كغم/ثا.م2 بالاقتران مع ثلاث معدلات لجريان الهواء(0.4and, 1.4, 2) كغم/ثا.م2 وثلاث قيم لدرجات الحرارة 313 و 318 و 323 كلفن.

لدراسة تأثير النهايات غير ارتفاع الحشوات لثلاثة قيم 100,75,50,25 سم وجرى تصحيح اداء البرج من هذه التاثيرات, و استخدمت عدد من المعادلات Correlation equation لتمثيل اداء البرج بدلالة كل من معدل جريان الهواء ومعدل جريان الماء. لحساب أداء البرج تم إعداد برنامجين للحاسبة الالكترونية مكتوبين ومنفذين بلغة فيجوال بيسك. البرنامج الأول يقوم لحساب ميل خط الربط والبرنامج الثاني المبني على موديل رياضي ياخذ بنظر الاعتبار كل من معدل تبخر الماء ومقاومة طبقة الماء لانتقال الحرارة و لحساب درجة الحرارة البينية. كما وأظهرت نتائج الحاسبة الاليكترونية لـ أداء البرج مقارنة جيدة مع نتائج اخرى.

تمت دراسة وتعين المعاملات الحجمية لانتقال الحرارة والكتلة. وتم تعين وايجاد المعادلات التي تربط هذه المتغيرات للمنظومة مع تلك المعاملات. وتمت دراسة توزيع درجات الحرارة بالاتجاه العمودي والافقي للبرج وعلاقة ومدى تاثير متغيرات المنظومة والتي تشمل كل من درجة حرارة الماء الداخل ومعدل جريان الماء ومعدل جريان الهواء وانثالبية الهواء الداخل وارتفاع الحشوات على توزيع درجات الحرارة خلال حشوات البرج وتم ايجاد معادلة لحساب توزيع درجات الحرارة في أي نقطة وجزء في البرج .

1. Introduction

The water cooling process is one of the simultaneous heat and mass-transfer in which sensible heat is transferred as a result of the difference of temperature between water and air, and a small proportion 'of the water is evaporated as result of the differential of vapor pressure between the water surface and the general air steam. The dual nature of the exchange process does not lend itself to an accurate treatment, but a fortunate relationship between the sensible heat and mass transfer coefficients allows the two transfers to be combined in one simple transfer equation with enthalpy as driving force and the mass transfer coefficient as overall coefficient for the combined transfer process. This total -heat method was originally suggested by Merkel in 1926[1] and has since been elaborated by several authors, notably Lichtenstein in 1943[2] in the design of mechanical draught cooling towers, also Woods and Betts in 1950 [3, 4] and Chilton in 1950 [5] in the design of natural draught cooling towers. Development of the combined equation for the dual transfer can be found in many papers, in particular one published by Carey and Williamson 1950 [6]. The equation of transfer over the full depth of the packing for a counter current flow tower may be written as follows:

$$K_G \, a = \frac{0.624 \; G \; (i_{G_2} - i_{G_1})}{Z \; A \; \Delta i_m} \qquad (1)$$

An examination of the equation (1) reveals that the value of $(i_{Gl}-i_{G2})/\Delta i_m$ is termed the number of transfer units (NTU) required for the specified cooling duty. The value of $0.623 G/A.K_G \, a$ therefore represents of the height of transfer unit (HTU).

Glenn in 1982 [7] discussed new methods for predicting and evaluation tower performance for spray cooling tower systems, although dimensions analysis techniques for heat and mass transfer are used, requirement for this solution are not always met. Proposed and present methods are compared. **James** in 1982 [8] demonstrated rather pointedly that cooling tower performance and operation are not as straightforwardly simple as it many times is thought to be. These misconception or "Old Cooling Tower Tales" can cost many in all phases of dealing with cooling tower. **Larry** in 1962 [9] presented a model which predicts the performance of an evaporative cooling system at other than the .tested operation points. The model is based upon an empirical correlation for convection heat transfer and a proposed form for this correlation is introduced. **George** in l982 [10] showed why the altitude is an important factor that should be taken into consideration when designing or testing a tower. Information also be presented which should be helpful in doing calculations for elevations other than sea level. Only counter flow towers discussed in this research because of important performance of that type of tower. **John** in 1984 [11] claimed that the entertainment of hot moist air from a cooling tower into the tower into the tower inlet air decreases both overall tower and plant performance. In this study characterizes recirculation on a circu1ar mechanical draft-cooling tower were obtained. The data of circular mechanical draft-cooling tower are compared to data from similar tests on rectangular draft cooling tower. **Marcel** in 1982 [12] presented a simple method to eliminate Merkel's theory approximations (The Merkel Theory was published in 1925 and demonstrated that heat transfer in evaporative cooling tower was approximately proportional to a difference of enthalpies. Approximation of the theory is very large, mainly when water temperatures are high). Hopefully this can be the base for a new future standard of the cooling tower industry. **Allen** in 1991 [13] compared the difference of results between using Merkel assumption to simplify the mathematical calculation and using computers and numerical methods which allowed for more precise determinations. **Branislav** in 1995[14] claimed than an exact analytical method for evaluation heat and mass transfer in closed circulated cooling towers, (previously developed by author), has been expanded and revised to provide a computerized means to predict the thermal performance and determined the associated energy requirement a specified tower design. The validity of the model has been verified and fine- tuned by extensive laboratory testing. After a brief overview of the analytical model, it is demonstrated how this model can be effectively applied to parallel flow spray water -airflow arrangement. **Goshayshi and Missenden** in 2000[15] studied experimentally the effect of form with corrugated packing on mass transfer and pressure drop characteristics in atmospheric cooling towers. Their results shows that the mass transfer coefficient decreased with increase in packing pitch and increase in the ratio of rib pitch to rib height. Friction factors were expressed by a dimensional equation which included pitch and distance between the packings, for both smooth and rough surfaces. From these results, the relationship between packing mass transfer coefficient and pressure drop was deduced. **Adriaan**

in 2001 [16] examined the effect of special variations of L/G within a cooling tower, on the overall thermal performance of the tower. Air temperature profiles above the fill, resulting from non-uniform water distribution profile will be presented. Theoretical versus actual results for the return water temperature will be compared. **James Willa** in 2009[17] lists many errors in specifications, bid evaluations, construction and operation. However, these errors continue to be made effecting cost and performance of cooling towers. This is a brief review of some of the most important of these errors and methods to correct them. **Nina** in 2009 [18] study film fills for cooling towers made of PVC and of a flame retardant PP which are tested by several different methods and international standards to evaluate the actual performance of these two materials. **Hamid** in 2009[19] proposed a new method of comparison of existing cooling tower fills have been developed and the performance of the best packing has been expressed in relation to the ideal packing. Hamid investigation is made using measurements of the mass transfer rates and pressure drops for a comprehensive range of PVC plastic packing producing an economic comparison to find the best geometry and range. In order to do this, heat transfer and pressure drop for turbulent conditions in fills used in the modern cooling tower have to be studied.

2. Experimental

A mechanical forced draught counter-flow cooling tower was designed fig (1). The general arrangement was made in a certain way to provide maximum accessibility to the tower section for observation and maintenance without restricting the operation. The equipment and instruments were arranged so that the overall material and energy balances could be readily accomplished. Water circulation during a run was maintained in a closed system. Water from the tower basin 8 ft^3 was pumped by means of a centrifugal pump. Water passes through constant vessel tank (gives steady state head), heated by four coil heaters, and finally to the tower distributing main.
Water was distributed on the packing top edge by nozzles; Figure (2) shows the water distribution nozzles. It insured good distribution. Water flow rate was measured by means of an independently calibrated rotameter with stainless steel float. The tower is 40 cm. by 40 cm. in cross - section and the height between inlet water distributor and inlet air distributor in the tower is 120 cm. A 6 mm thick Perspex sheet is bolted to the front side of the tower. This was used to give more flexibility of opening the tower and observing water movement. Air forced into the test section from multiple entries at the bottom of the tower. This arrangement provides a counter current between falling water and upward air. A mist eliminator made out of porous sponge pad (40cm × 40cm) was placed on the top of the water distribution chamber. Air volume flow rates were measured by means of an independently calibrated inclined U-manometer was connected through airflow orifice plate. A centrifugal fan supplying air through was used. The water in the basin down the packing was kept at constant level by an overflow pipe 1.25 cm,

which was connected to the overflow tank. The packing was made from different packing materials, with dimension's 40 cm width, 6 mm thickness, and 25, 50, 75, and l00 cm height. Different heights were used in order to study the end effects. The packing consists of (18) sheets. Figure (3) shows a sketch of the method used for holding the packing plates in position. In order to avoid splashing, the distance between the water distribution nozzles and the top of packing is 5 cm. Twelve thermocouples of type T (copper as positive, copper-nickel, constantan, as negative) were used for temperature measurement, located in a manner such that the weighted average temperature of air or water were determined at each point. The thermocouples reading and measured variable are listed in table (1). All the thermocouples are calibrated with calibrated mercury in glass thermometers simultaneously into, a thermostat bath. Distilled water was used and the temperature range was 0 C° -60C°.

Table (1): The thermocouples reading and measured variable

Thermocouple number		Measured variable
1	2	Inlet water temperature
3	4	Outlet air dry bulb temperature
5	6	Outlet air wet bulb temperature
7	8	Outlet water temperature
9	10	inlet air dry bulb temperature
11	12	inlet air wet bulb temperature

To obtain the correlation defining the water temperature profile along the cooling tower at different air and water conditions, thirty-six thermocouples type T were used. Every wire of the thermocouples are connected to four digital recorder labeled A, B, C and D. The thermocouples are labeled according to the numbers on the digital recorder such as thermocouple number 1A, 2A,.... , IB, 2B, 3B, ….1C, 2C, 3C ,and…..1D,2D,3D,…..etc were adopted. Sixteen thermocouples are placed in the first layer which is 25cm. away from the top of the packing. These thermocouples are labeled from 1A to 16A which are arranged in the form of 4x4 matrix, Fig. (4.a).The second set of thermocouples are placed 50 cm away from the top of the packing which are labeled 1BA to 16B, which are arranged in the form of 4 ×4 matrix, Fig.(4.b). The third set of thermocouples are placed 75 cm , away from the top of the packing which are labeled 1C-16C .The thermocouples are arranged in the form of 4x4 matrix, Fig. (4.c).The final set of

thermocouples are placed 100 cm, away from the top of the packing which are labeled 1D to 16D, Fig.(4.d).

3. Computational Model

This model is based on dividing the tower volume into finite increment volumes, all increments will by considered to have the same differential performance coefficient ($K_G.a.dZ/ L$) (but not necessarily to have the same volume).The performance of cooling tower was investigated using the finite difference technique. The tower volume was divided into incremental volume. In this model the evaporation rate is neglected. The solution starts from the incremental volume -at the down tower and proceeds towards up-tower. Energy balance and material balance equations are applied for each step (increment). The finite difference technique will give all the required conditions (like temperature, enthalpy, and evaporation, ...) for the bulk water , bulb air and interfere for all the increments of the tower .The present solution is handled in two computer programs , and the calculation based" on enthalpy potential theory with its basic equations.

$$\frac{L.C_L}{G} = \frac{di_G}{dt_L} \tag{2}$$

$$N.T.U = \frac{K_G aZ}{L} = \int_{t_{L1}}^{t_{L2}} \frac{C_L\, dt_L}{i_i - i_C} \tag{3}$$

In this analysis, the performance coefficient is obtained by:

$$\frac{K_G aZ}{L} = N.\frac{K_G aZ}{L} \tag{4}$$

N: Total number of incremental volumes within the tower.

3-1 Assumptions for the Analysis

1.The property of fluid varies vertically.
2.The tower body was isolated thermally from environment
3.Heat and mass transfer coefficients are constants throughout the tower.
4.The air / water interface is saturated vapor at the interfacial temperature (ti).
5.The liquid side heat transfer is negligible.
6.The evaporation rate is not negligible.

4. Results and Discussion

A sound interpretation of the gained data necessitates its graphical presentation. The tower characteristics (K_G a Z/L) is shown in Figs. (5 and 6), plotted against values of water to air ratio (L/G), for packing at a heights of 25 cm and nominal inlet water temperature t_{L2} = 318 K (45 ºC). It can be observed that straight nearly parallel lines suffice to fit the above data. Analogous behavior was reported by other authors who addressed themselves to the problem as Glenn.

In general, for constant value of air flux G, the larger the water to air ratio the smaller the tower characteristics. This behavior can be attributed to fact that, increase of water flux L at a constant value of air flux G, means increase in heat load that in turn decreases the packing capability for dissipating this excess in heat load. In other words, increasing the value of L decreases the cooling range (t_{L2} - t_{L1}), and since

$$\frac{K_G aZ}{L} = \int_{t_{L1}}^{t_{L2}} \frac{C_L \, dt_L}{i_i - i_G} \tag{6}$$

The integral limits of the above equation stand for the cooling range t_{L2}- t_{L1}. As this value decrease the integral value will decrease too, and vice-versa.

To reveal the influence of inlet water temperature on tower characteristics. Figs. (7 and 8) indicate that for a fixed value of water to air ratio (L/G), as the inlet water temperature increase the tower characteristics will decrease. This confirms that increasing the heat load decreases the tower characteristics; The experimental results showed that the reduction of (K_Ga z/L) amounting on average only about (7 %) for each 5 K (5°C) increase in the inlet water temperature.

The influence of inlet water temperature associated with air flux G on volumetric mass transfer coefficient (K_G a) is depicted in Fig. (9). It is clear that increasing the inlet water temperature decreases the volumetric mass transfer coefficient, and this occurs due to decrease in the value of tower characteristics (K_G a z/L), as indicated in Figs. (7 and 8). On the other hand, when the value of air flux increases form 1.4 to 2 Kg/s.m^2, (K_G a) increases about (20%) , since the rate of evaporation is directly proportional to air flux G .

The effect of inlet water temperature and air flux G on volumetric heat transfer coefficient (h_G a) is entirely analogous to their effect on (K_G a), as shows in Fig. (10) Since it is calculated from Lewis relationship:

$$h_Ga = K_Ga \ .Cs \tag{7}$$

Figure (11) compares between the tower characteristics (K_G a Z/L) for different height of packing (t_{L2} = 318k (45°C)). The characteristics decrease with of (L/G) for constant G. It as reported in the increasing the value of investigators in the cooling tower field have correlated the tower characteristics (K_G a Z / L) with water to air ratio (L/G) as follows:

$$\frac{K_G aZ}{L} = c\,(L\,/\,G\,)^{m} \tag{8}$$

This correlation (8) is extensively used for estimating the tower characteristics in terms of water to air ratio. Each curve in Fig. (11) can be expressed in a form of equation (8). Thus, twelve tower characteristics are show in Table (2):

Table (2): (K_Ga Z/ L) tower characteristics for steel grid packing

Height (cm)	Correlation	
100	$\dfrac{K_G aZ}{L} = 0.34\,(\dfrac{L}{G})^{-0.36}$	
75	$\dfrac{K_G aZ}{L} = 0.25\,(\dfrac{L}{G})^{-0.36}$	$G{=}2\ \mathrm{kg/m^2.s}$
50	$\dfrac{K_G aZ}{L} = 0.16\,(\dfrac{L}{G})^{-0.37}$	
25	$\dfrac{K_G aZ}{L} = 0.11\,(\dfrac{L}{G})^{-0.355}$	
100	$\dfrac{K_G aZ}{L} = 0.36\,(\dfrac{L}{G})^{-0.1.1}$	
75	$\dfrac{K_G aZ}{L} = 0.423\,(\dfrac{L}{G})^{-1.03}$	$G{=}1.4\ \mathrm{kg/m^2.s}$
50	$\dfrac{K_G aZ}{L} = 0.29\,(\dfrac{L}{G})^{-1.06}$	
25	$\dfrac{K_G aZ}{L} = 0.145\,(\dfrac{L}{G})^{-1.08}$	
100	$\dfrac{K_G aZ}{L} = 0.361\,(\dfrac{L}{G})^{-0.74}$	
75	$\dfrac{K_G aZ}{L} = 0.275\,(\dfrac{L}{G})^{-0.8}$	$G{=}0.4\ \mathrm{kg/m^2.s}$
50	$\dfrac{K_G aZ}{L} = 0.18\,(\dfrac{L}{G})^{-0.79}$	
25	$\dfrac{K_G aZ}{L} = 0.092\,(\dfrac{L}{G})^{-0.92}$	

The magnitude of, end effects, is shown in Figs. (12, 13,and 14). It is determined and tested at various heights with constant value of air flux G. The value of tower characteristic for end effects gained upon extrapolation to Zero height; hence an intercept on the vertical axis will give the value of (K_G a Z/L) eq., the number of transfer units corresponding to end effects only which will be subtracted from the value of uncorrected tower characteristics (K_GaZ/L); while the intercept with the horizontal axis correspond to the negative value of (Z eq.), the equivalent height of end effects Once again, a comparison is conducted between tower characteristics (K_G a Z/L) at different packing heights, after excluding the values of end effects shown in Fig. (15).

For steel grid packing

$$\frac{NTU}{Z} = 0.352 \ (L)^{-0.671} \ (G)^{0.34} \tag{9}$$

For steel corrugated packing

$$\frac{NTU}{Z} = 0.355 \ (L)^{-0.666} \ (G)^{0.34} \tag{10}$$

The correlation of tested data consists of finding the basic curve that coincides with all other curves, and taking L and G separately to account the variation of air flux. The expected error is given with $\pm$ 2 % probability and similar term to this will be associated with each correlation equation. The value of $\frac{T - To}{t_{1.2} - To}$ are plotted in Cartesian coordinates versus $\frac{\bar{z}}{z}$ for all runs. The dimensionless parameters R and L/G are calculated for each run. The curves obtained agree with the following function:

$$\frac{T - T_O}{t_{L2} - T_O} = 1 - \frac{e^{b*\frac{-z}{z}} \ e^{-b*\frac{-z}{z}}}{e^{b} - e^{-b}} \tag{11}$$

Where b is a function of R and L/G for details see Fig. (16) for steel grid packing and Fig.(17) for steel corrugated. Different values of b are determined for the proposed function, eq (11), to fit the experiment result of the temperature profiles. The values of b are plotted versus L / G on log-log paper for given values of R and it is found that these two variables are independent of each other as shown in Fig.(18) for steel grid packing and Fig (19) for steel corrugated The relation between b and R is found to be as follows:

$$b= 3.36 \ R^{1.02} \tag{12}$$

Steel corrugated packing

$$b = 3.7R^{1.01} \qquad\qquad (13)$$

Steel grid packing
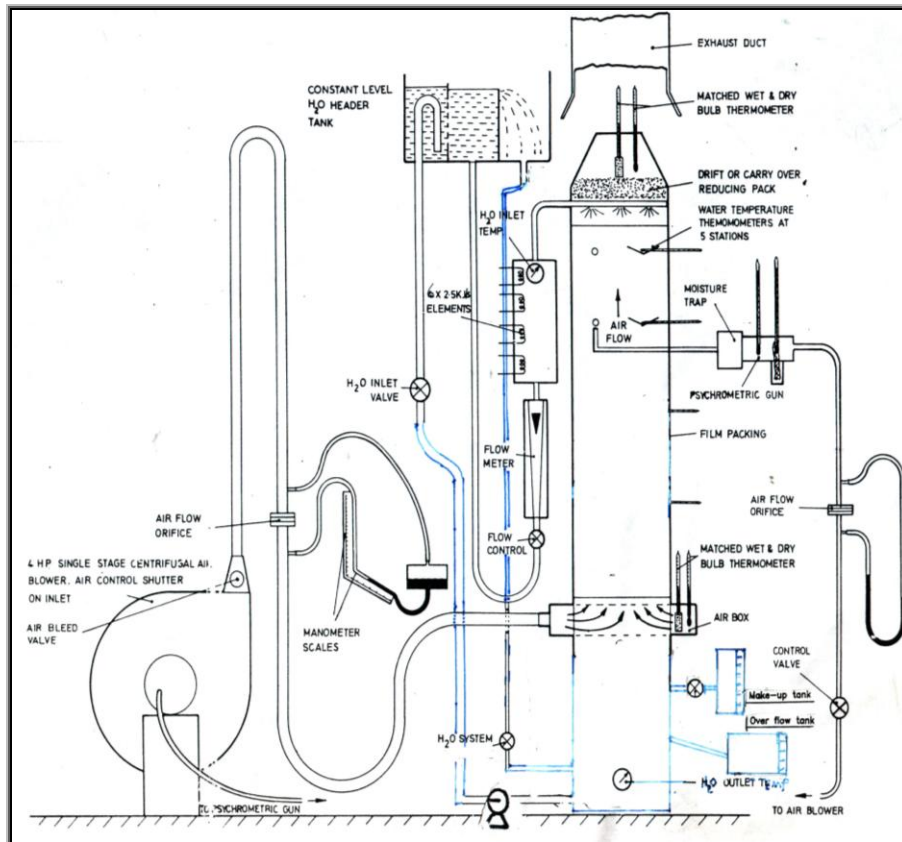
Fig. (1): Cooling tower system
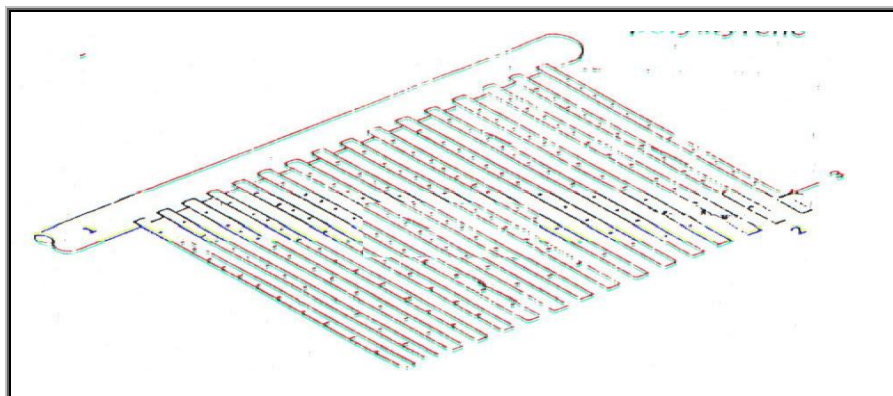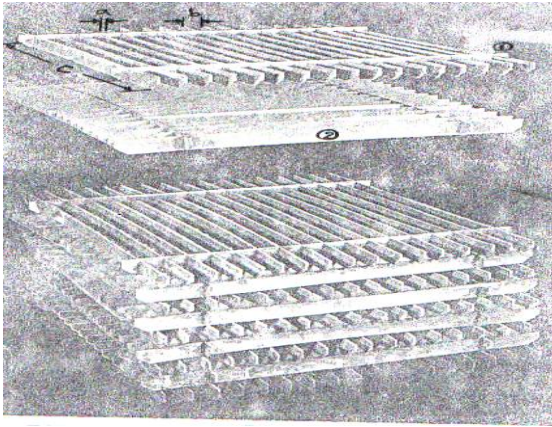

Fig.(2):Water Distribution system

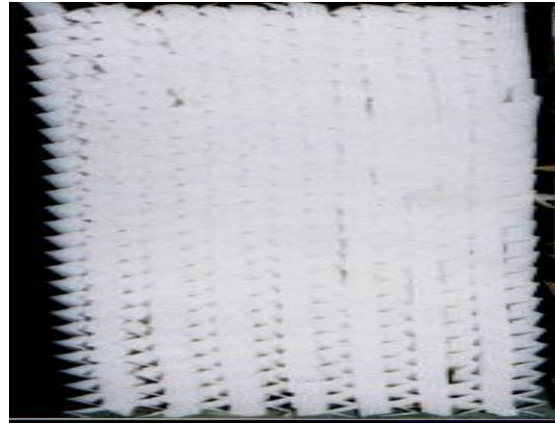*Fig.(3.a):The arrangement of grid packing*

Fig.(3.b):The arrangement of Corrugate
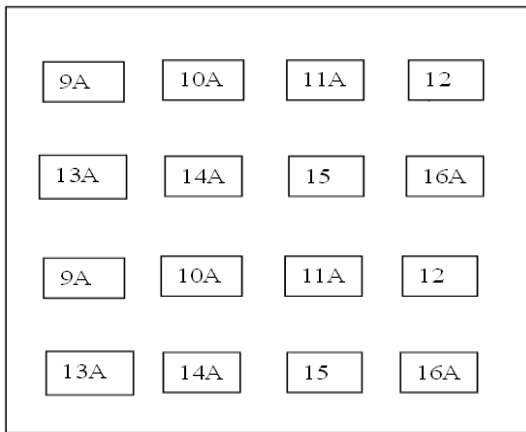
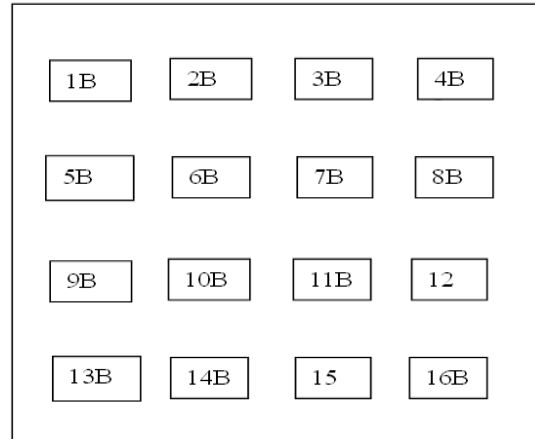**Fig. (3): Packing shapes**

Fig.(4.a): First Layer Of Thermocouple
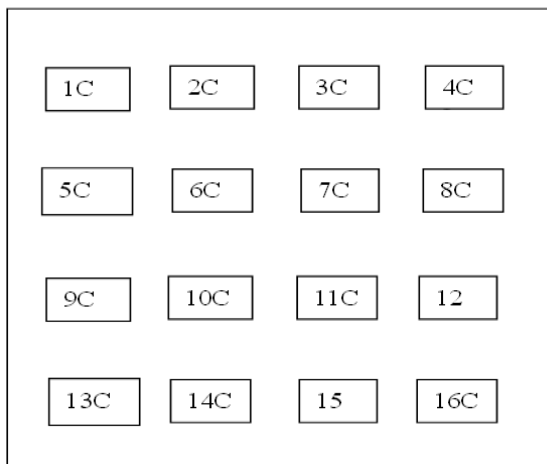

Fig.(4.b): Second Layer Of Thermocouple
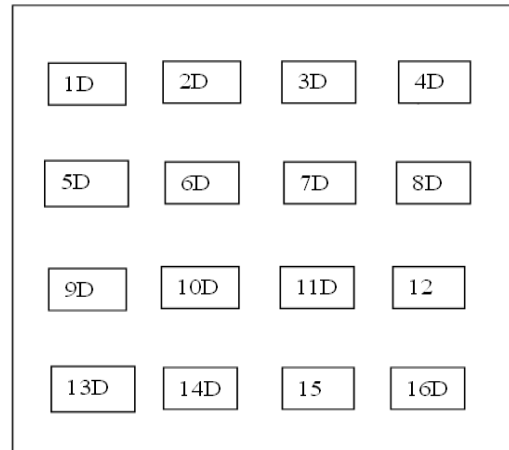

Fig.(4.c): Third Layer Of Thermocouple
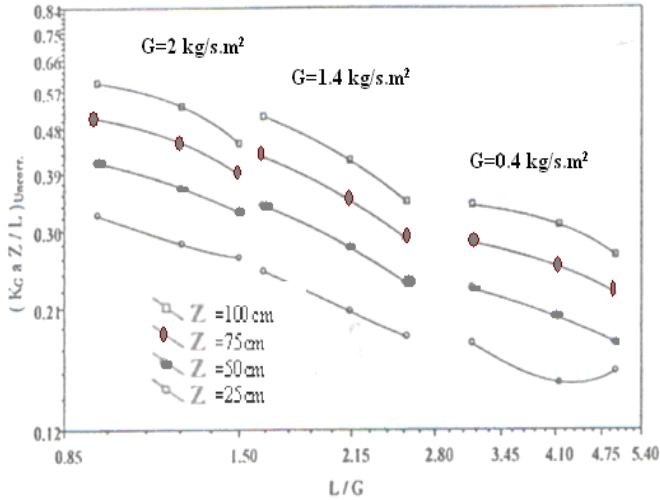

Fig.(4.d): Fourth Layer Of Thermocouple

Fig.(4) Thermocouple Layer

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Fig.(5):Uncorr. N.T.U vs L/G for steel grid Packing ,tL2=45 0C
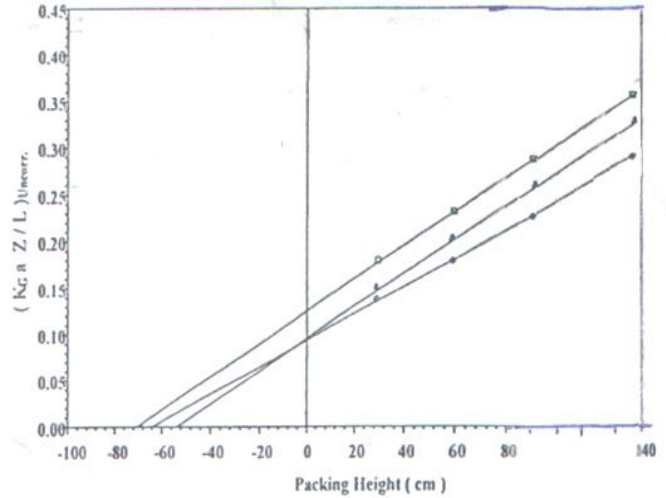
Fig.(8):Uncorr.N.T.U. vs L/G for steel corrugated packing ,G=1.4 kg/s.m2
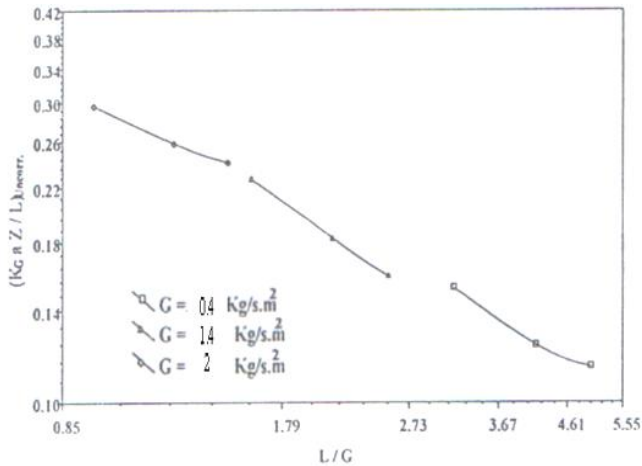
Fig.(6):Uncorr. N.T.U vs. L/G for steel corrugated packing tL2=45 0C ,and Z=25 cm
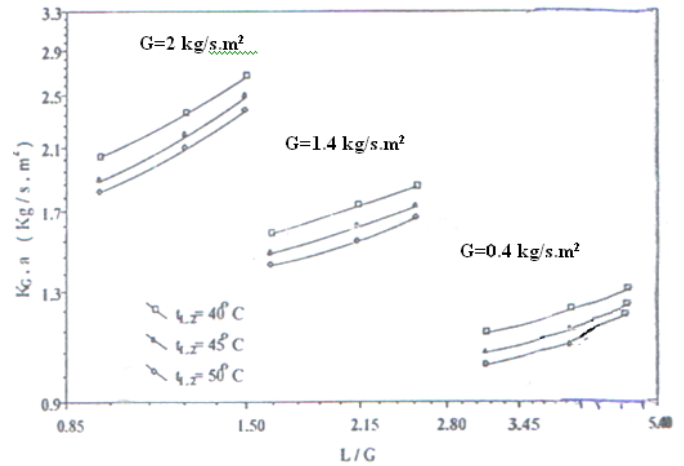
Fig.(9):Volumetric mass transfer coefficient vs. L/G for steel grid packing=25 cm
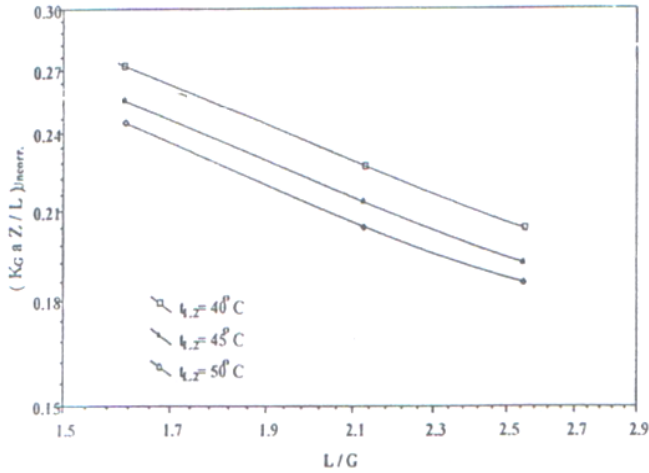
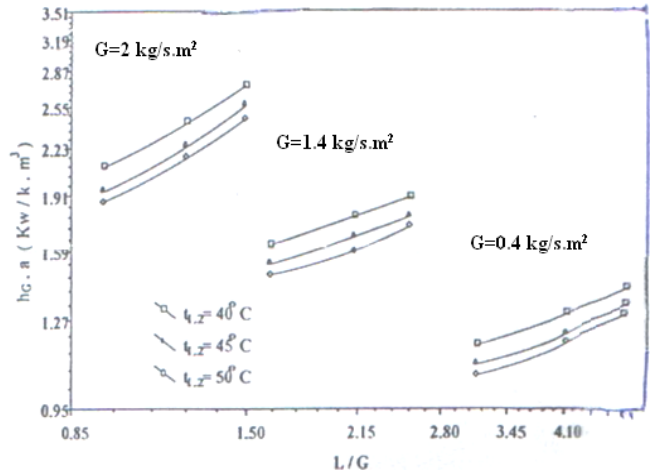**Fig.(7):Uncorr. N.T.U. vs L/G for steel grid packing, G= 1.4 kg/s.m² and Z=25 cm**

Fig.(10): Volumetric heat transfer coefficient vs. L/G for steel grid packing ,and Z=25 cm

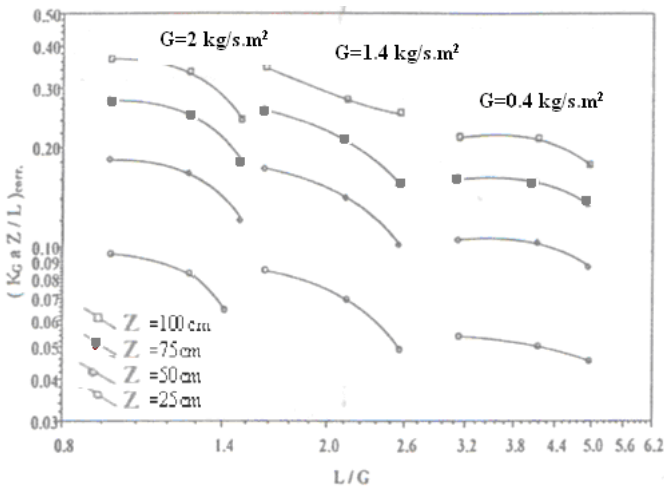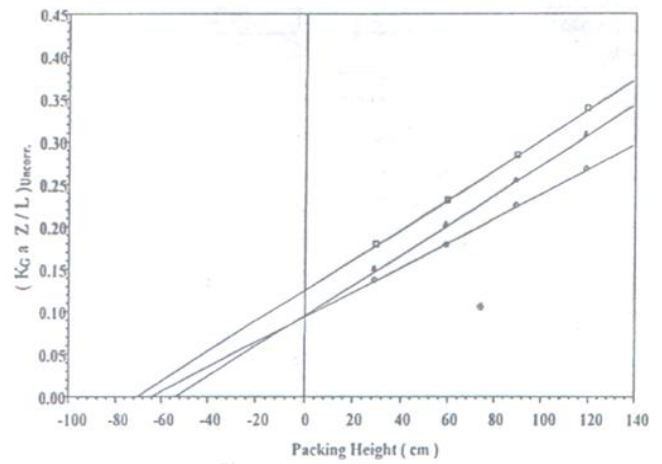**Fig.(11):Uncorr. N.T.U. vs. L/G steel grid Packing t_{L2}=45 ⁰C.**

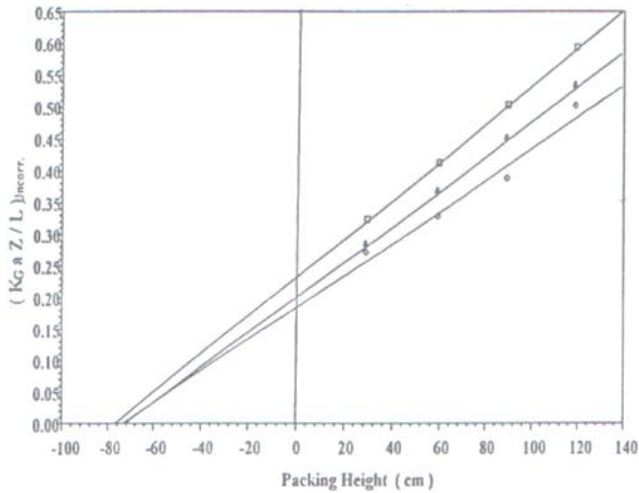**Fig.(14): Uncorr. N.T.U. vs. packing height for steel grid packing ,t_{L2}=45 ⁰C,and G=0.4 kg/s.m²**

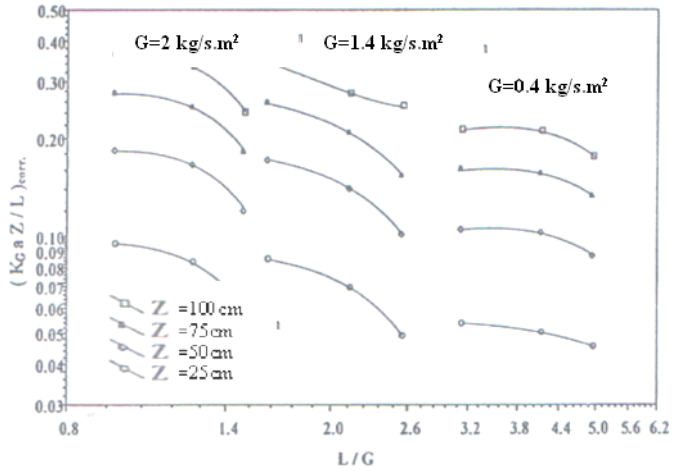**Fig.(12):Uncorr. N.T.U.vs. packing height for steel Grid packing ,t_{L2}=45 ^{0}C and G=2 kg/s.m^2**

Fig.(15):Corr. N.T.U. vs. L/G for steel grid packing, t_{L2}=45 ^{0}C
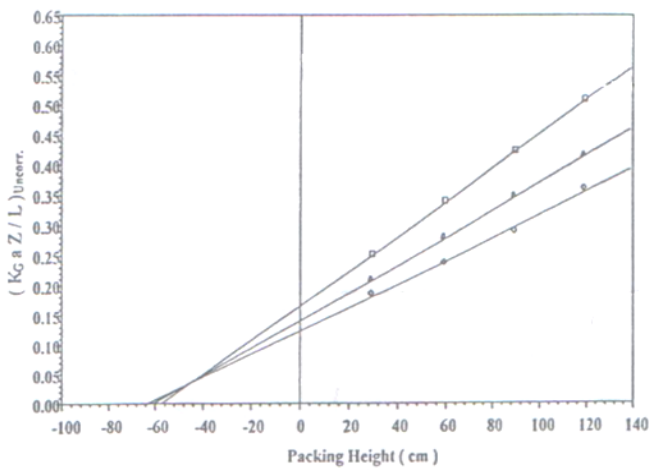

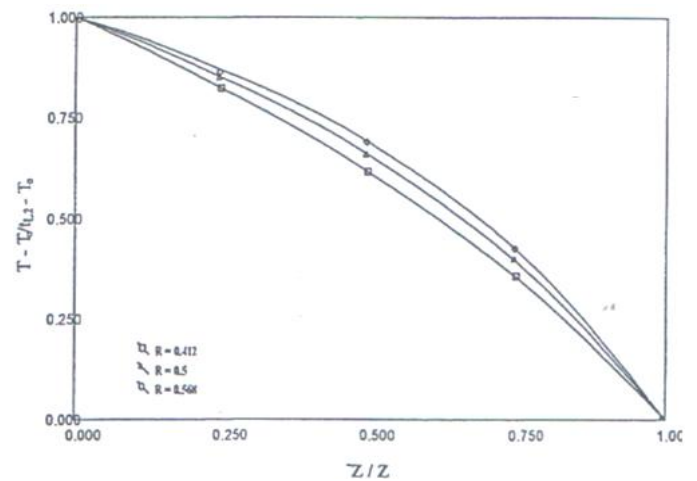**Fig.(13):Uncorr. N.T.U. vs. packing height for steel**

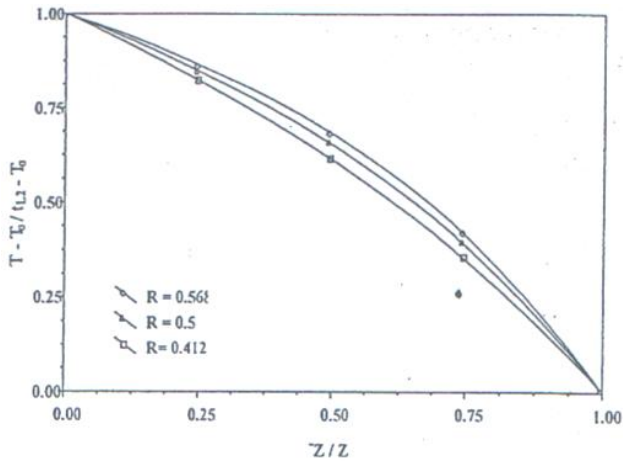**Fig.(16):Temperature profile vs. ⁻Z/Z for steel grid packing**

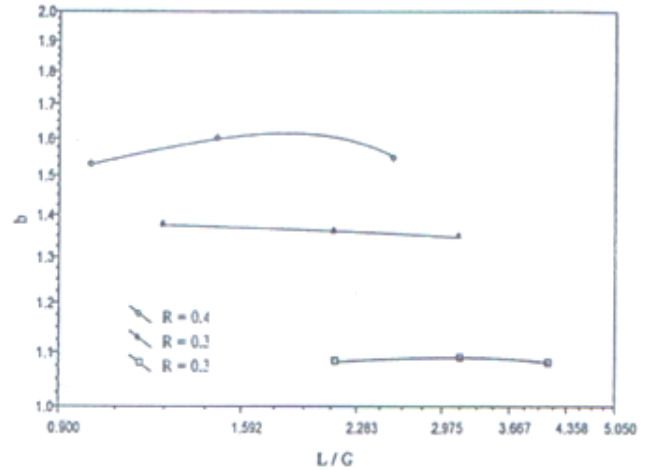**Fig.(17):Temperature profile vs. Z/Z for steel corrugated packing**

Fig.(18):Relation between b and L/G at different R for steel grid packing
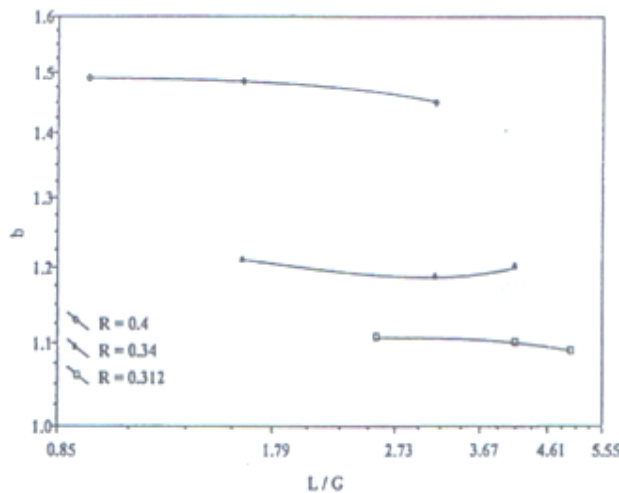
Fig. (19) Relation between b and L/G at different R for steel corrugated packing.

5. Conclusions

1. Maximum performance (K_G a Z/L) in a given volume of tower packing may be obtained with minimum water and air flow ratio (L/G).

2. Maximum mass transfer coefficient in a given volume of water packing may be obtained with maximum airflow rate and minimum liquid flow rate.

3. The end effects include (spray chamber above the packing, and also the open space below the packing), some cooling materials; made to estimate the corrected value of tower characteristics

form these effects. The resulted correlation equation per unit depth of packing height is given as:

$$\frac{NTU}{Z} = A \, (L)^{B} \, (G)^{C} \tag{14}$$

below the packing), some cooling materials; made to estimate the corrected value of tower characteristics form these effects. The resulted correlation equation per unit depth of packing height is given as:

Packing		A	B	C
Material	Shape			
Steel	Parallel	0.34	-0.78	0.35
Steel	Grid	0.38	-0.77	0.34

4. The individual volumetric coefficients showed   to be affected mainly by the system variables; such as air and water flux as well as the inlet water temperature. Also, least square method used   to   express   these coefficients in term of G and L in a form analogous to equation (l4). (h_Ga) values may be attained by using Lewis relationship.

5. The water temperature profiles have been obtained for different parameters concerning the tower performance. The correlation is sufficiently capable of defining the water temperature profile along the tower for different air and water conditions.

$$\frac{T - T_{O}}{t_{L2} - T_{O}} = 1 - \frac{e^{b*\frac{-z}{z}} \; e^{-b*\frac{-z}{z}}}{e^{b} - e^{-b}} \tag{15}$$

Where:

$$b = 3.53 \, R^{1.1} \qquad \text{(Steel, corrugated packing)} \tag{16}$$

$$b = 3.6 \, R^{1.12} \qquad \text{(Steel, grid packing)} \tag{17}$$

From the results one can see that; The temperature variation along the tower for given inlet water temperature and cooling range, is a function of the air inlet enthalpy, as well as, the position but is not a function of the air water flow rates.

6. Nomenclature

A   Cross- sectional area m^2

b   Ratio between the difference inlet air enthalpy and outlet air enthalpy to the inlet air enthalpy

G   Airflow rate kg/s.m^2

h$_G$ a   Volumetric heat transfer coefficient in gas phase   kw/m^3. K.

L   Water flow rate kg/s.m^2.

K$_G$ a   Volumetric mass transfer coefficient kg/s .m^3

Z   Packing height cm

H$_L$ a   Volumetric heat transfer coefficient in liquid phase kw/m^3.K.

t$_{L2}$ Temperature of water at top of packing K

T$_o$ Temperature of water at bottom of packing K

i$_{G2}$ Enthalpy of air at the tower outlet kJ/kg

i$_{G1}$ Enthalpy of air at the tower inlet   kJ/kg

C$_L$ Specific heat of water kJ/kg.K

C$_s$ Specific heat of moist air kJ/kg.K

c   Constant

m   Constant

R   The ratio between (enthalpy of air saturation minus enthalpy of air at the bottom of packing) and (enthalpy at the top)

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