PERFORMANCE OF COUNTERCURRENT FLOW COOLING TOWERS

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ABSTRACT

This research is concerned with performance of different shapes (parallel plate, and grid) and material (polypropylene) of packing for an air –water-cooling tower. A Mechanical forced draught counter current flow cooling tower of 300 mm × 300 mm, cross-sectional area and 1.65m. High was constructed.

Air flow rates of 0.533,1.035, and 1.774 kg/m2.s were used in conjunction with water flow rates of 1.686, 2.218, and 2.66 kg/m2.s, and water temperature of 40,45,50 0C .

The tower characteristics (K_G a Z/L), overall 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 air flow rates simultaneously .

The work was extended to include the longitudinal and transverse temperature profiles within the tower for different parameters.

By employing packed heights of polystyrene , and polypropylene of 120,90,60, and 30 cm , end effect 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 .

The numerical analysis was done by preparing two VISUAL BASIC computer program to reduce each set of data obtain form the plant and to evaluate the performance coefficient, volumetric mass transfer coefficient, volumetric heat transfer coefficient ,rejected heat , and evaporation rate.

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 1926) and has since been elaborated by several authors, notably (Lichtenstein 1943) in the design of mechanical draught cooling towers , also (Woods and Betts 1950)and (Chilton 1950)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). The equation of transfer over the full depth of the packing, for a counter current flow tower may be written as follows:

$$K_G\,a = \frac{0.624\,G\left(i_{G2} - i_{G1}\right)}{Z\,A\,\Delta i_m} \qquad (1)$$

An examination of the equation (1) reveals that the value of (iG1–iG2)/△im is termed the number of transfer units (NTU) required for the specified cooling duty. The value of $0.623G/A.K_G$ a therefore is representative of the height of a transfer unit (HTU).

Sulaymon (1972) found the characteristic of the mechanical induced draught counter-flow-cooling tower. The packing (polystyrene spheres) has been studied theoretically and experimentally both as a fixed and fluidized bed. Table tennis spheres were used as packing in a fixed and fluidized mechanical forced draught counter flow cooling tower (Shooul 1975). (AL.Arigi 1979), and (Sulaymon 1980) .

Glenn (1982) 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 1982) demonstrated rather pointedly that cooling tower performance and operation are not so 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 towers.

(Larry 1982) 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.

(George1982) 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 1984)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 circular mechanical draft-cooling tower were obtained. The data of circular mechanical draft- cooling tower are compared to data from similar tests on rectangular mechanical draft cooling tower.

(Mercel 1982) 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 are very large, mainly when water temperature are high). Hopefully this can be the base for a new future standard of the cooling tower industry.

(Allen 1991) 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 1995) 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 –air flow arrangement.

(Adriaan 2001) 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 vs. actual results for the return water temperature will be compared

EXPERMANTAL WORK

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. The water from the tower basin 2 × 2 × 1.65 ft. was pumped by means of a centrifugal pump. The water passes through constant vessel tank (gives steady state head, then to the stainless steel water heating tank with 6 × 2.5 Kw (240 volt) immersion elements and then to the tower distributing main.

Water was distributed on the packing top edge by means of 14 P.V.C tubes, 10-mm diameter, each had 14 holes, 2.5 mm diameter. Figure (2) shows the water distribution tubing. It insured film flow of water

Water flow rates were measured by means of an independently calibrated rotameter with stainless steel float.

The tower is 300 mm. by 300 mm. in cross – section and the height between inlet water distributor and inlet air distributor in the tower is 1.4 m.

A 6 mm thick Perspex is bolted to the front side of the tower. This was used to give more flexibility of opening the tower and observing the water movement.

The air forced into the test section from multiple entry at the bottom (right and left sides of the tower column). This arrangement provides a counter current between falling water and upward air. A mist eliminator made out of porous sponge pad (300 mm × 300 mm) 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, this manometer is connected through air flow orifice plate (designed in accordance with British standard 1042, part 1, 1964).

A centrifugal fan supplying air through the tower 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 15 x 10^{-3} m^3

The make-up tank was allowed to feed the tower basin. This was connected by a 1.25 cm hose with adjustment value. The tower basin had a drain connection for the purpose of cooling tower circuit water drainage.

The vehicle of heat transfer in the cooling tower was the packing. The packing was made from different packing materials, with dimension's 300 mm width , 6 mm thickness , and 30 , 60 , 90 , and 120 cm height . This altered height which was in order to study the end effects. The packing consists of (14) 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 tubes and the top of packing is 3.75 cm.

The instrument used for air and water temperatures measurement were a thermocouple of type T (copper as positive, copper-nickel, constantan, as negative) Twelve thermocouples were used for this purpose, located in a manner such that the weighted average temperature of air or water were determined at each point, except the inlet water temperature was achieved by a single thermocouple .

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 is used and the temperature range is 0 C° – 60C°

Table(1)

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 three digital recorder labeled A, B, and C.

The thermocouples are labeled according to the numbers on the digital recorder such as thermocouple number 1A, 2A, …. , 1B, 2B, 3B, and 1C, 2C, 3C …etc were adopted. Precautions were taken to ensure the strength of these connections.

Nine thermocouples are placed in the first layer which is 30cm. away from the top of the packing. These thermocouples are labeled from 1A to 9A which are arranged in the form of 3×3 matrix, Fig. (4.a).

The second set of thermocouples are placed 60 cm away from the top of the packing which are labeled 10A, 11A, 12A, 1B, 2B, 3B, 4B, 5B, and 6B, which are arranged in the form of 3×3 matrix, Fig.(4.b).

The third set of thermocouples are placed 90 cm , away from the top of the packing which are labeled 7B, 8B, 9B, 10B, 11B, 12B, 1c, 2C, 3C .The thermocouples are arranged in the form of 3×3 matrix, Fig. (4.c).

The final set of thermocouples are placed 120cm , away from the top of the packing which are labeled 4C, 5C, 6C, 7C, 8C, 9C, 10C, 11C, and 12C of 3×3 matrix , Fig. (4.d).

Three calibrated thermometers are used in different places in the water thermostat temperature. The relation between the measured temperature and the calibrated temperature are:

$$t_C = 0.51 + 1.001\, t_m \qquad (2)$$

Computational model

An accurate and time – saving method is described here for correlating countercurrent cooling tower performance (model) by using computers. 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) this method used called, numerical finite – difference procedure. In this model the evaporation rate is not 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 (see Fig.(5 , 6)), and the calculation based on enthalpy potential theory with its basic equations.

$$\frac{L.C_L}{G} = \frac{di_G}{dt_L} \tag{3}$$

$$\text{N.T.U} = \frac{K_G\, a\, Z}{L} = \int_{t_{L1}}^{t_{L2}} \frac{C_L\, dt_L}{i_i - i_G} \tag{4}$$

In this analysis, the performance coefficient is obtained by:

$$\frac{K_G aZ}{L} = M.\frac{K_G aZ}{L} \tag{5}$$

M: Total number of incremental volumes within the tower.

Assumptions for the Analysis

1. Counter flow, film type, and direct contact tower of constant cross section.
2. Adiabatic cooling tower.
3. The air water properties varies vertically.
4. Heat and mass transfer coefficients are constants throughout the tower.
5. The air / water interface is saturated vapor at the interfacial temperature (ti).
6. The liquid side heat transfer is not negligible.
7. The evaporation rate is not negligible.

RESULTS AND DISCUSSION

A sound interpration of the gained data necessitates its graphical presentation. The tower characteristics (KG a Z/L) is shown in Figs . (7,and 8), plotted against values of water to air ratio (L /G) , for packing at a heights of 30 cm and nominal inlet water temperature t12 = 313 K (40 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 for 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 (tL2 - tL1), and since

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

The integral limits of the above equation stand for the cooling range tL2 – tL1 . 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.(9,and 10) 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 (KG a z/L) amounting on average only about (8 %) for each 5 K (5 °C) increase in inlet water temperature .

The influence of inlet water temperature associated with air flux G on volumetric mass transfer coefficient (KG a) is depicted in Fig.(11) . 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 (KG a z/L), as indicated in Figs. (9, and 10). On the other hand, when the value of air flux increases form 1.035 to 1.774 Kg/s.m2 , (KG a) increases about (25%) , 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 (hG a) is entirely analogous to their effect on (KG a) , as shows in Fig.(12) since it is calculated from Lewis relationship :-

$$hG\ a = KG\ a\ .\ Cs \qquad (7)$$

Figure.(13) compares between the tower characteristics (KG a Z/L) in different packed heights for packing (tL2= 313 k (40 ^{0}C)) . The characteristics decrease with increasing the value of (L/G) for constant G. It was reported in the literature that the majority of investigators in the cooling tower field have correlated the tower characteristics (KG a Z / L) with water to air ratio (L/G) as follows :

$$\frac{K_G\ a\ z}{L} = c\ (L/G)^m \qquad (8)$$

Formula of type equation (8) is extensively used for estimating the tower characteristics in terms of water to air ratio and constants(c),(m) . Each curve in Fig.(13) can be expressed in a form of equation (8) . Thus twelve corrected tower characteristics are show in Table (2):

Table (2): (K_G a Z / L) Uncorr.for polypropylene grid packing

Height (cm)	Uncorrected Correlation	
120	K_G a Z/L = 0.5316 $(L/G)^{-0.664}$	G = 1.774 $K_g/s.m^2$
90	K_G a Z/L = 0.4453 $(L/G)^{-0.674}$	
60	K_G a Z/L = 0.3592 $(L/G)^{-0.368}$	
30	K_G a Z/L = 0.2731 $(L/G)^{-0.477}$	
120	K_G a Z/L = 0.5316 $(L/G)^{-0.923}$	G = 1.035 $K_g/s.m^2$
90	K_G a Z/L = 0.46183 $(L/G)^{-0.910}$	
60	K_G a Z/L = 0.6 $(L/G)^{-0.923}$	
30	K_G a Z/L = 0.3147 $(L/G)^{-0.838}$	
120	K_G a Z/L = 0.5842 $(L/G)^{-0.338}$	G = 0.533 $K_g/s.m^2$
90	K_G a Z/L = 0.4991 $(L/G)^{-0.919}$	
60	K_G a Z/L = 0.4096 $(L/G)^{-0.677}$	
30	K_G a Z/L = 0.3154 $(L/G)^{-0.770}$	

The magnitude of end effects, is shown in Figs.(14,15,and 16) . 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 (KG 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 (KG a Z/L); while the intercept with the horizontal axis correspond to the negative value of (Zeq) , the equivalent height of end effects .

Once again, a comparison is conducted between tower characteristics (KG a Z/L) at different packing heights , after excluding the values of end effects are show in Fig. (17) .

for polypropylene grid packing

$$\frac{NTU}{Z} = 0.387\,(L)^{-0.781}(G)^{0.365} \qquad (9)$$

for polypropylene parallel plate packing

$$\frac{NTU}{Z} = 0.388\,(L^{)-0.79}(G)^{0.37} \qquad (10)$$

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.

Water and air flux are markedly affected the slope of Tie – line values Fig.(18) ; thus the curves tend to have a sharp increase particularly with the water flux L , while the type of packing shows to have very few effect resulting from the small difference in the characteristic , thus the data of the types of packing were in close resemblance. It was found that the final results could completely be represented by equation, given as:

$$\text{Tie – line slope} = 17(L)^{1.263}\,(G)^{0.059} \qquad (10)$$

Polypropylene grid packing

$$\text{Tie- line slope} = 16.59\,(L)^{1.225}\,(G)^{0.074} \qquad (12)$$

Polypropylene parallel plate packing

The above equations are used to estimate the tie line slope directly instead of the trial and error method.

The liquid side heat transfer coefficient,(h_L a), is shown is Fig.(19). The data takes the same general shape of figures of tie-line since the liquid side heat transfer coefficient is a direct function of tie line slope (see tie-line equation). It is almost certain that the packing type does slightly affect values of (h_L a), but it believed that the following equations are adequate for correlating the data of packing:

$$h_L a = 22.67 \ (L)^{1.745} \ (G)^{0.563} \quad (13)$$

Polypropylene grid packing

$$h_L a = 16.77 \ (L)^{1.735} \ (G)^{0.699} \quad (14)$$

Polypropylene parallel plate packing

If the value of tie-line slope fed as near as infinity to the second program; i.e; the usual assumption of negligible resistance to heat transfer in the liquid side; the tower characteristics will fall about (13 %)as can be seen in Fig.(20).

From Table (3) , it is clear that the results of computer programs seem to be in very good agreement with those reported by(Thomas 1999) , who also considered the liquid side heat transfer resistance in their solution . But it should be noted that the values obtained from the second program are believed to be more accurate because much greater precision involved in the computer solution , coming from accounting the rate of water evaporation which was ignored in his solution .

Table.(3): Data were reported by Tomas comparing with Data were calculating by the programs

Data	Run1	Run2	Run3	Run4
T_{oi}(°C)	26.67	26.56	26.22	26.44
T_{wi}(°C)	20.22	20.17	20.11	20.11
T_{oo}(°C)	30.16	30.81	31.38	32.44
T_{wo}(°C)	28.67	29.69	30.47	31.78
t_{L2} (°C)	43.78	43.44	43.33	43.39
T_{LB}(°C)	30.33	31.39	32.06	33.33
L	1.356	1.709	2.03	2.713
G	2.28	2.28	2.28	2.28
Tie-line(prog)	1.41	1.48	1.54	1.66
Tie-line$_{(Ref)}$	1.34	1.4	1.48	1.59
(K_G a Z / L) (prog)	1.26	1.37	1.44	1.59
(K_G a Z / L) (Ref)	1.184	1.29	1.363	1.52

The value of $\frac{T - T_O}{t_{L2} - T_O}$ are plotted in Cartesian coordinates versus –Z / Z for all runs. The dimensionless parameters R and L/G are calculated for or each runs.

The curves obtained agree with the following function:

$$\frac{T - T_O}{t_{L2} - T_O} = 1 - \frac{e^{b \times \frac{-Z}{Z}} - e^{-b \times \frac{-Z}{Z}}}{e^{b} - e^{-b}} \quad (15)$$

Where b is a function of R and L/G for details see Fig. (21) for Polypropylene grid packing and Fig .(22) for Polypropylene parallel plate.

Different values of b are determined for the proposed function, eq (15), 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.(23) for polypropylene grid packing and Fig (24) for polypropylene parallel plate.

The values of b are platted against R and showed a linear relationship Fig.(25) for polypropylene grid packing and Fig (26) for polypropylene parallel plate packing.

The relation between b and R is found to be as follows:

$$b = 3.63 \ R^{1.02} \quad (16)$$

polypropylene, parallel plate packing

$$b = 3.7 \ R^{1.01} \quad (17)$$

polypropylene ,grid packing

CONCLUSIONS

The conclusions drawn form this analysis are enumerated as follows:

1. Maximum performance 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 air flow rate and minimum liquid flow rate.
3. Maximum volumetric heat transfer coefficient in gas phase and liquid phase in a given volume of tower packing may be obtained with maximum airflow rate and minimum liquid flow rate.
4. Least square method was used to correlate the experimental results, the dependent variable (KG a Z/L) correlated with water and air flow ratio (L/G) by fitting Log-log data . The exponents of the equations lied in the range between 0.89 and 0.28.

5. The end effects include (spray chamber above the packing , and also the open space

$$\frac{NTU}{Z} = A(L)^{B}(G)^{C} \qquad (18)$$

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			
Polypropylene	Parallel	0.388	- 0.79	0.37
Polypropylene	Grid	0.387	- 0.781	0.365

6. The individual volumetric coefficients of heat and mass transfer (h_L a , h_G a , and K_G a) 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(18) . (h_G a) values may be attained by using Lewis relationship . The liquid side heat transfer coefficient (h_L a) depends mainly on water flux L rather than air flux G , thus the correlation's of (h_L a) gives an exponent of L greater in a range about (32 – 40 %) than that of G .
7. 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.

$$\frac{T - T_O}{t_{L2} - T_O} = 1 - \frac{e^{b\times\frac{\bar{Z}}{Z}} - e^{-b\times\frac{\bar{Z}}{Z}}}{e^{b} - e^{-b}} \qquad (19)$$

Where

$b = 3.63\, R^{1.02}$ (20)

(Polypropylene, parallel plate packing)

$b = 3.7\, R^{1.01}$ (21)

(Polypropylene ,grid packing)

According to this function, the following conclusions are made :

a-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 .

b-The mean tower position is closer to the bottom of a counter – current water cooling tower and is only a function of the inlet air enthalpy for a given water inlet temperature . this has been found graphically for all runs according to the following function .

$$\frac{Z_{m1}}{Z} - 0.5 = \frac{\sinh^{-1}(0.5\sinh 3.63R^{1.02}) - 1.815R^{1.02}}{3.63R^{1.02}} \qquad (22)$$

for polypropylene, parallel plate packing

Z_{m1}: mean tower position for parallel plate

$$\frac{Z_{m2}}{Z} - 0.5 = \frac{\sinh^{-1}(0.5\sinh 3.7R^{1.01}) - 1.815R^{1.01}}{3.7R^{1.01}} \qquad (23)$$

for polypropylene, grid packing

Z_{m2}: mean tower position for grid

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 Air flow 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

T_O Temperature of water at bottom of packing

i_{G2} enthalpy of air at the tower outlet .

i_{G1} enthalpy of air at the tower inlet .

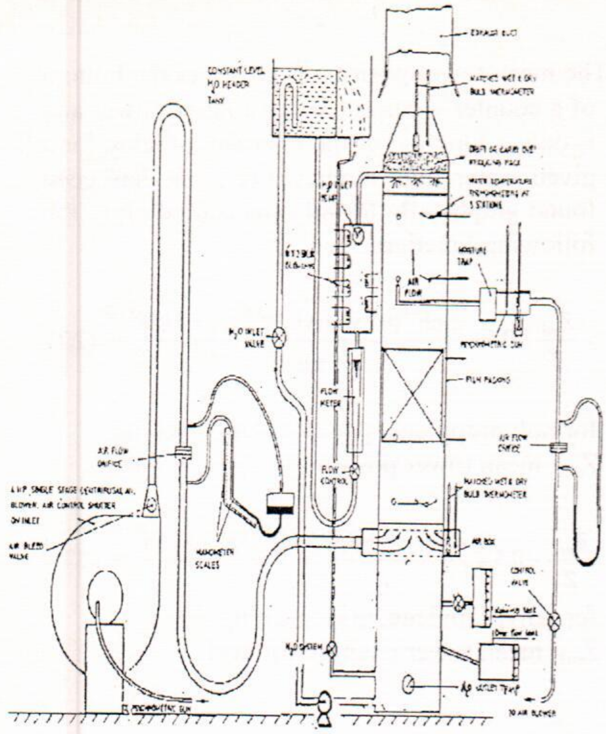


Fig.(1):Diagram of Experimental Cooling Tower System

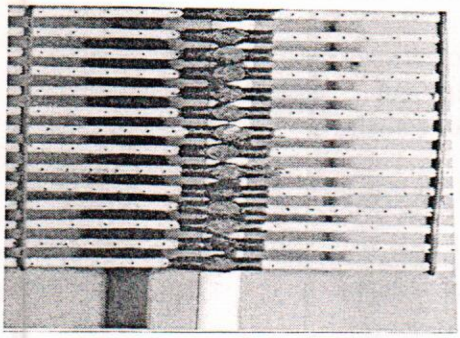
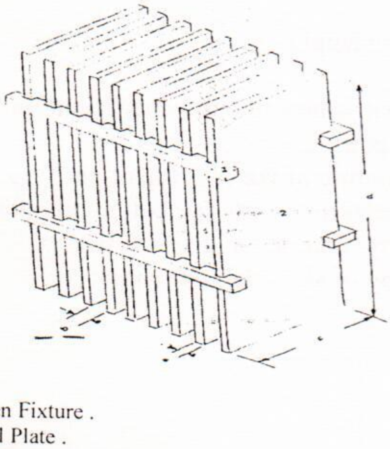


Fig.(2): Water Distribution System



1.Wooden Fixture .
2.Parallel Plate .
a-Plate Thin.= 6mm b-Plate Wid. 300 mm c-Plate Height
Fig.(3.a) The arrangement of Parallel Plate

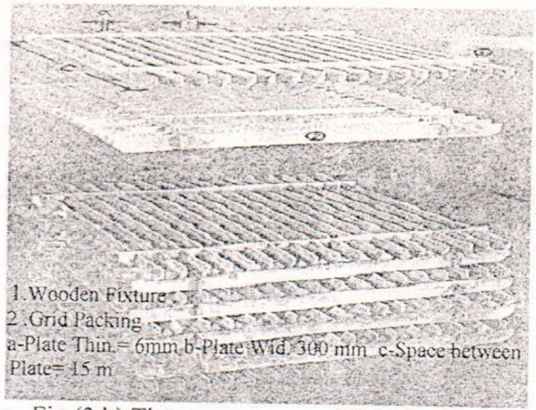


Fig.(3.b) The arrangement of Grid Packing

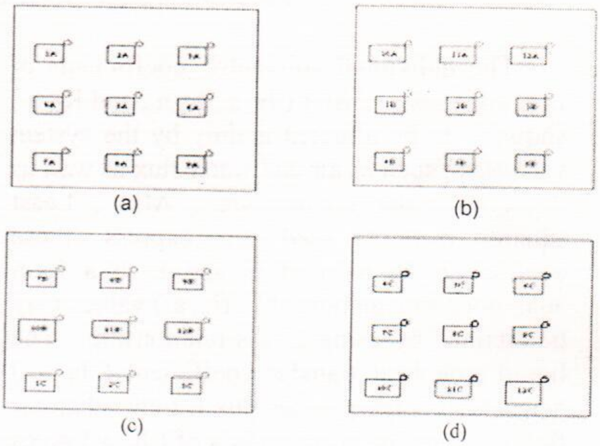


Fig.(4): Thermocouples Layer.
a- First Layer of Thermocouple.
b-Second Layer of Thermocouple.
c-Third Layer of Thermocouple.
d-Fourth Layer of Thermocouple.

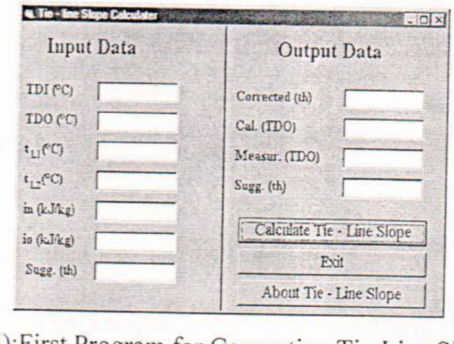


Fig. (5):First Program for Computing Tie-Line Slope

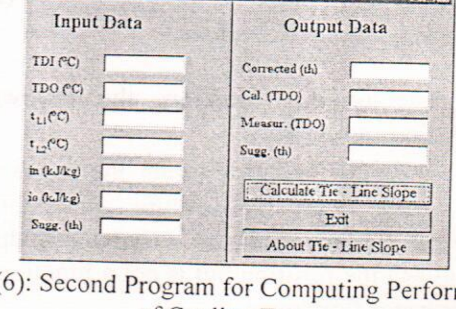


Fig. (6): Second Program for Computing Performance of Cooling Tower

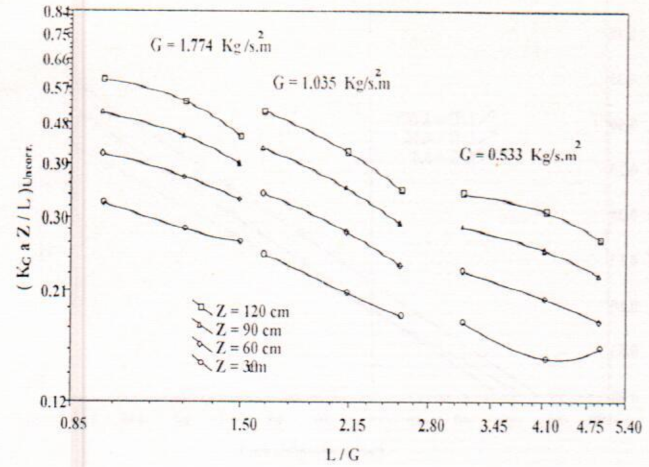


Fig (7): Uncorr. N.T.U vs. L / G for Polypropylene Gird Packing, t_{L2} = 40 °C , and Z = 30 cm.

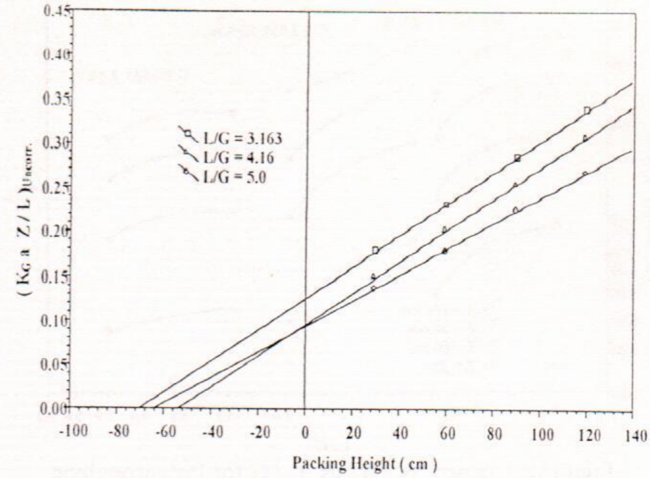


Fig.(10):Uncorr. N.T.U vs. L / G for Polypropylene Parallel Packing, G = 1.035 kg/ s.m^2 , and Z = 30 cm.

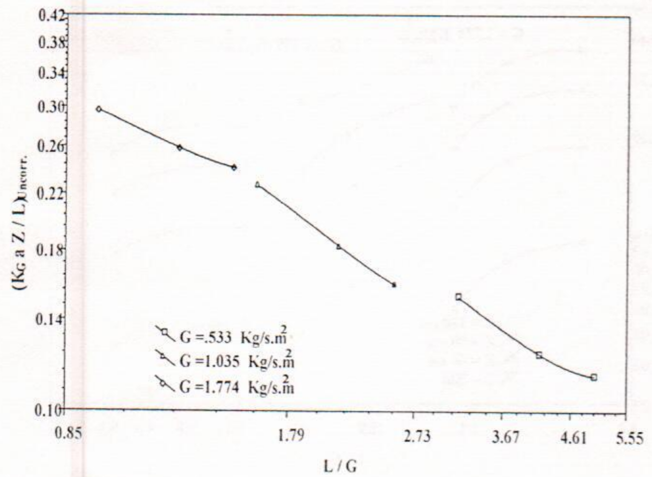


Fig.(8): Uncorr .N.T.U vs . L / G for Polypropylene Parallel Packing, t_{L2} = 40 °C , and Z = 30 cm.

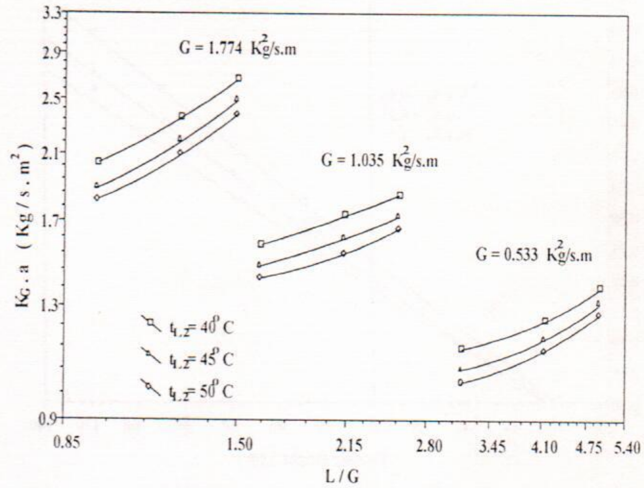


Fig.(11):Volumetric Mass Transfer Coefficient vs. L /G for Polypropylene Gird Packing, and Z = 30 cm .

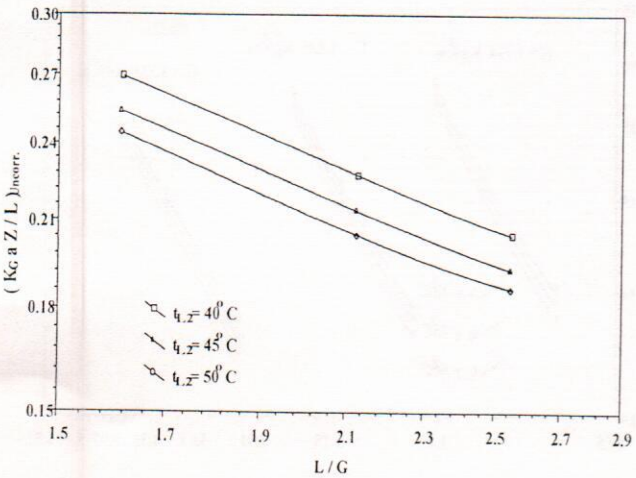


Fig.(9): Uncorr. N.T.U vs. L / G for PolypropylenGird Packing, G = 1.035 kg/ s.m^2 , and Z = 30 cm.

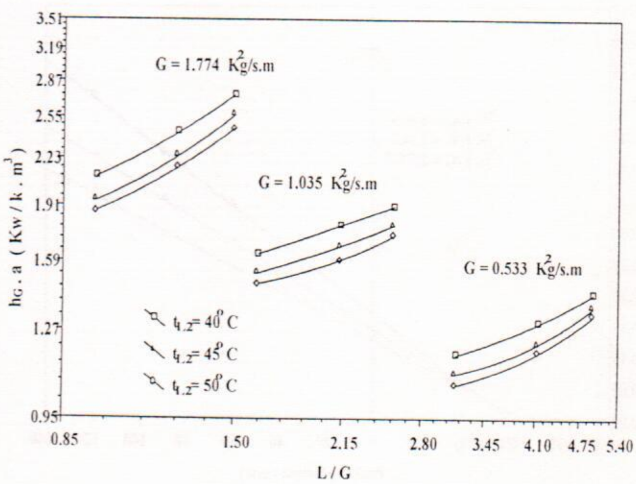


Fig.(12):Volumetric Heat Transfer Coefficient vs. L / G for Polypropylene Gird Packing, and Z = 30 cm

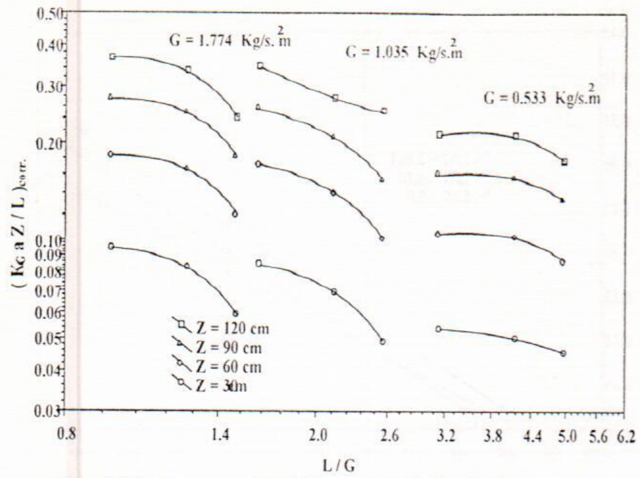


Fig.(13): Uncorr. N.T.U vs. L / G for Polypropylene Gird Packing, $t_{L2} = 40$ °C.

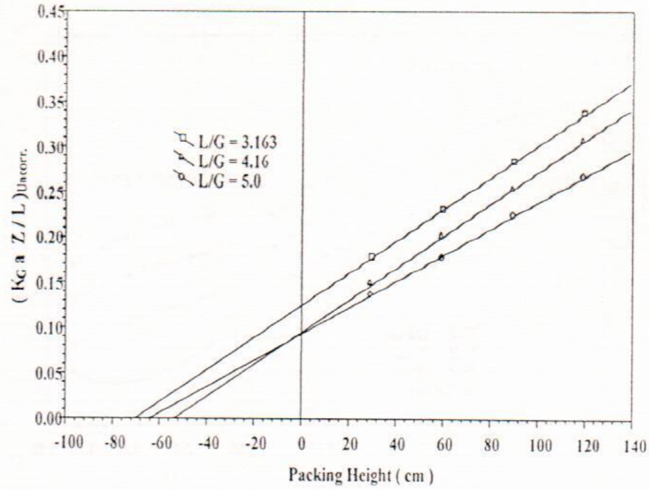


Fig.(16): Uncorr. N.T.U vs. Packing Height for Polypropylene Gird Packing , $t_{L2} = 40$ °C, and G = 0.533 kg/s.m^2.

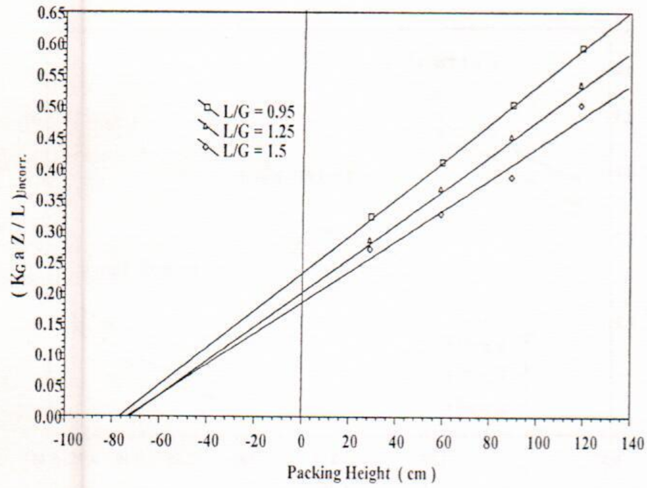


Fig.(14): Uncorr. N.T.U vs. Packing Height for Polypropylene Gird Packing , $t_{L2} = 40$ °C, and G = 1.774 kg/s.m^2

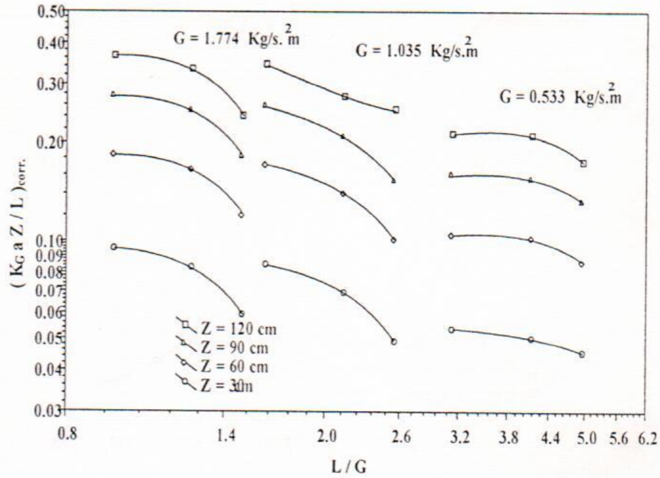


Fig.(17):Corr.N.T.U vs. L / G for Polypropylene Gird Packing, $t_{L2} = 40$ °C

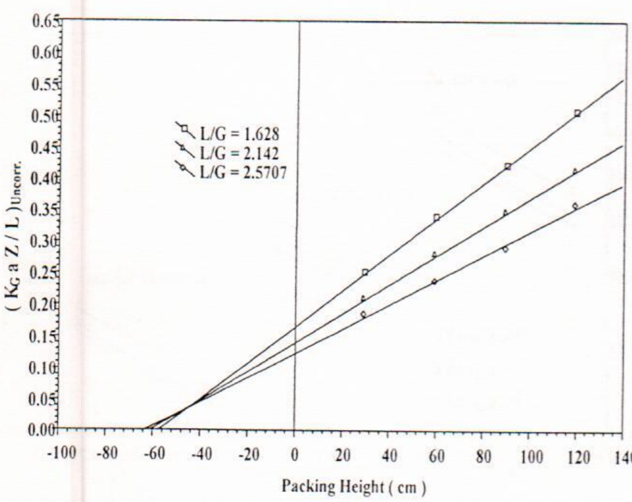


Fig.(15): Uncorr. N.T.U vs. Packing Height for Polypropylene Gird Packing , $t_{L2} = 40$ °C, and G = 1.035 kg/s.m^2.

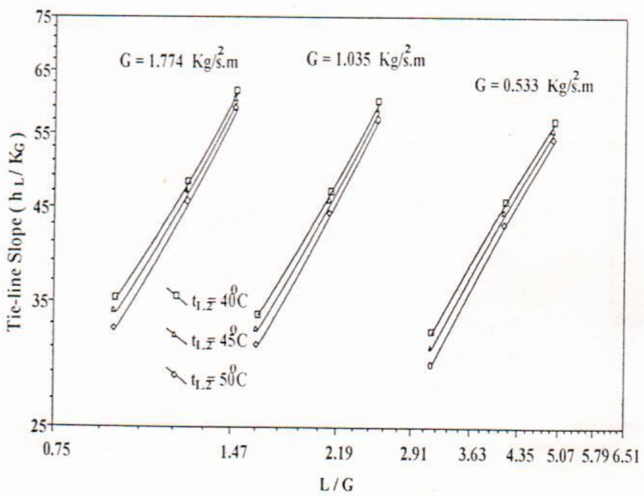


Fig.(18):Tie–line Slope vs. L / G for Polypropylene Gird Packing, and Z = 30 cm

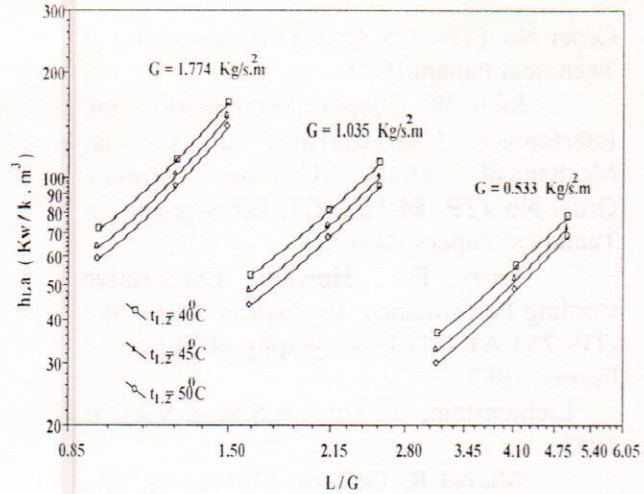


Fig.(19):Liquid Side Heat Transfer Coefficient vs. L / G for Polypropylene Gird Packing,and Z = 30 cm .

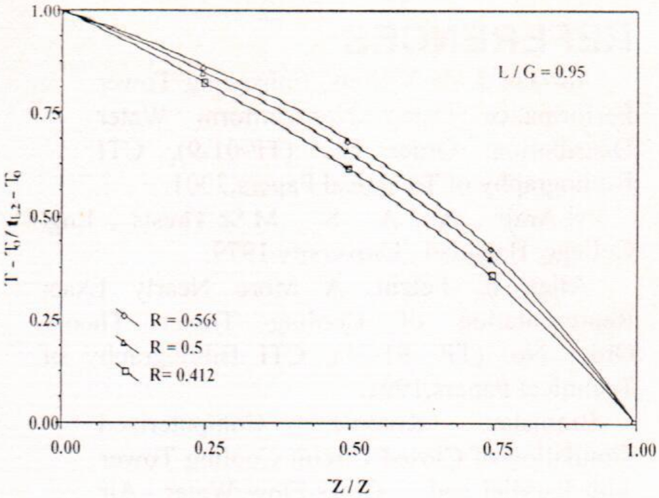


Fig.(22) : Temperature Profile Versus ¯Z / Z for Polypropylene Parallel Plate Packing at L / G = 0.95

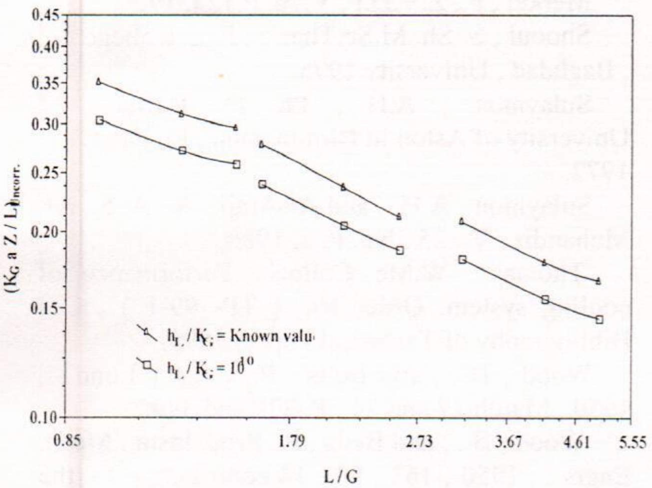


Fig.(20):Uncorrected NTU vs. L / G for Polypropylene Grid Packing at t_{L2}=40 °C ,and Z = 30 cm.

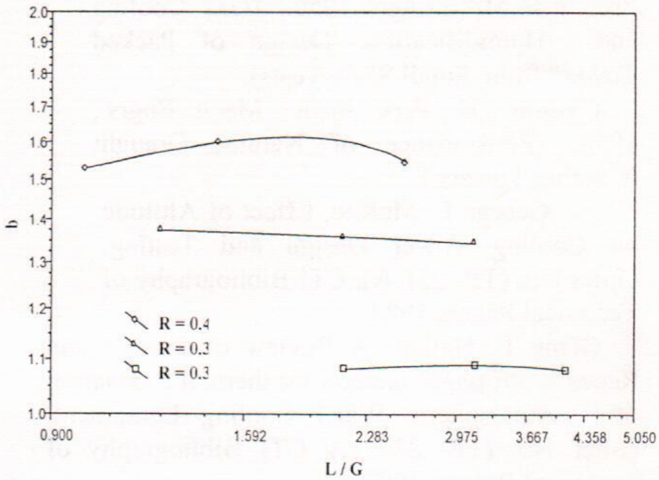


Fig.(23): Relation Between b and L / G at Different R for Polypropylene Grid Packing .

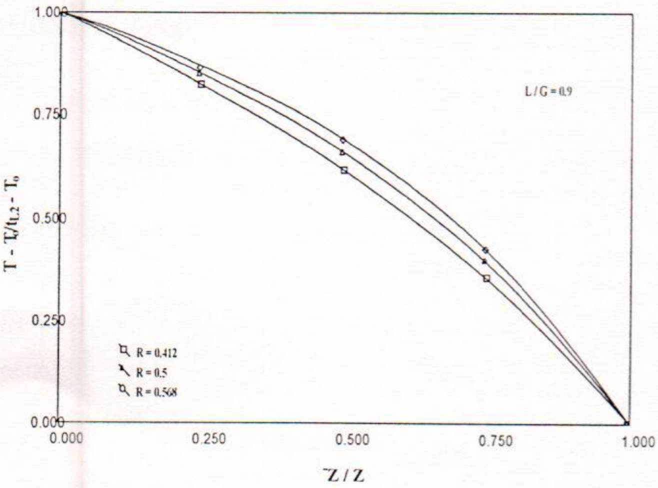


Fig.(21): Temperature Profile Versus ¯Z/Z for Polypropylene Grid Packing at L/G = 0.95 .

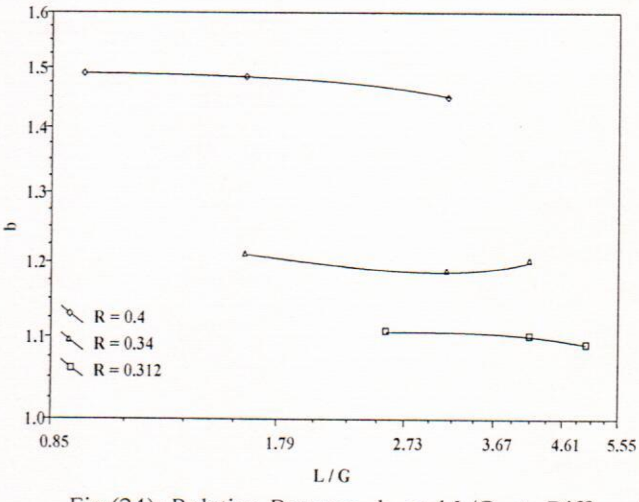


Fig.(24): Relation Between b and L/G at Different R for Polypropylene Parallel Packing

REFERENCES

Adriaan J. de Villiers, Enhancing Tower Performance Using Non-Uniform Water Distribution. Order No. (TP-01-9), CTI Bibliography of Technical Papers,2001.

Al-Araji , A . A . S ., M.Sc.Thesis , Eng. College, Baghdad , University 1979.

Allen E. Felzin, A More Nearly Exact Representation of Cooling Tower Theory Order No. (TP- 91-21), CTI Bibliography of Technical Papers,1991.

Branislav Korenic, Computerized Simulation of Closed Circuit Cooling Tower with Parallel and Cross-Flow Water –Air Flow Design. Order No. (TP- 95-11), CTI Bibliography of Technical Papers,1995.

Carey , W.F , and Williamson , C. J. Proc.Instn.Mech.Engrs.,1950., (Gas Cooling and Humidification: Design of Packed Towers from Small Scale Tests).

Chilton , H., Proc. Instn . Mech. Engrs., 1950, (Performance of Natural Draught (Cooling Towers).

George E. McKee, Effect of Altitude on Cooling Tower Design and Testing. Order No. (TP- 251 A), CTI Bibliography of Technical Papers,1982.

Glenn F. Hallett, A Review of Present and Recently Proposed method for thermal Evaluation of Atmospheric Water Cooling Equipment. Order No. (TP- 224 A), CTI Bibliography of Technical Papers ,1982.

James L . Willa, Misconceptions Concerning Cooling Tower Performance. Order No. (TP- 225 A), CTI Bibliography of Technical Papers,1982.

John W. Cooper, Recirculation and Interference Characteristics of Circular Mechanical Draft Cooling Towers. Order No .(TP- 84-18)), CTI Bibliography of Technical Papers,1984.

Larry F . Howlett, Evaporative Cooling Performance Evaluation. Order No. (TP- 253 A), CTI Bibliography of Technical Papers, 1982.

Lichtenstein, J., Tans. A.S.M.E, V.65, P.779, 1943.

Marcel R. Lefevre. Eliminating The Merkel Theory Approximations. Order No. (TP- 90-21), CTI Bibliography of Technical Papers,1982.

Merkel , F., Z.V.D.I , V.70, P.123, 1926.

Shooul , S. Sh. M.Sc.Thesis , Eng. College , Baghdad , University 1975.

Sulaymon , A.H , Ph. D. Thesis , University of Aston in Birmingham , England, 1972.

Sulaymon , A.H. , and Al-Araji , A . A. S., Al-Muhandis , V . 25 , No. P. 3, 1980.

Thomas W.Mc Colloch, Performance of cooling system. Order No. (TP- 99-1) , CTI Bibliography of Technical Papers,1999.

Wood , B . , and Betts , P., Engr. (Lond . , 1950 , March 17 and 24 , P. 339 and 349.

Wood , B . , and Betts , P., Proc. Instn . Mech. Engrs . , 1950 , 163 , 54 , 14 contribution to the theory of Natural Draught Cooling Towers .