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# Effect of Tropical Weather on the Thermal Capability of Cooling Towards Constructed in the Arabian Gulf Area.

A. El-Hadik
*Mechanical Engineering Department, Faculty of technology, Studien, kuwait,*

A. El-Obaid
*Chemical Engineering Department, Faculty of Technology Studien, Kuwait*

# "EFFECT OF TROPICAL WEATHER ON THE THERMAL CAPABILITY OF COOLING TOWERS CONSTRUCTED IN THE ARABIAN GULF AREA"

# تاثير المناخ القارى على المقدرة الحرارة لاداء ابراج التبريد التى تعمل بدول الخليج العربى

A.A.El-Hadik & A.M.Al-Obaid
Mechanical Eng. Dept., Faculty of Tech.Studies Kuwait Kuwait.
Chemical Eng. Dept., Faculty of Tech.Studies Kuwait.

الخلاصة : تبحث هذه المقالة فى تأثير المناخ القارى على المقدرة الحرارية لابراج التبريد المنشاة فى دول الخليج العربى . وقد تمت هذه الدراسة تحليليا وتجريبيا . درجة الحرارة الجافة والرطوبة النسبة المرتفعتان . قد تصل قيمتهما فى بعض الاحيان الى ٦٠ م ، ١٠٠٪ على التوالى ( طبقا للمعلومات المناخية بدولة الكويت على سبيل المثال ) لها تاثير واضح على المعدل التبخيرى للمياه المستخدمة فى أبراج التبريد هذه :

ودرجة الحرارة المبتلة للهواء عند الدخول لها تأثير ظاهر على حجم برج التبريد والاتزان الحرارى للنظام . معدل السريان والحمل الحرارى يحددان مقدرة البرج الحرارية وقد تمت . هذه الدراسة على عينة من أبراج التبريد من نوع سيراميك حيث أنها من الأنواع المنتشرة للعمل فى منطقة الخليج العربى

كما وقد تمت المقارنة بين كل من النتائج التحليلية والمعملية مع المعلومات التصنيعية ووجد ان هناك تباين واضح بينهم .

ABSTRACT

This paper studies the affect of tropical weather on the thermal capability of cooling towers constructed in the ARABIAN GULF AREA. This study has been carried out numerically and experimentally.

High dry-bulb temperature, and relative humidity(Sometimes reaches 60 C , and 100% respectively according to Kuwait climatology data) have a significant effect on the rate of water evaporation

The entering air wet-bulb temperature and required system temperature level combine with cooling tower size to balance the heat rejection system at a specified approach. The combination of flow rate and heat load dictates the range which a cooling tower must accommodate.

The ceramic type of cooling towers is chosen as a sample of this study, because this type is used extensively in the arabian gulf area.

Finally, the obtained results have been compared with the manufactury results, and noticeable deviations have been found.

NOMENCLATURE :

| | |
|---|---|
| $a_E, a_W, a_N, a_S$ | = Coefficients of finite difference equation |
| $C_{P.air}$ | = Specific heat of dry air, J/Kg °C |
| $C_{P.\ vapor}$ | = Specific heat of vapor, J/Kg °C |
| $C_{P_G}$ | = Specific heat of moist air, J/Kg °C |
| $C_{P_w}$ | = Specific heat of water, J/Kg °C |
| $f_G$ | = Moisture fraction of moist air, Kg/kg |
| $f_G^*$ | = Moisture fraction of dry air Kg/Kg |
| $f_s$ | = Moisture fraction of saturated moist air, Kg/Kg |
| $f_x, f_y$ | = Resistances to air flow in x and y-directions respectively, N/m³ |
| G | = Mass flow of dry air, Kg/s |
| G' | = Mass flux of dry air, Kg.m²s |
| g | = Gravitational acceleration, m/s² |
| H | = Heat transfer coefficient, W/m² °C |
| $h_G$ | = Specific enthalpy of moist air, J/Kg |
| $h^*_G$ | = Specific enthalpy of dry air, J/Kg |
| $h_T$ | = Specific enthalpy of transferred substance J/Kg |
| $h_{fg}$ | = Latent heat of vaporization, J/Kg |
| K | = Mass transfer coefficient ,Kg/m² s |
| L | = Mass flow rate of water, Kg/s |
| L' | = Mass flux of water, Kg/m²s |
| $\dot{m}_v'''$ | = Rate of mass transfer per unit volume, Kg/m³ s |
| P | = Pressure , $P_a$ |
| R | = Universal gas constant, J/Kg$_{-mol}$ °K |
| $Q_{th}$ | = Total heat transferred from water to air, W |
| $\dot{q}'''$ | = Rate of heat transfer per unit area of transfer surface, W/m² |
| T | = Temperature, °C |
| u | = Vertical component of velocity, m/s. |
| v | = Horizontal component of velocity, m/s |
| x | = Vertical cartesian coordinate, m |
| y | = Horizontal cartesian coordinate, m |

| | |
|---|---|
| $\Gamma_{eff}$ | = Effective exchange coefficient, Kg/ms |
| $\mu_{eff}$ | = Effective viscosity, Kg/ms |
| $\rho$ | = Density of moist air, Kg/m$^3$ |
| $\sigma_{eff}$ | = Effective Prandtl number, dimensionless |
| $\Phi$ | = Dependent variable |

Subscript

| | |
|---|---|
| A | = Air |
| amb | = Ambient |
| DB | = dry-bulb |
| F, W | = water |
| G | = moist air |
| in | = entering, |
| out | = leaving |
| S | = saturated |
| WB | = wet bulb |

## 1. INTRODUCTION :

Most air conditioning systems and many industrial processes generate heat which must be removed and dissipated. Water is commonly used as a heat transfer medium to remove heat from refrigerant condenser or industrial process heat exchangers.

A cooling tower is a device that uses a combination of heat and mass transfer to cool water. The thermal performance of a cooling tower is affected by weather conditions. Atmospheric wet-bulb has a significant effect on the cooling tower thermal capability, while the other atmospheric conditions such as dry-bulb and relative humidity have a great effect on the rate of water evaporation.

The thermal capability of any cooling tower may be defined by the following parameters :

1- Entering and leaving water temperatures.

2- Entering air wet-bulb or entering air wet-bulb and dry-bulb temperatures.

3- Water flow rate.

The direct-contact cooling tower is commonly used as counter-flow heat transfer, type, water is downflow, while the air may be upflow.

In comparison with most other industrial equipment, the water cooling tower is a simple device, based on the direct contact of two of the earth's most common substances, air and water. In 1983 Willa James et Al [9]. discussed the following topics on the cooling towers:

service factors, drift, losses, evaporation; and performance changes with humidity. Also, in 1987, Markatos et al [7] developed the computer model to study thermal energy released into the environment. A typical application of their model is the study of the behavior of cooling tower enffluent under different weather and operating conditions. In May 1985, Zafar et al [6] presented a linear approximate model of wet surface heat exchangers for the effect of Lewis number.

## 2. ANALYTICAL METHOD :

The present model treats airflow to be two-dimensional and cartesian, while the water flow is considered to be one dimensional. It obtains simultaneous solution of conservation equations for :

a) Mass continuity of air

b) Mass fraction of moisture in air

c) Mass continuity of water

d) Air momentum in vertical direction

e) Air momentum in active horizontal direction

f) Enthalpy of air

g) Enthalpy of water

Figure 1 shows the calculation domain. Following are the governing conservation equations in Cartesian coordinates.

Mass of Air :

$$\frac{d}{dx}(\rho u) + \frac{d}{dy}(\rho v) = \dot{m}_v''' \quad \ldots (1)$$

Mass of water

$$\frac{d}{dx}(\rho_f u_f) = \dot{m}_v''' \quad \ldots (2)$$

x- Direction Momentum :

$$\frac{d}{dx}(\rho u^2) + \frac{d}{dy}(\rho uv) - \frac{d}{dx}(\mu_{eff}\frac{dv}{dx}) - \frac{d}{dy}(\mu_{eff}\frac{dv}{dy})$$

$$= -\frac{dp}{dx} - f_x - (\rho - \rho_{amb})\, g \quad \ldots (3)$$

Y-Direction Momentum :

$$\frac{d}{dx}(\rho uv) + \frac{d}{dy}(\rho v^2) - \frac{d}{dx}(\mu_{eff}\frac{dv}{dx}) - \frac{d}{dy}(\mu_{eff}\frac{dv}{dy})$$

$$= -\frac{dp}{dy} - f_y \quad (4)$$

Air Enthalpy :

$$\frac{d}{dx}(\rho u\; h_G) + \frac{d}{dy}(\rho v\; h_G) - \frac{d}{dx}\left(\Gamma_{eff}\frac{dh_G}{dx}\right) - \frac{d}{dy}\left(\Gamma_{eff}\frac{dh_G}{dy}\right) = \dot{q}^{m} \qquad ..(5)$$

Water Enthalpy :

$$\frac{d}{dx}(\rho_f\; u_f\; h_f) = -\dot{q}^{m} \qquad ..(6)$$

Moisture Fraction of Air :

$$\frac{d}{dx}(\rho u\; f_G) + \frac{d}{dy}(\rho v\; f_G) - \frac{d}{dx}\left(\Gamma_{eff}\frac{d f_G}{dx}\right) - \frac{d}{dy}\left(\Gamma_{eff}\frac{d f_G}{dy}\right) = \dot{m}_v^{m} \qquad ..(7)$$

Equation of State :

$$\rho = \frac{P\; w_G}{R\;(T_{DB} + 273)} \qquad ..(8)$$

The following features of the conservation equations (1-8) may be noted. All conservation equations for air are coupled through connective fluxes ( $\rho u$ and $\rho v$). In addition, momentum equations (3) and (6) are coupled through pressure. The gravity term in equation (3) uses density differences ($\rho - \rho_{amb}$) rather than density ( $\rho$ )
The pressure P, used in these equations is reduced pressure (see ref.[1] i.e., the relative pressure with reference to the ambient pressure at the same elevation. This practice of using reduced pressure and density difference is based on the exact transformations employing the following relation :

$$P = P_{static} + \rho_{amb}\; g\; x$$

$\dot{m}_v^{m}$ and $\dot{q}^{m}$ represent the sources of mass and enthalpy, and $f_x$ and $f_y$ represent resistances to air flow due to the presence of solid obstacles. Expressions for calculating $\dot{m}_v^{m}$, $\dot{q}^{m}$, $f_x$, and $f_y$ are described later.

$h_G$ is specific enthalpy of moist air, and $f_G$ is the fraction of vapor content of moist air. Since hydrodynamic equations are solved for velocities and pressure of moist air, it is appropriate to solve for specific enthalpy and moisture fraction of moist air. However, specific enthalpy of dry air, $h^*_G$ , and moisture fraction of dry air, $f^*_G$ , can be evaluated from the following expressions

$$h^*_G = h_G / (1 - f_G) \qquad \ldots (9)$$

$$f^*_G = f_G / (1 - f_G) \qquad \ldots (10)$$

3.1 Finite Difference Equations:

The calculation domain is subdivided into finite number of control cells (Fig.2). The finite difference equations are obtained by integrating the partial differential equation over the finite volume represented by a cell.

Typical grid distributions for a mechanical draft crossflow cooling tower have been shown in Fig. 2. Nonuniform grid distributions are employed with larger number of control cells located in the fill region. All scaler quantities (such as P, $h_G$ , $f_G$ , etc) are calculated at the center of control cells; and velocity components (u and v) are calculated at the cell faces.

The general form of finite difference equation is

$$\Phi_p = \frac{a_E \Phi_E + a_W \Phi_W + a_N \Phi_N + a_s \Phi_s + a_u}{a_E + a_W + a_N + a_s - S_p} \qquad \ldots (11)$$

where $\Phi$ stands for any dependent variable such as u, v, $h_G$, $h_f$, and fG ; and link coefficients $a_E$ , $a_W$ , $a_N$ and $a_s$ express the effects of convection and diffusion between grid point, P, and its neighboring grid nodes in East, West, North, and South direction, respectively. $S_u$ and $S_p$ are the components of source terms, s, which is linearized as

$$S = S_u + S_p \Phi_p$$

3.2 Boundary Conditions :

The following quantities are specified as system boundary conditions :

1- Hot water flow rate

2- Either cooling range or hot water temperature

3- Dry bulb temperature, $T_{DB}$

4- Wet-bulb temperature, $T_{WB}$

5- Ambient pressure

The specified conditions for each dependent variable at all four boundaries of the calculation domain are also shown in Fig. 2.

The moisture fraction and enthalpy of inlet air are calculated from the dry-bulb and wet-bulb temperatures by employing the following thermodynamic relations.

$$f_{G,amb} = fs - \frac{C_G\ L_e\ (T_{DB} - T_{WB})\ (1 - f_s)}{h_{fg}} \quad . . \quad (12)$$

$$h_{G,\ amb} = (1 - f_{G,\ amb})\ C_G\ T_{DB} + f_{G,\ amb}\ h_{FG} \quad . . \quad (13)$$

Pressures are specified at the inlet and outlet boundaries, while the velocities are calculated from the local differences between the ambient pressure and the pressure inside the tower. Pressures at the inlet and outlet sections of a tower are the same as that of ambient. For mechanical draft towers, the fan and stack are simulated by way of a "point" model, i.e., no distributions of flow variables are calculated in the stack. The pressure at the bottom of the stack, $P_{bs}$, is calculated from Bernoulli's equation by considering the input power and efficiency of fan and the area changes in stack. The final form is

$$P_{bs} = P_{amb} - \frac{\rho\ \eta_{fan}\ P}{\dot{m}_G} - 1/2\ \rho\ u^2_{fan} \left[ \left( \frac{A_{fan}}{A_{plenum}} \right)^2 \right.$$

$$\left. \left( \frac{A_{fan}}{A_{stack}} \right)^2 - N_{stack} \right] - (\rho_{amb} - \rho) g\ H_{stack} \quad . . \quad (14)$$

where P is the input power to fan and $N_{stack}$ is the number of velocity heads ( $= \rho u^2_{fan} / 2$) lost in the stack.

### 3.4 Solution Procedure:

Since the governing equations are coupled and nonlinear, they have to be solved by means of an iterative procedure. An implicit solution scheme based on the procedures of [1] is employed. The convergence of numerical scheme is checked by the normalized residual errors of the conversation equations in each control cell of the calculation domain.

## 4. Experimental Apparatus and Procedure:

Fig. 3 is an outline of the experimental apparatus. Air is drawn into the cooling tower by mechanical draft fan (5) which is driven by by an electrical motor, three phase, 380 volt, and 50 Hz. The supply current is controlled by varying resistance (rheostat), so by reading the current and resistance the power consumption of an motor can be calculated. Hot water is pumped into the spray nozzles (6), through a calibrated flowmeter (3). Therefore, the hot water flow rate can be measured. Also the hot water temperature may be measured by a calibrated thermocouple (4), located at the hot water inlet. Part of heat is transferred from water into air by evaporating mass. So the amount of evaporation water can be measured by a calibrated flow-meter (1) located at the inlet of make-up water. There is a ceramic fill (7), for making a good mixing between water and air. The exit water temperature can be measured by calibrated thermocouple (8). Therefore, the second part of heat which is transferred from water into air can be calculated by :

$$Q_{th} = ( L_{in} - L_{out} ) *[C_{PW}* (T_{in} - T_{out} ) + h_{fg} \quad .. (15)$$

By running up the experimental apparatus during the twelve months of whole year 1988. The following data can be given as a sample of these readings.

Table : 1 Temperature Relationship Between Water and Air in a Cooling Tower Constructed in Kuwait City at 31st August 88.

| Local Time hr. | Hot Water inlet Temp°C | Cold Water Outlet Temp°C | Am-bient Air Wet-bulb Temp C | Am-bient Air dry-bulb Temp C | Am-bent air rela tive humi % | Hot Water flow rate experi mental data lit/s | Hot Water flow rate design data Lit/s |
|---|---|---|---|---|---|---|---|
| 6 | 36.66 | 32.22 | 28.4 | 31 | 84 | 189.3 | 145 |
| 7 | 36.67 | 31.67 | 28.5 | 32 | 77 | 189.3 | 125 |
| 8 | 37.22 | 32.22 | 28 | 33.3 | 75 | 189.3 | 170 |
| 9 | 39.99 | 34.44 | 24.4 | 35.6 | 40 | 189.3 | 1119.2 |
| 10 | 33.33 | 27.78 | 25.1 | 36.3 | 40 | 189.3 | 158.5 |
| 11 | 33.33 | 27.78 | 29 | 39 | 48 | 170.3 | 112.7 |
| 12 | 34.99 | 29.44 | 27.6 | 39.9 | 40 | 170.3 | 148.5 |
| 13 | 38.66 | 33.11 | 29.1 | 40.5 | 43 | 170.3 | 109 |
| 14 | 36.66 | 31.11 | 29.5 | 41.2 | 41 | 170.3 | 131.67 |
| 15 | 36.65 | 31.11 | 30.0 | 39.7 | 47 | 161 | 100.6 |
| 16 | 36.11 | 30.56 | 28.8 | 39 | 47.5 | 161 | 131 |
| 17 | 36.66 | 31.11 | 30.2 | 38.4 | 56 | 132.5 | 94.3 |
| 18 | 36.11 | 30.56 | 30.1 | 37.4 | 59 | 132.5 | 96.7 |
| 19 | 35.56 | 30.56 | 30.2 | 35.4 | 69 | 132.5 | 96.7 |
| 20 | 35 | 30 | 29.2 | 34.8 | 69 | 132.5 | 96.7 |
| 21 | 35 | 30 | 29 | 34.1 | 69 | 132.5 | 96.7 |

Results and Discussions:

The experiment was carried out over the ranges of the atmospheric air condition wet-bulb temperature from 17.5 to 32.5°C , dry-bulb temperature from 30 to 50°C , and relative humidity from 12% to 96%.

Figures 4, 5 and 6 show the temperature relationships between water and air in a counterflow cooling tower installed in Kuwait city at 23rd July, 31st August, and 31st July 1988 respectively. Effect of ambient air dry-bulb temperature on the approach and range temperatures is shown in these figures. By increasing the dry-bulb temperature decreases the approach temperature and increases the range temperature.

The thermal capability of a cooling tower is affected also by the entering air wet-bulb temperature. Figures 4, 5 and 6 are displayed by increasing the ambient wet-bulb temperature decreases the approach temperature and increase the range temperature, when the dry-bulb temperature and relative humidity remain constant. While the entering air relative humidity has an insignificant effect on the cooling tower thermal performance but it is affected on the rate of water evaporation. Fig. 7 shows water flow rate of ceramic type of cooling tower installed in Kuwait city at 23rd November 1988. There is a comparison between actual amount (experimental data), and hypothetical amount (design data) of water flow rate for small cooling tower types (waterflow rate less than 150 lit/S), the difference is too large, while for big units (water flow rate more than 200 lit/S) the design and experimental data are nearly the same amount of water flow rate. Fig. 8 shows the typical one of ceramic type cooling tower, which is installed at Ministries Complex in Kuwait city and is chosen for this study.

5. Conclusions:

The results present in this work may be summarized as follows:

1- In general, two-dimensional mathematical model applicable for mechanical draft cooling tower of counter flow arrangements has been described.

2- During whole year some experimental data are recorded and compared with theoretical one.

3- Entering air wet and dry-bulb temperatures have a significant effect on the thermal capability of cooling towers, while the entering air relative humidity has no effect.

REFERENCES :

1- Majumdar A.K., Singhal A.K., and Spalding D.B (November 1983) "Numerical Modeling of wet cooling towers-part 1 : Mathematical and Physical Models". ASME Journal of heat transfer, Vol. 105, pp. 728-735.

2- Majumdar A.K., Singhal A.K., Reilly H.E., and Bartz J.A. November 1983) " Numerical modeling of wet cooling tower-Part 2: Application to Natural and Mechanical Draft Towers,. "ASME. Journal of Heat Transfer. Vol. 105, pp. 736-743.

3- Fahim M.A., Al-Ameeri R.S., and Wakao N. (December 1986) : Equations for calculating Temperatures in Adiabatic Air-water contact towers". Chem. Eng. Commun. Vol. 36, pp. 1-8.

4- Stewart William E. JR., and Smith Donald R. (1984) "Cooling Tower Evaporation Rates at High Site Elevations. "Heat Transfer Engineering Vol. 5 nos. 3-4 pp. 48-55.

5- Aihara T., and Sago R. (February 1983) "Performance of a compact Colling Unit Utilizing iAir-Water Mist Flow". ASME. Journal of Heat Transfer, Vol. 105, pp. 18-24.

6- Zufar M., Maclaine-Cross I.L., (May 1985) "General Theory of Wet surface Heat Exchangers Modified for the Effect of Lewis Number" Third Australasian Conference on heat and Mass Transfer Melbourne Aust. PP. 513-520.

7- Markatos N.C., Pericleous K., and Simitovic R., "(March 1987) : Hydrometeeorogical, Three-Dimen-sional Model of Thermal Energy Releases into Environmental Media" Int. Journal for Numerical Methods in fluids, Vol. 7. N.3., PP. 263-276.

8- Lotevre Marcel R. (August 1984): "Heat Transfer Technology Atmospheric Cooling Equipment, Water Consumption." American Institute of Chemical Engineers, Summer national meeting, Philadelpphia, USA PP. 19-22.

9- Willa James L., and Campbell John C. (December 1983). "Common Misconceptions About Cooling Towers." Chemical Engineering Progress. Vol. 79, n 12, PP. 23-27.

10- "Cooling Towers" ASHRAE Handbook 1983 Equipment Volume. PP.21.1 - 21.18. American Society of Heating, Refreigerating and Air Conditioning Inc. Georgia, U.S.A.

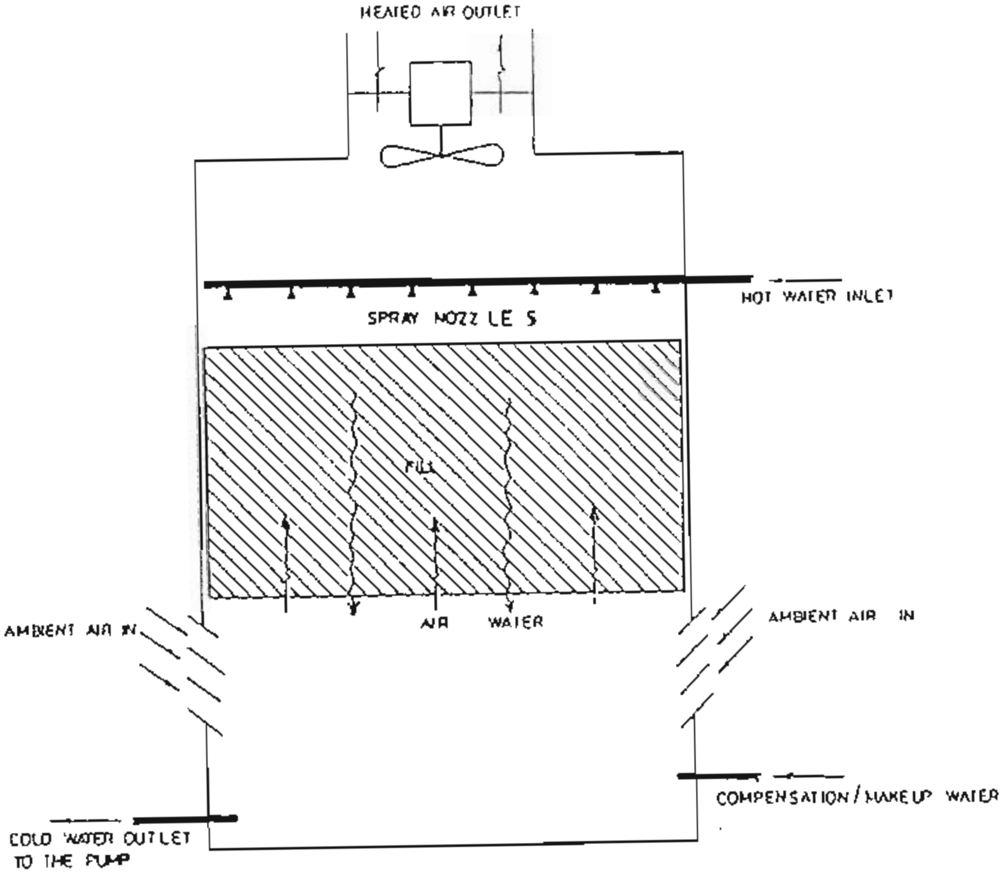



FIG-1. THE COORDINATE SYSTEM FOR RECTANGULAR MECHANICAL DRAFT COUNTER FLOW COOLING TOWER

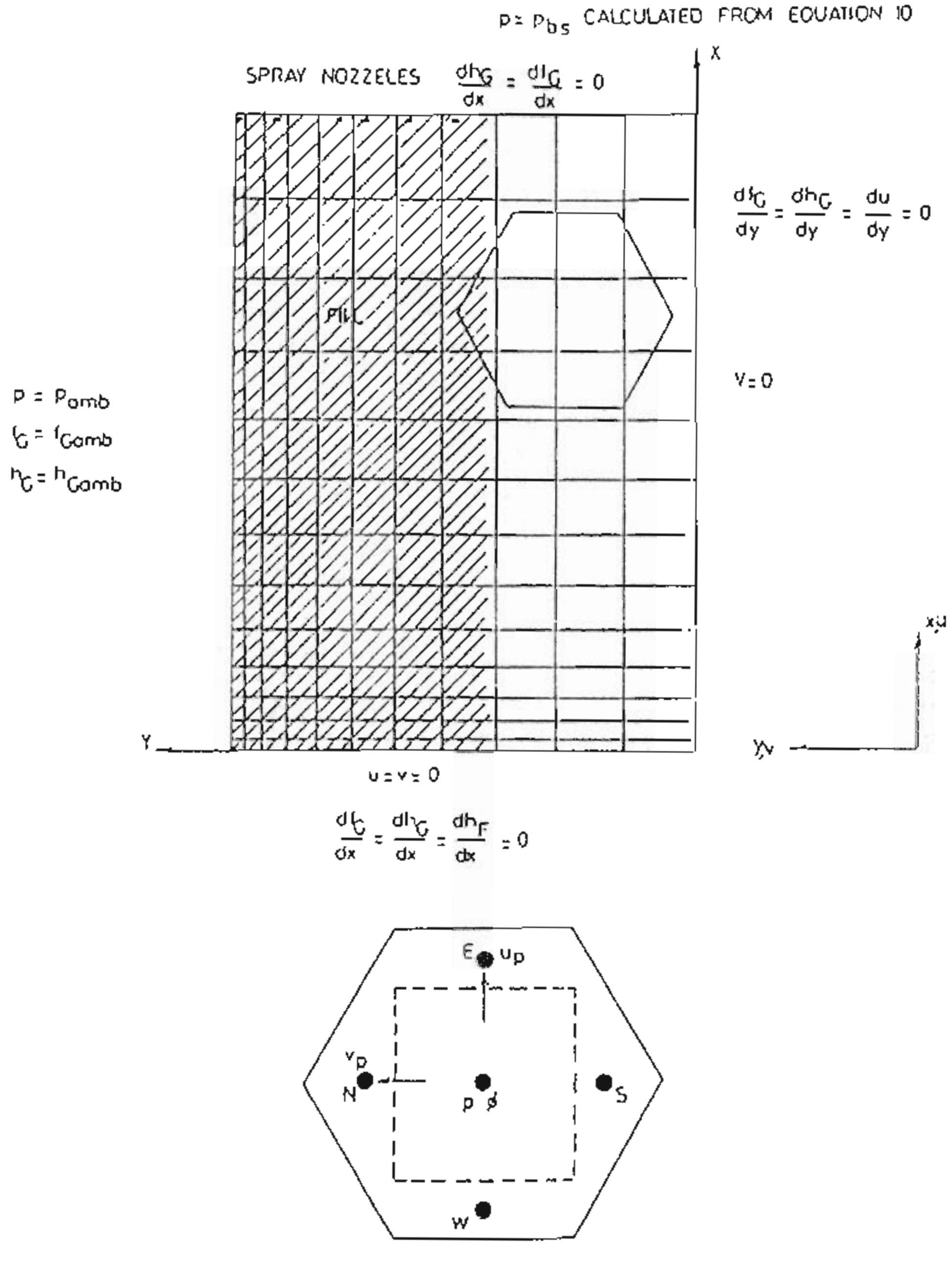



FIG-2. CALCULATION DOMAIN GRID LAYOUT AND BOUNDARY CONDITIONS FOR MECHANICAL-DRAFT COUNTER FLOW COOLING TOWER

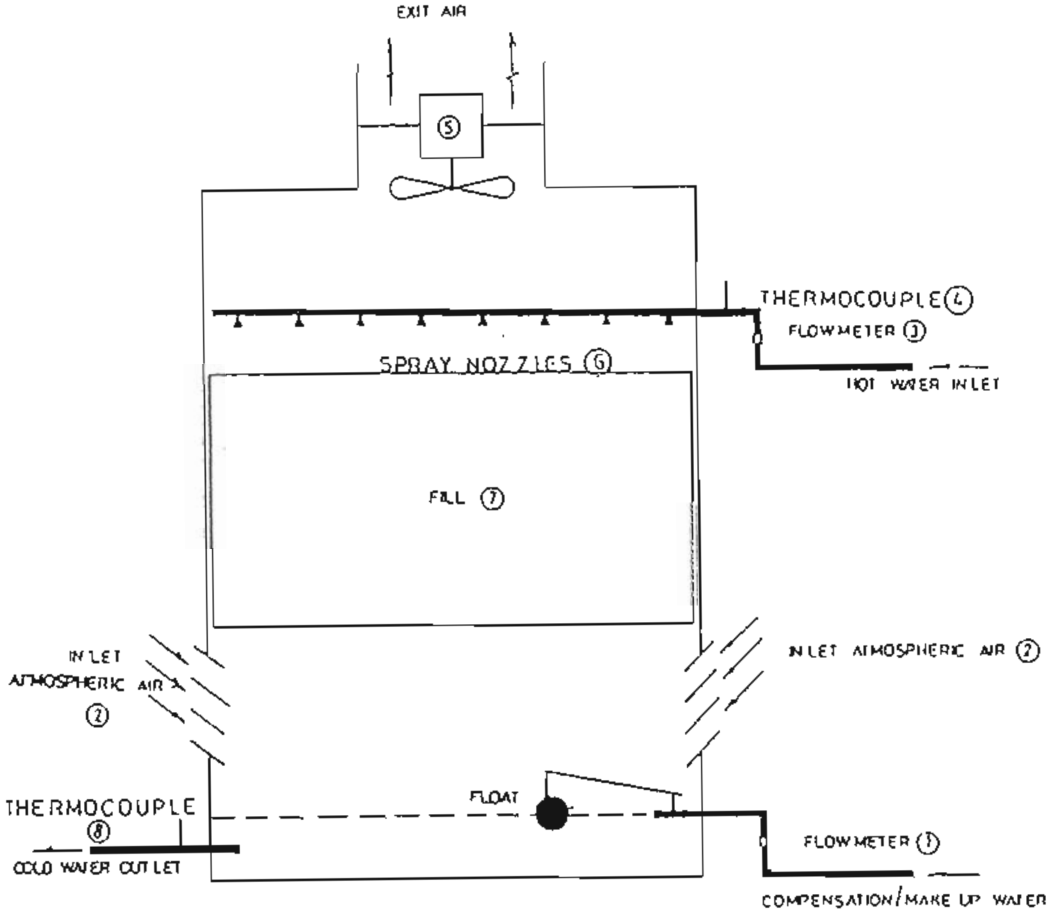



FIG-3. OUTLINE OF EXPERIMENTAL APPARATUS CONVENTIAL MECHANICAL INDUCED DRAFT COUNTER FLOW COOLING TOWER

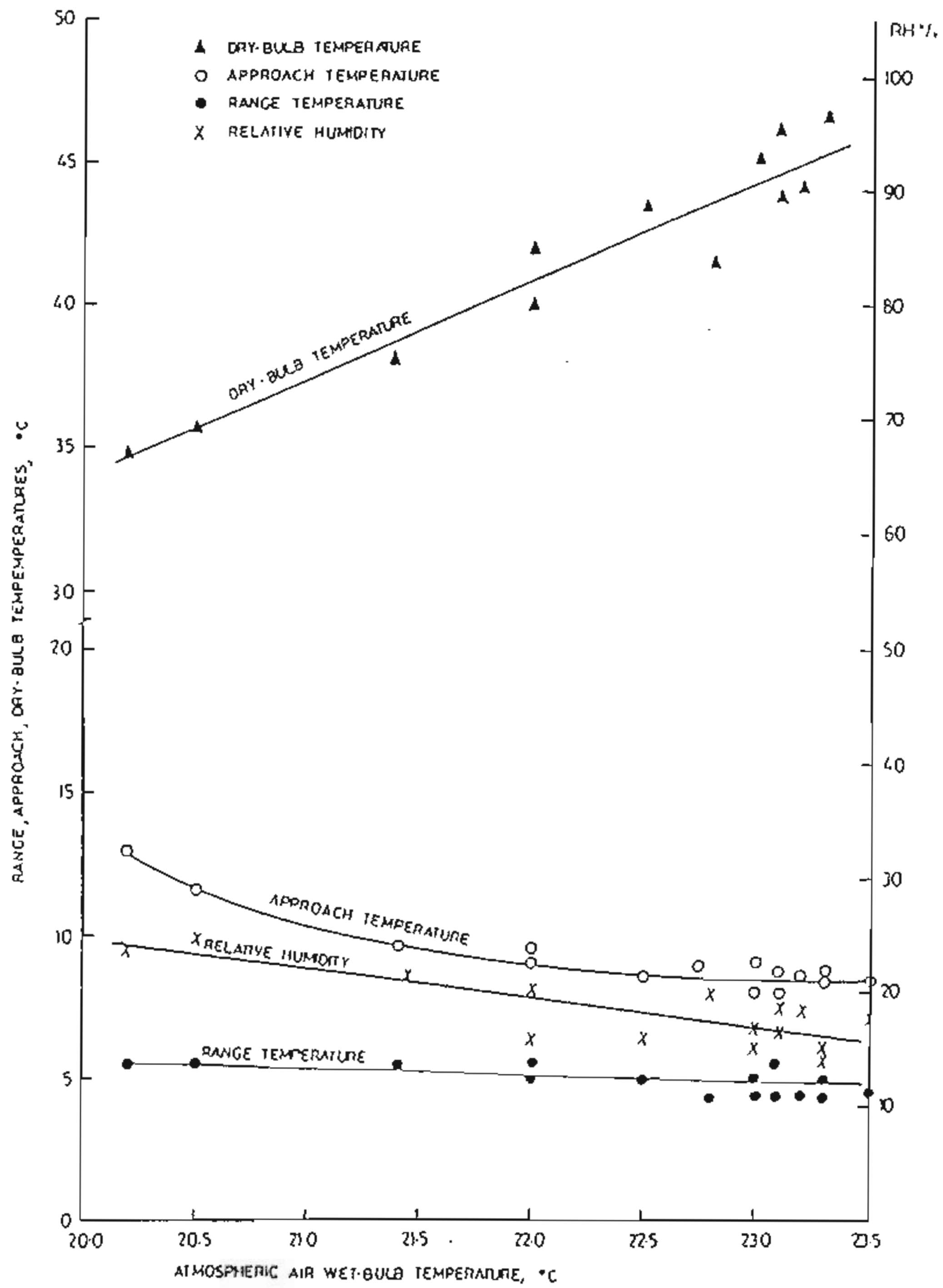



FIG 4 TEMPERATURE RELATIONSHIP BETWEEN WATER AND AIR IN A COUNTER FLOW COOLING TOWER INSTALLED IN KUWAIT CITY ON 23rd JULY 1988.

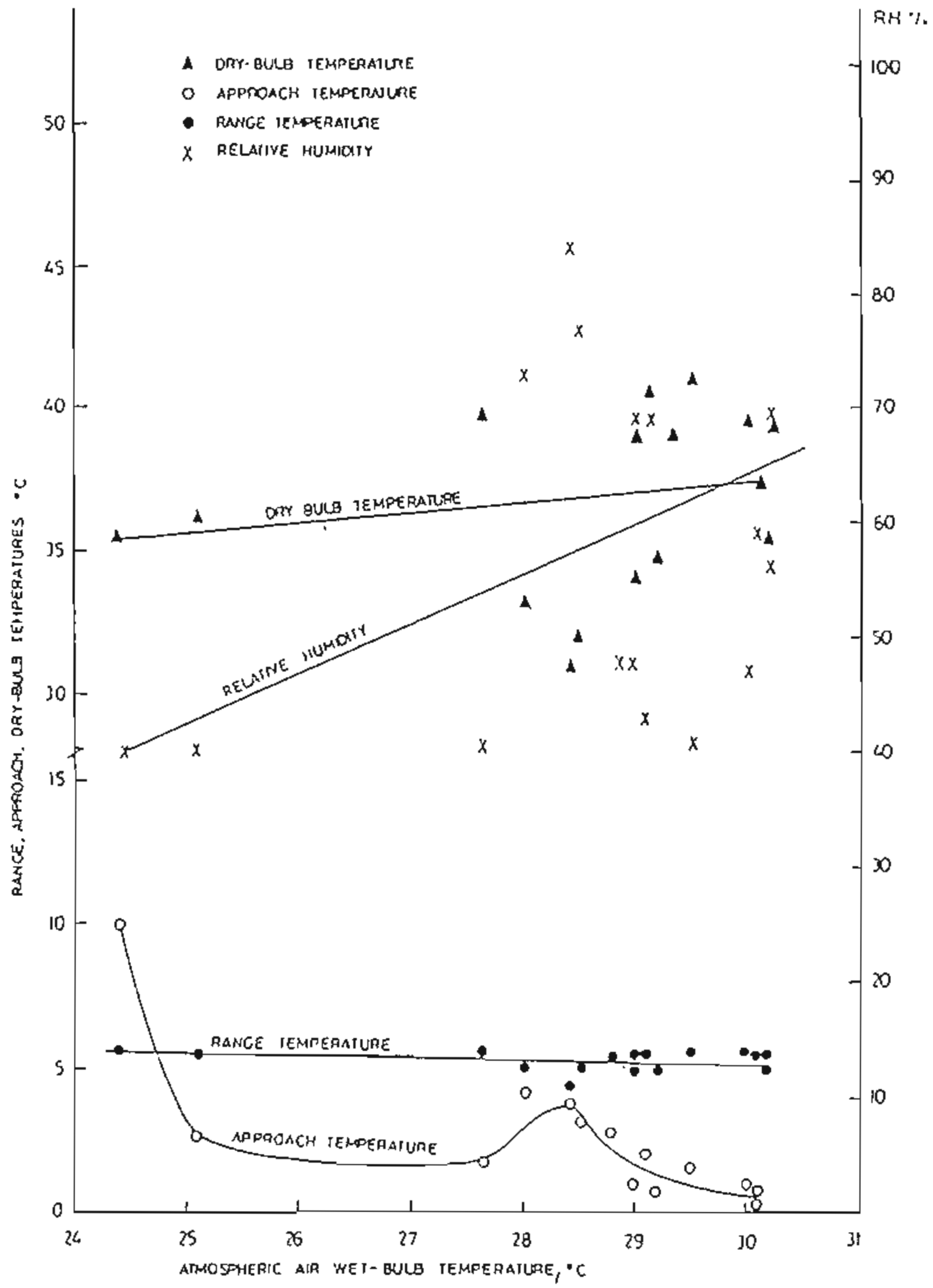
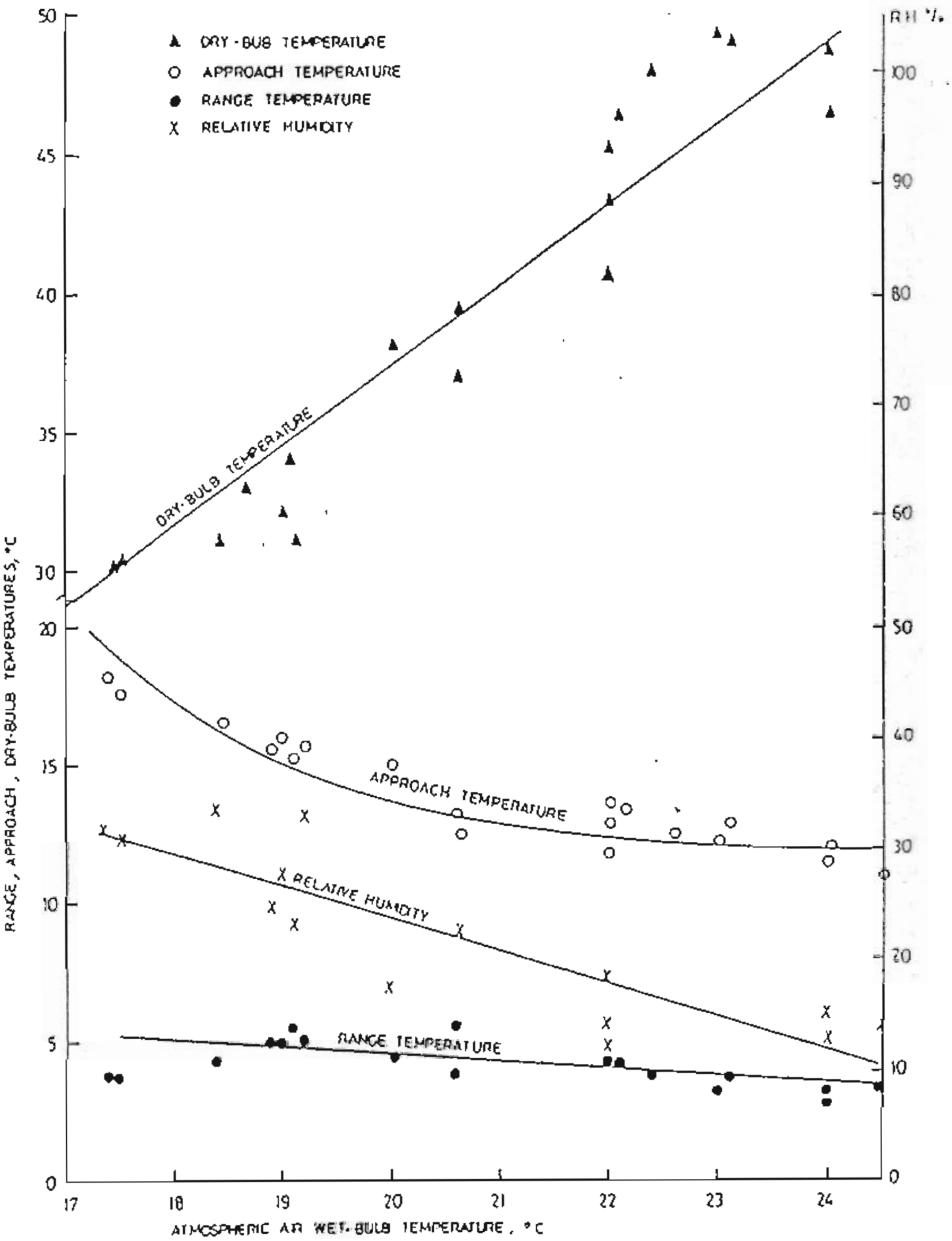



FIG-5, TEMPERATURE RELATIONSHIP BETWEEN WATER AND AIR IN A COUNTERFLOW COOLING TOWER INSTALLED IN KUWAIT CITY ON 31st AUGUST 1980




FIG·6· TEMPERATURE RELATIONSHIP BETWEEN WATER AND AIR IN A COUNTER FLOW COOLING TOWER INSTALLED IN KUWAIT CITY ON 31st JULY 1988

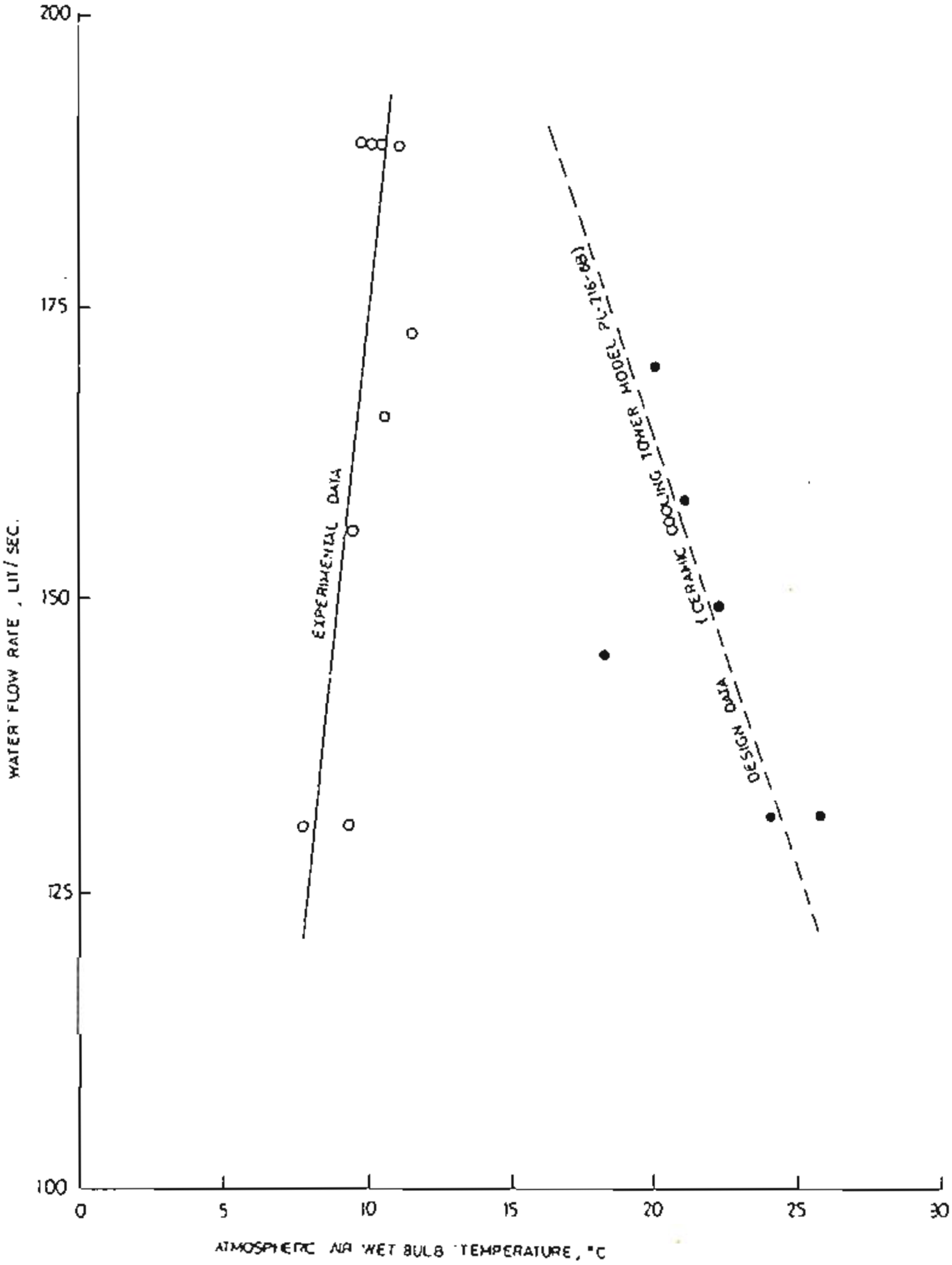



FIG-7 WATER FLOW RATE OF CERAMIC COOLING TOWER INSTALLED IN KUWAIT CITY ON 23rd NOVEMBER 1988

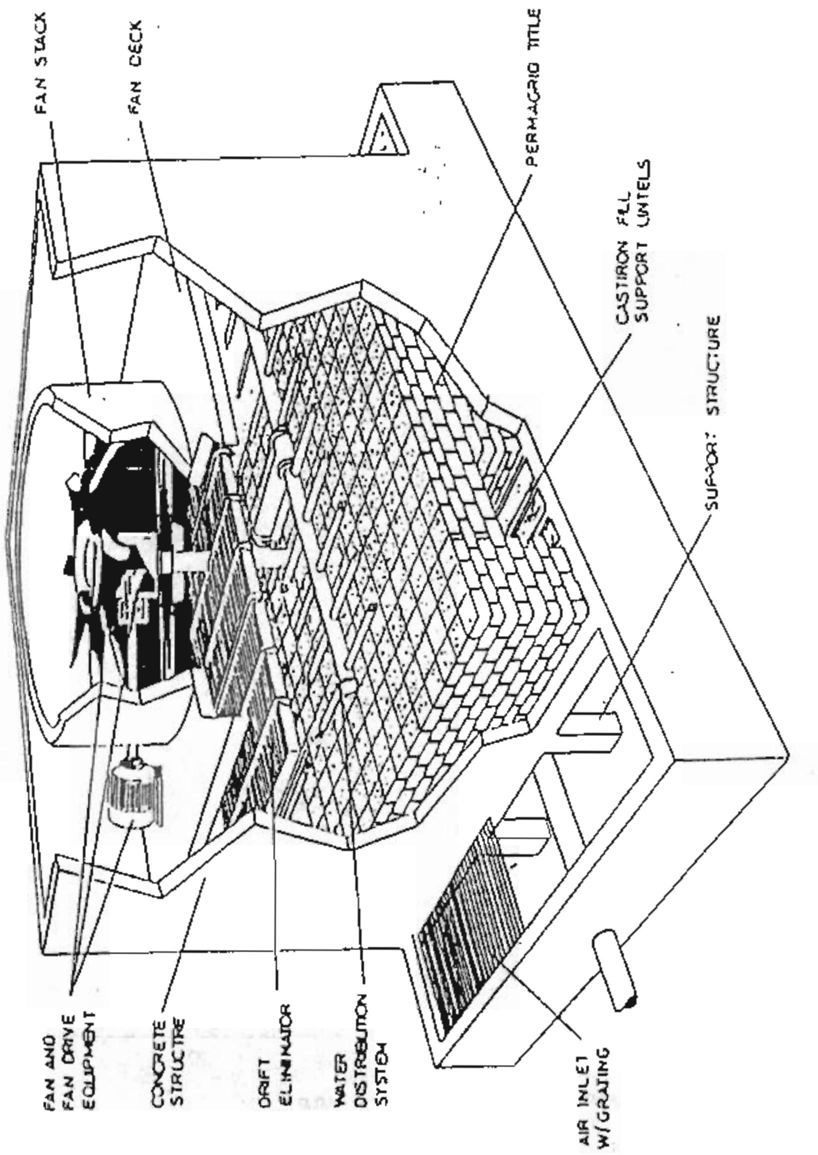



FIG-3 A CERAMIC COOLING TOWER