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MASSACHUSETTS INSTITUTE OF TECHNOLOGY

THE ENERGY LABORATORY

ADVANCED WET-DRY COOLING TOWER CONCEPT
PERFORMANCE PREDICTION

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Energy Laboratory

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Energy Laboratory

In Association With

Heat Transfer Laboratory
Department of Mechanical Engineering

Massachusetts Institute of Technology

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January 31, 1977

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ABSTRACT

The purpose of this years' work has been to test and analyze the new dry cooling tower surface previously developed. The model heat transfer test apparatus built last year has been instrumented for temperature, humidity and flow measurement and performance has been measured under a variety of operating conditions.

Tower Tests showed approximately 40-50% of the total energy transfer as taking place due to evaporation. This can be compared to approximately 80 to 85% for a conventional wet cooling tower. Comparison of the model tower test results with those of a computer simulation has demonstrated the validity of that simulation and its use as a design tool. Computer predictions have been made for a full-size tower system operating at several locations.

Experience with this counterflow model tower has suggested that several design problems may be avoided by blowing the cooling air horizontally through the packing section. This crossflow concept was built from the previous counterflow apparatus and included the design and fabrication of new packing plates.

Instrumentation and testing of the counterflow model produced data with an average experimental error of 10%. These results were compared to the predictions of a computer model written for the crossflow configuration. In 14 test runs the predicted total heat transfer differed from the measured total heat transfer by no more than 8% with most runs coming well within 5%. With the computer analogy's validity established, it may now be used to help predict the performance of fullscale wet-dry towers.

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Principle Symbols

| Symbol | Description |
|--------------------|---|
| A | area |
| C | heat capacity |
| cp | heat capacity |
| D_h | hydraulic diameter |
| h | height |
| h | specific enthalpy |
| $h_{dry\ surface}$ | dry surface heat transfer coefficient |
| $h_o\ h_D$ | wet surface mass transfer coefficient |
| h_{fgT} | latent heat to vaporization of water at T |
| h_{fg}^o | latent heat to vaporization of water at T_o |
| H | Total enthalpy |
| ITD | $(T_{lin} - T_{a\ in})$ |
| k | thermal conductivity |
| \dot{M}, \dot{m} | mass flow rate |
| Nu | Nusselt Number |
| P | pressure |
| Pr | Prandtl Number |
| Q | heat transfer rate |
| Q_{design} | design heat transfer rate for performance models of Chapter 4, 20,000 BTU/min/tower |
| Re | Reynolds Number |
| R H | relative humidity |
| t | plate thickness |

| Symbol | Description |
|-----------------------|--|
| T | temperature |
| T_o | reference temperature |
| V | velocity |
| Z | length of fin |
| W.C. | water consumption |
| W.C. _{wet} | water consumption of wet tower |
| W.C. _{basis} | fixed rate of water consumption used for comparison 0.82 lb _m /1000 BTU rejected |
| W | specific humidity of air-water vapor mixture |

Greek

| | |
|----------|----------------|
| η_f | fin efficiency |
| ρ | density |

Subscripts

| | |
|-------|---------------------------|
| a | air |
| in | inlet |
| l | liquid (water) |
| mix | mixture (air-water vapor) |
| out | outlet |
| v | water vapor |

CHAPTER I

INTRODUCTION

1.1 History of Project

Early in 1975, work was started at M.I.T. to develop a single heat transfer surface which would give low evaporation rates at costs competitive with equal wet tower capacity. The wet-dry concept is the result of this work. In this design, water is distributed onto a packing plate fabricated (see Figure 1-1) to keep the hot water in discrete channels over which cooling air is blown. These channels serve to restrict the free surface area of the water, thus reducing evaporation and water loss. The packing plates are made of conductive material which acts as a fin heated at the base by the channeled water and cooled by forced convection [1]. By conceding a small amount of evaporation the fabrication costs for the wet-dry surface have been reduced far below those of a completely closed or dry heat exchanger.

To demonstrate this concept a model test tower was built during 1975. In addition, a computer program was written based on analytic studies of evaporative cooling. Tests of the partially instrumented tower indicated high heat transfer rates while the initial results of the computer analogy indicated a substantial reduction in evaporation rate. This year's work has led to more quantitative results and predictions.

1.2 Progress This Year

Work this year has included the instrumentation and testing of the model cooling tower referred to above. Comparison (Chapter 3) of these test results with the computer program predictions has shown the program to accurately predict the heat transfer performance of the V-trough packing section under the available range of operating conditions.

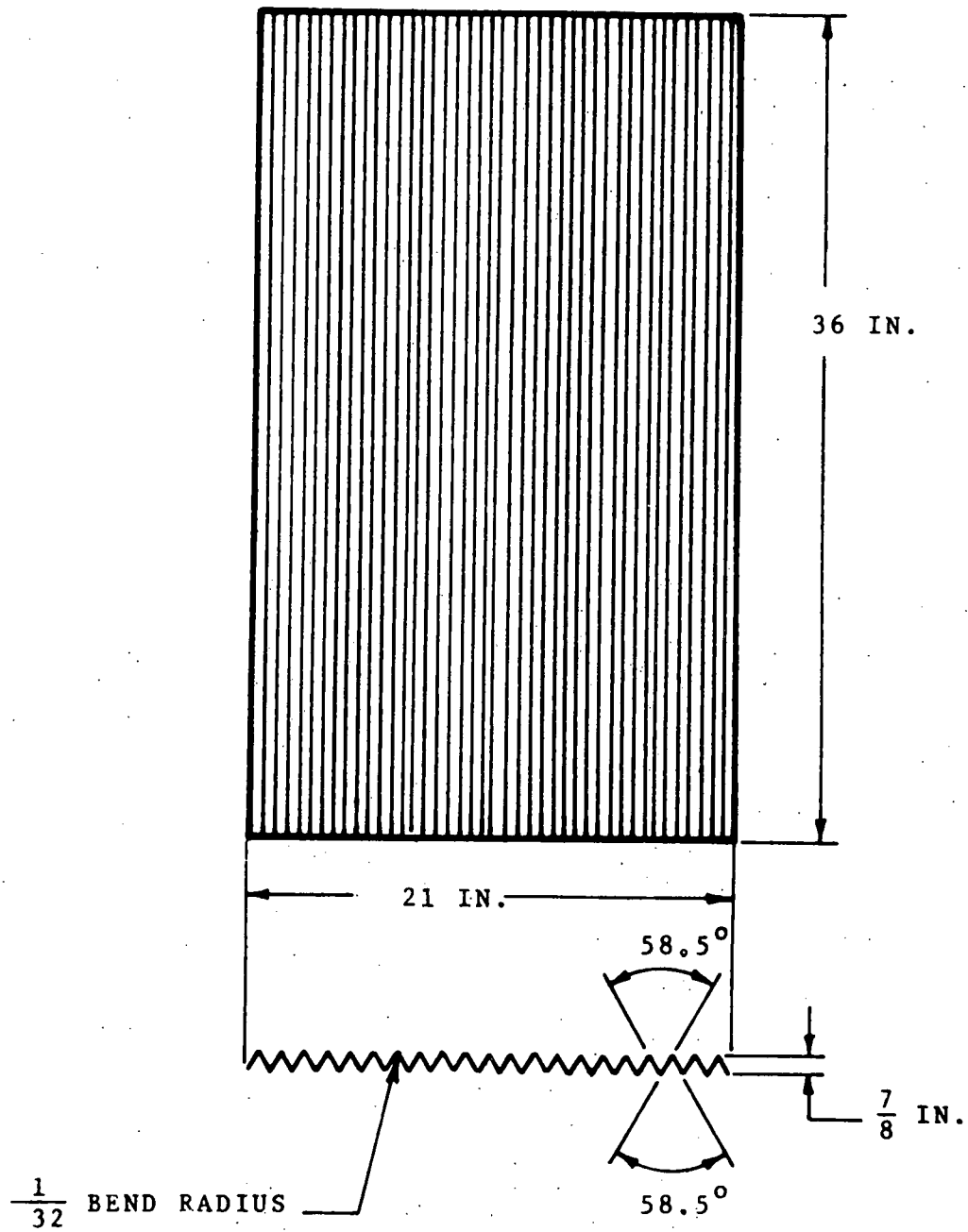


FIGURE 1-1 V-TROUGH PACKING PLATE

Using the computer analogy as a design tool several types of cooling tower systems were sized and compared under a wide range of inlet conditions. These comparisons are revealed in Chapter 4 and include comparisons for several sites around the United States.

During the course of the experimental evaluation several design drawbacks were found to the present air-water counterflow design. A proposed solution to these shortcomings was to alter the air flow to blow horizontally across the packing section. The advantages of this crossflow design are described in Chapter 5 as well as a discussion of the appearance of a full size cross-flow tower.

Chapter 6 contains a simplified estimation of the costs of a full size set-dry towers system based on cost estimation procedures described in the WASH-1360 report [16]. Volume 2 of the report describes the adaptation of the counterflow model tower to a crossflow configuration. This new design required a new type of packing plate as well as complete re-instrumentation of the former model tower.

The final project conclusions, along with recommendations for future work can also be found in Volume 2. Technical and special interest subjects are included as appendices at the end of each part.

CHAPTER 2

EXPERIMENTAL APPARATUS AND DESCRIPTION OF INSTRUMENTATION MODEL2.1 Model Tower

The model tower tested for this report is essentially the same as described in Reference [1], and instrumented as described in the following sections.

As a brief review, the model tower (Figure 2-1) is an induced-draft counterflow design with a plan area of four square feet (0.37 m^2). The heat transfer packing section holds the packing plates. Hot water is distributed at the top of the packing section and collected in troughs at the bottom. Ambient air from the laboratory enters from the bottom of the packing section and exits at the top.

There are 14 V-trough packing plates each (Figure 2-2) with 21 troughs making up a total heat transfer area of 280 ft^2 (26 m^2) [1]. Air flow was fixed by fan size and was approximately $350 \text{ ft}^3/\text{min}/\text{plan foot square area}$ ($1.8 \text{ m}^3/\text{s}/\text{plan m}^2 \text{ area}$). Inlet water flow rate could be varied from 2.7 to $6.0 \text{ gpm}/\text{plan ft}^2$ ($1.8 \text{ to } 4.1 \text{ l/s}/\text{plan m}^2$) with temperatures as high as 150 F (66 C). [1].

Instrumentation of the model included installation of 55 thermocouples measuring air and water temperature at the inlet and exhaust of the packing section. Calibrated rotometers measured water flow rate and a pitot tube was used to check airflow rate. Changes in moisture content of the air were measured by an optical dewpoint hygrometer (Section 2.3).

Error analysis (Section 2.5) has predicted a maximum error between the air and water energy balance of about 15%. The highest error observed in the nine test runs was 24% with the average error approximately 13% (See section 3.3).

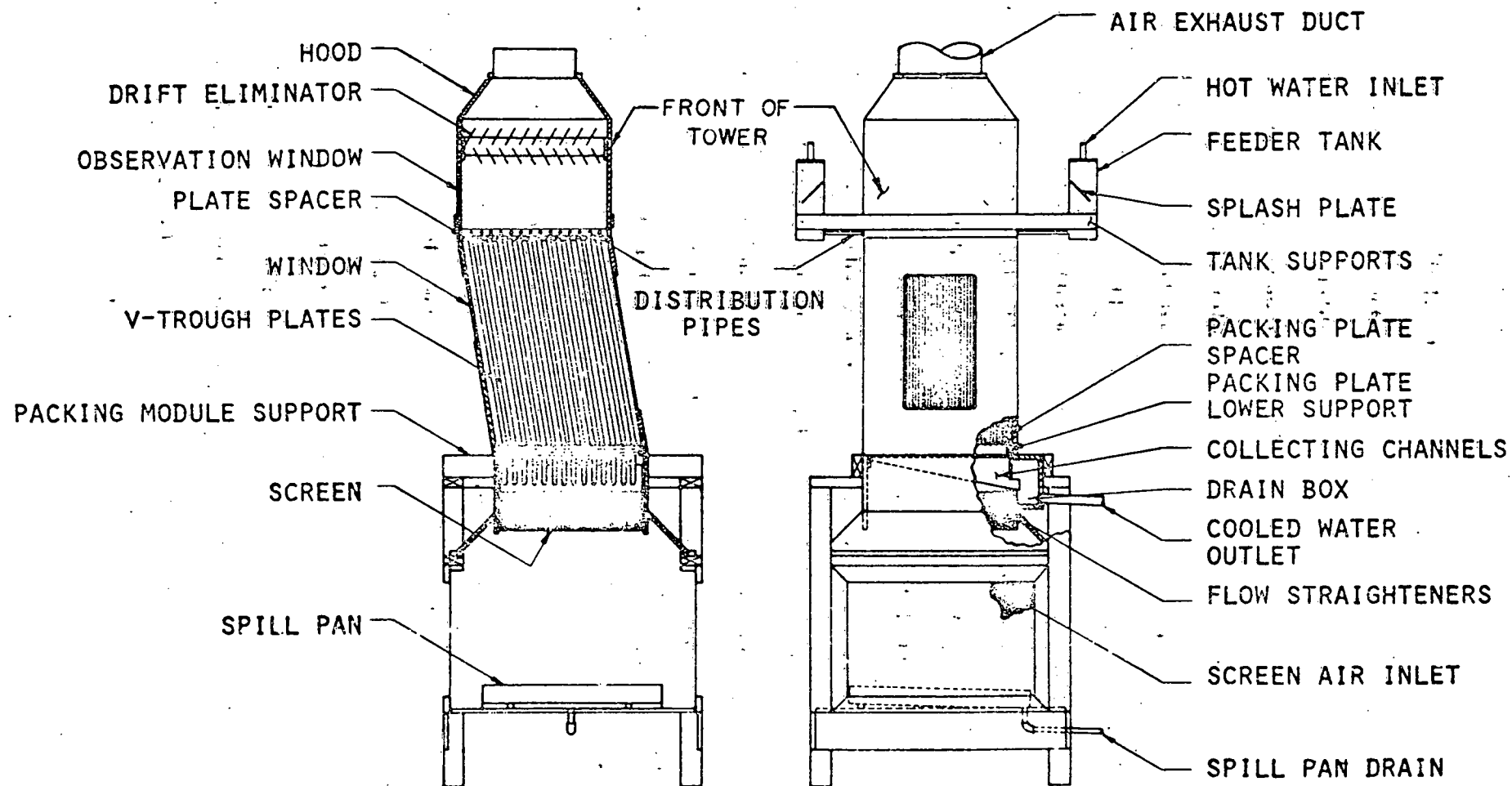


FIGURE 2.1 HEAT TRANSFER TEST APPARATUS ASSEMBLY

2.2 Temperature Measurement

The principal goals of the tower instrumentation were to fully describe the performance of the tower and to check the validity of the computer program. To accomplish this it was necessary to determine the inlet and outlet conditions of both the air and water. The specific measurements required included the temperature change experienced by the water and air streams, the amount of water transferred to the air and the air and water flow rates. It was decided that an accuracy of at least $\pm 10\%$ be required for the temperature differences measured, and the water evaporation rate. This would result in possible errors in the energy balance of less than 10%.

Copper constantan thermocouples were chosen for the temperature measurement. Preliminary computer runs predicted a water temperature drop of less than 4°F for some typical laboratory operating conditions. In order to maintain the accuracy required for this change, the two water temperature measurements would have to be repeatable to about $.2^{\circ}\text{F}$. However, thermocouple wire manufacturers do not guarantee this accuracy for all lengths of wire. Therefore, before installation a calibration check was made of the fifty-four thermocouples after assembly with switches and other hardware (See Appendix A). A single ice bath junction was used between the switches and the readout device. The calibration method was to use one thermocouple as a standard and to compare each of the others to it, when placed in the same constant temperature bath. A steam bath was first used in an attempt to maintain a constant temperature. However, it was found that a calibration to better than $.5^{\circ}\text{F}$ was difficult using this method. A stirred silicon oil bath was then obtained which was

thermostatically controlled. The two thermocouples being tested were placed in a narrow glass tube with oil covering the junctions. The temperature of the bath inside the tube was held constant to $\pm .1^{\circ}\text{F}$ at about 176°F . However, these fluctuations were slow enough so that the output of the thermocouples could be measured and compared to $\pm .05^{\circ}\text{F}$. It was found that while the thermocouples produce a steady voltage for a short time, their calibration will change by about $\pm .1^{\circ}\text{F}$ over a period of several hours. In addition, the thermocouples were in general with $\pm .1^{\circ}\text{F}$ of each other. Therefore, the thermocouples were found to have a repeatability of $\pm .1^{\circ}\text{F}$ not only with time, but also with respect to each other. Accuracy of the system was of lesser importance than repeatability, but comparison with an NBS calibrated thermometer readable to $\pm .5^{\circ}\text{F}$ showed that the thermocouple millivolt outputs could be converted to temperatures by use of standard conversion tables. Also, testing at a lower temperature (104°F) indicated that the error tends to decrease as the temperature decreases.

During both calibration tests and tower runs, a digital voltmeter was used to measure the thermocouple output. This meter had a resolution of ± 1 microvolt which corresponds to about $\pm .05^{\circ}\text{F}$. An accurate potentiometer was used to check the calibration of the voltmeter. Since the meter had high input impedance ($10\text{ M}\Omega$) compared to the wire resistance (50Ω) the effect of the small current flow on the thermocouple voltage is much smaller than 1 microvolt sensitivity of the meter. Potentiometers were not used for the runs, because a device of the accuracy required is often bulky and sensitive to vibration. The digital meter also made it possible to read all fifty-four thermocouples in a shorter time.

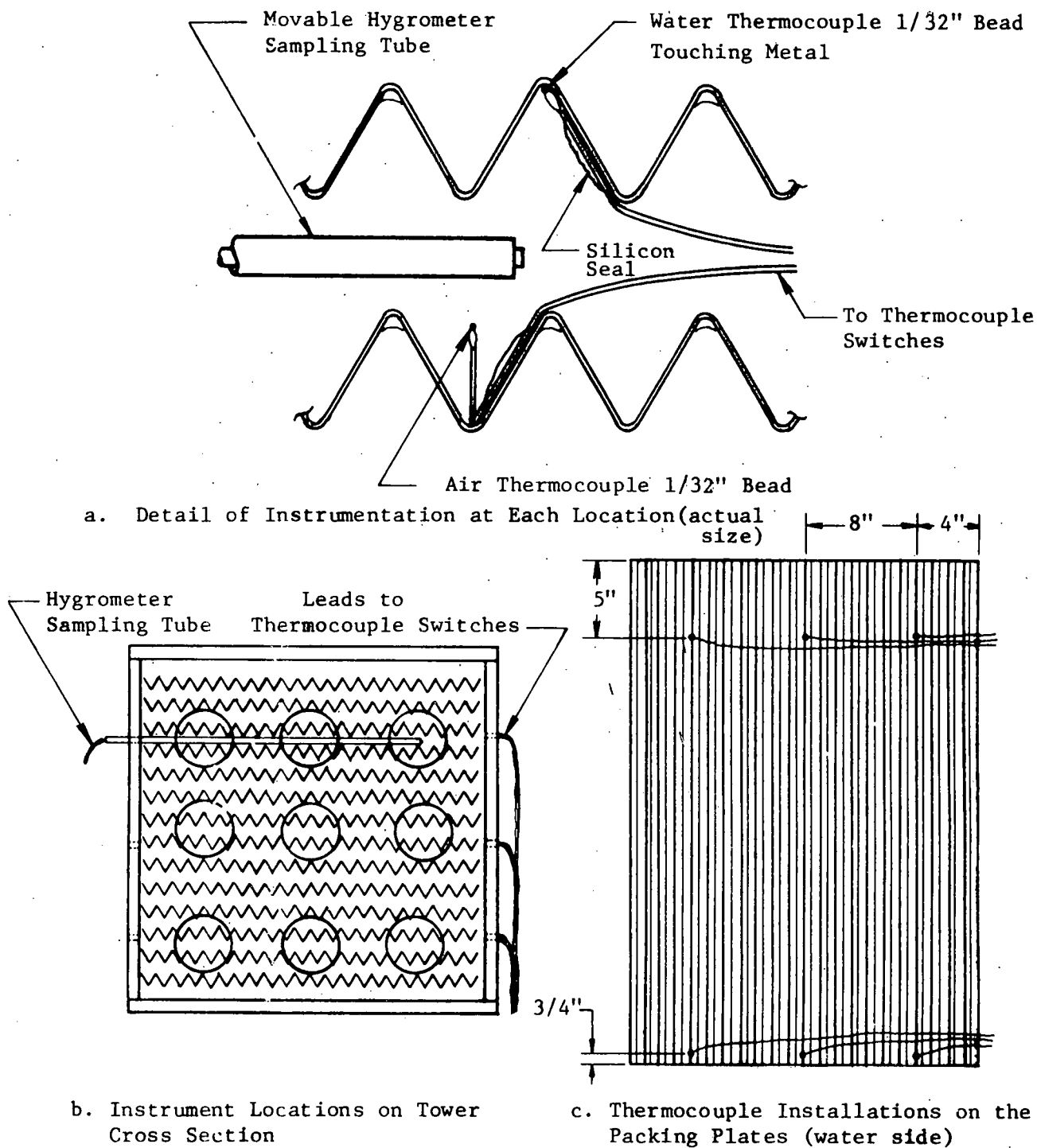


Figure 2.2 Instrumentation of the Packing Plates Including Locations of Thermocouples and Hygrometer Sampling Tube

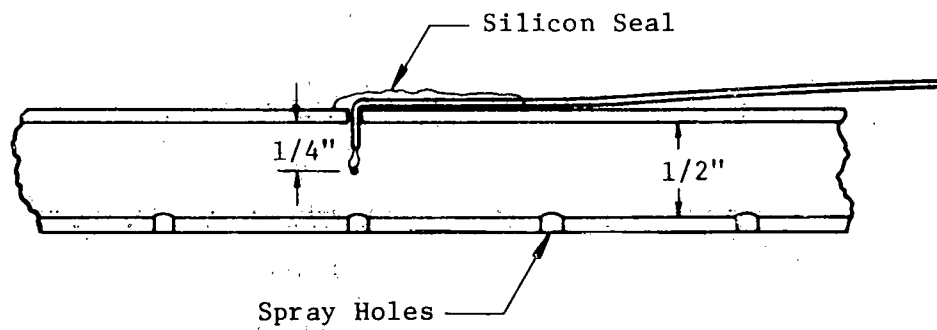


Figure 2-3 Thermocouple Installation in Distribution Pipes (Actual Size)

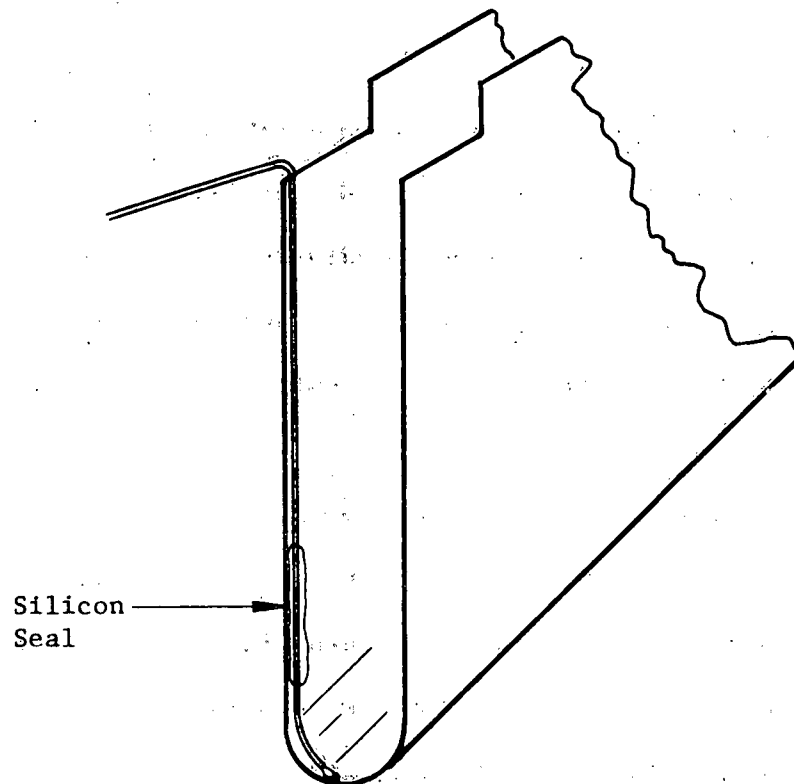


Figure 2-4 Thermocouple Installation in Outlet of Collection Channels (Approximately Actual Size)

Temperatures were measured at eight locations. The water temperature was measured inside the distribution pipes, at the top and bottom of the packing plates and at the outlet of the collection channels. The air temperature was measured below the collection channels, at the bottom and top of the packing and above the distribution pipes. To obtain an average over the cross-section of the tower, nine measurements were made at each location, except in the collection channels, where three measurements were made. The air and water temperature on the plates were measured by using nine separate thermocouples for each measurement. They were fastened at three locations at the top and bottom of three different plates. (See Figure 2-2). The temperature in the water collection channels was measured at the outlet of the channels below the plates instrumented for air and water measurements (See Figure 2-4). The thermocouples in the pipes were placed in the same nine locations as in the packing section. (Figure 2-3). The measurement of the air temperature above the pipes and below the channels was accomplished using rakes with three thermocouples (Figure 2-5). These rakes were moved to three positions as at the other tower locations. The air and water measuring thermocouples on the plates and in the troughs, were fastened in place with silicon seal. In the distribution pipes, the thermocouples were inserted through holes in the pipes and similarly cemented in place.

2.3 Evaporation Measurement

In order to determine the rate of water loss in the tower, two methods were considered. These were a direct measurement of the water in the system before and after a run, and a measurement of the air

humidity at both ends of the packing section. The first method was not used since it was decided that leakage in the tower would greatly affect the measurement of the small amount of loss expected. Several different types of hygrometers were investigated. It was determined that $\pm .3^{\circ}\text{F}$ accuracy was required for a dew point measurement, to ensure less than $\pm 10\%$ error in the evaporation rate, assuming that approximately .1% of the water is evaporated. Hygrometers which use variable resistance probes are in general greatly affected by wetting. They are also subject to some drift and are marginally acceptable when calibrated. An optical dew point device was found to be sufficiently accurate, as well as reliable. A demonstrator model was used for the test runs, which was found to be repeatable to at least $\pm .3^{\circ}\text{F}$ (See Figure 2-2).

2.4 Flow Measurements

Two rotometers were used to determine the water flow rate through the tower, each measuring the flow to one of the plexiglass feeder tanks. Each rotometer had a capacity of approximately 20 gpm and were calibrated by means of a weight tank. They were readable to $\pm .2$ gpm ($\pm 1\%$ maximum flow). For the lowest flow rate used in the tower (5 gpm per meter) this error was $\pm 4\%$.

To speed the process by which the air flow rate was determined in the packing section, it was decided to fix a pitot tube in place between the plates and take only one measurement for each data run.

The air flow between the individual plates was assumed to be turbulent with maximum velocity at the midpoint of the separating gap. Measurements of this midpoint velocity showed that it was nearly uniform throughout the packing section.

The pitot tube was fixed between the plates about 1/3 of the way back and 1/3 of the way (Figure 2-6) in from the sides of the packing section. Great care was taken to align the tube parallel to the plates and in the center of the gap. A movable pitot tube was positioned parallel in the plenum above the packing section and used to scan the airflow at that point.

Data taken at various air flow rates were then compared and a ratio was found between bulk air flow as measured by the scanning pitot tube and mid-point air velocity measured by the fixed pitot tube.

For a pitot tube in a low velocity airstream the velocity is given by the relation:

$$\frac{1}{2} \rho V^2 = \Delta P \quad (2-1)$$

Thus, for any two pitot tubes in air at the same temperature and pressure, the ratio of velocities is:

$$\frac{V_1}{V_2} = \frac{\sqrt{\Delta P_1}}{\sqrt{\Delta P_2}} = \frac{\sqrt{h_1}}{\sqrt{h_2}} \quad (2-2)$$

and the ratio of the average velocity measured in the plenum and the velocity measured by the fixed pitot tube is given by:

$$\frac{V_{avg}}{V_{fixed}} = \frac{(\sqrt{h_1} + \sqrt{h_2} \dots \sqrt{h_9})/9}{\sqrt{h_{fixed}}} \quad (2-3)$$

Where $h_1 - h_9$ are the manometer heights taken at the 9 scan locations.

Eight sets of measurements were made at varying airflow and averaged to give the velocity ratio. (See Table 2-1).

For the analytic comparison,

$$Re = V D_h / \nu \quad V = 300 \text{ to } 500 \text{ ft/min (1.52 to 2.54 m/s)}$$

$$D_h = 0.25 \text{ ft (.08 m)}$$

$$Re = 6400 \text{ to } 11,000$$

From Reference [12] by integrating the relation:

$$\frac{V_x}{V_{CL}} = (y/r_o)^{1/n} \quad \text{where}$$

| Re | n | V/V_{CL} |
|---------|---|------------|
| 4,000 | 6 | 0.791 |
| 110,000 | 7 | 0.817 |

Indicating a ratio of approximately 0.80 for this range of Reynold's numbers. This compares favorably with the experimentally measured ratio of 0.81.

2.5 Error Analysis

To determine the significance of the instrument errors, their effect on the energy balance must be shown. The energy balance for a heat exchanger reduces to [4]:

$$\text{Rate of enthalpy in} = \text{Rate of enthalpy out}$$

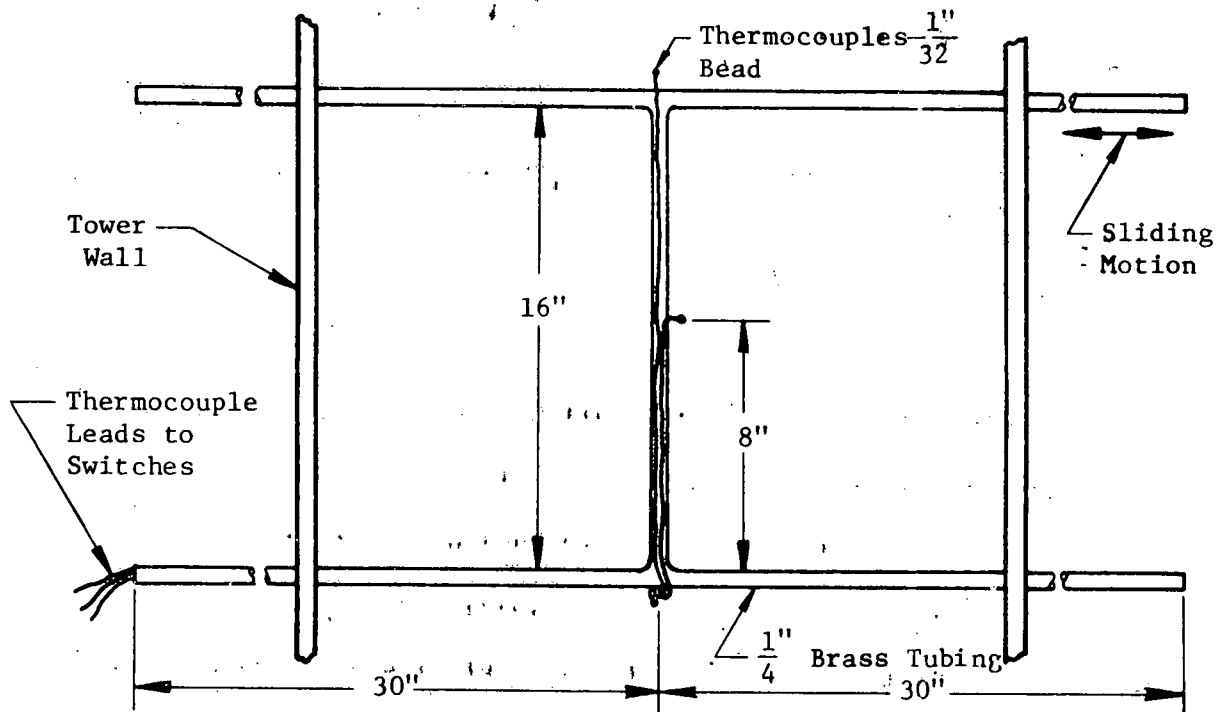
In the case of the cooling tower, this becomes,

TABLE 2-1

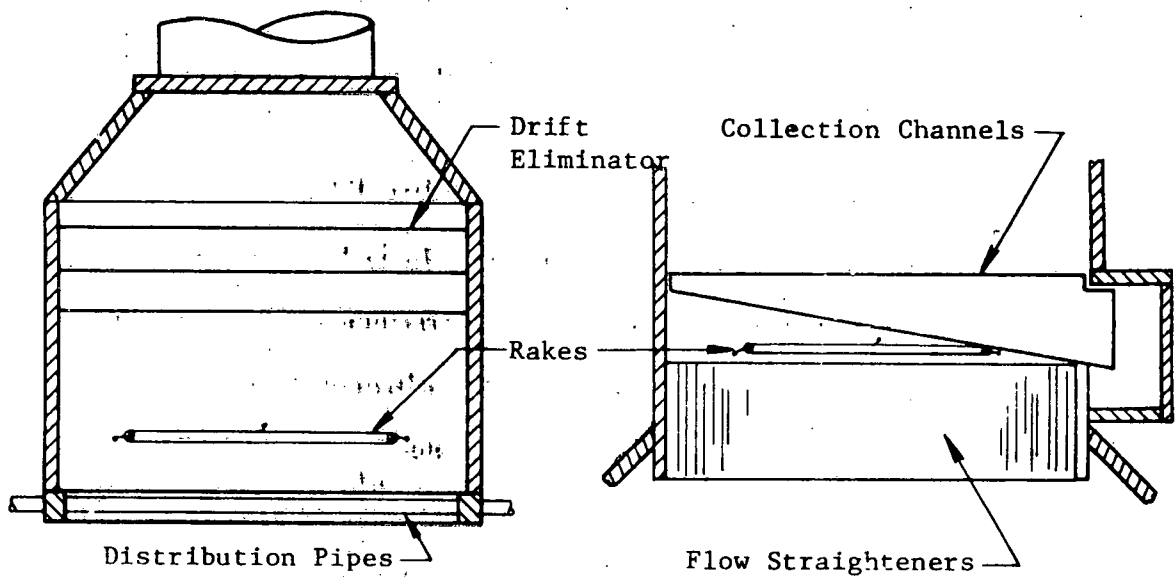
Average and Centerline Air Velocities Between Packing Plates

| RUN | $\sqrt{h_{\text{above avg plates}}}$ | $\sqrt{h_{\text{fixed}}}$ | Ratio |
|-----|--------------------------------------|---------------------------|--------------------------------|
| | ($\sqrt{\text{in}}$) | ($\sqrt{\text{in}}$) | $V_{\text{avg}}/V_{\text{CL}}$ |
| 1 | 0.1213 | 0.1367 | 9.89 |
| 2 | 0.0779 | 0.1095 | 0.71 |
| 3 | 0.1232 | 0.1500 | 0.82 |
| 4 | 0.1406 | 0.1590 | 0.88 |
| 5 | 0.1397 | 0.1711 | 0.82 |
| 6 | 0.1176 | 0.1539 | 0.76 |
| 7 | 0.1186 | 0.1410 | 0.84 |
| 8 | 0.0864 | 0.1090 | 0.79 |

Average Ratio = 0.81



a. Top View of Rake



b. End View Above Distribution Pipes

c. End View Below Collection Channels

Figure 2-5: Location of Sliding Rakes Used to Measure Air Temperature Above the Distribution Pipes and Below the Collection Channels

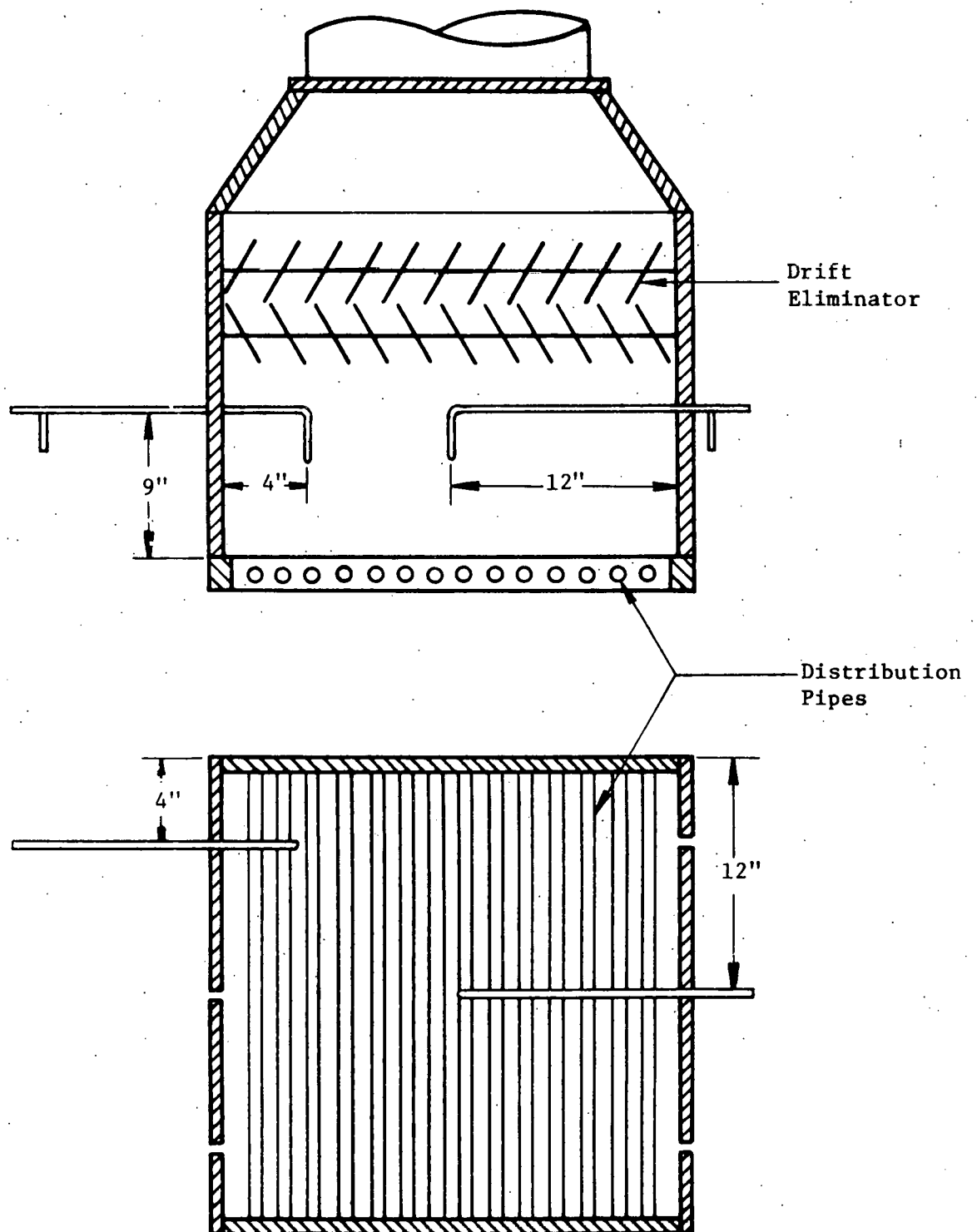


Figure 2-6: Pitot Tube Installation Showing Two Locations of the Pitot Tube

$$\begin{aligned} \dot{Q} = & \dot{m}_{\text{air}} (h_{\text{air}_{\text{in}}} + w_{\text{in}} h_{\text{vapor}_{\text{in}}} - h_{\text{air}_{\text{out}}} - w_{\text{out}} h_{\text{vapor}_{\text{out}}}) = \\ & - \dot{m}_{\text{water}} c_{p_{\text{water}}} (T_{\text{water}_{\text{in}}} - T_o) + (\dot{m}_{\text{water}} - \Delta \dot{m}_{\text{water}}) c_{p_{\text{water}}} \\ & (T_{\text{water}_{\text{out}}} - T_o) \end{aligned} \quad (2-4)$$

where \dot{m} , h , w , and T are mass flow rates, enthalpies, absolute humidities and temperature respectively. \dot{Q} is the heat transfer rate and $c_{p_{\text{water}}}$ is the specific heat at constant pressure for water, which is assumed to be 1 BTU/lb_m °F. T_o is the reference temperature at which the enthalpies are evaluated. Here T_o is 32°F. The reference for the water vapor is liquid water at 32°F. $\Delta \dot{m}_{\text{water}}$ is the amount of water transferred to the air stream, and is given by:

$$\Delta \dot{m}_{\text{water}} = \dot{m}_{\text{air}} (w_{\text{out}} - w_{\text{in}}) \quad (2-5)$$

The value of \dot{m}_{air} is determined by the pitot tube measurement, and is subject to the associated error. h_{air} and $h_{\text{air}_{\text{out}}}$ are determined by the air temperature measurements as are h_{vapor} and $h_{\text{vapor}_{\text{out}}}$. Errors in these values are determined by inaccuracies in the thermocouple reading, as are errors in $T_{\text{water}_{\text{in}}}$ and $T_{\text{water}_{\text{out}}}$. w_{in} and w_{out} are measured with the dew point hygrometer.

To evaluate the effect of instrument limitations, an uncertainty analysis is made using the following form of equation 2-4, combined with equation 2-5.

$$\begin{aligned}
\delta \dot{Q} = & \dot{m}_{\text{air}} (h_{\text{air}_{\text{in}}} + w_{\text{in}} h_{\text{vapor}_{\text{in}}} - h_{\text{air}_{\text{out}}} - w_{\text{out}} h_{\text{vapor}_{\text{out}}}) \\
& + \dot{m}_{\text{water}} c_{p_{\text{water}}} (T_{\text{water}_{\text{in}}} - T_o) - c_{p_{\text{water}}} \\
& (\dot{m}_{\text{water}} - \dot{m}_{\text{air}} (w_{\text{out}} - w_{\text{in}})) (T_{\text{water}_{\text{out}}} - T_o)
\end{aligned} \quad (2-6)$$

where $\delta \dot{Q}$ is the discrepancy between the air and water sides of heat balance. The uncertainty $u_{\delta \dot{Q}}$ is calculated by the following equation [9]:

$$u_{\delta \dot{Q}} = \left[\left(\frac{\partial \delta \dot{Q}}{\partial x_1} u_1 \right)^2 + \left(\frac{\partial \delta \dot{Q}}{\partial x_2} u_2 \right)^2 + \dots + \left(\frac{\partial \delta \dot{Q}}{\partial x_n} u_n \right)^2 \right]$$

where x_1, x_2, \dots, x_n are the humidities, enthalpies, temperatures and flow rates in equation 2-6 and u_1, u_2, \dots, u_n are their respective uncertainties. The following values which satisfy the energy balance might be typical test conditions for the model tower:

$$\dot{m}_{\text{air}} = 150 \pm 8 \text{ lb}_m/\text{min}$$

$$\dot{m}_{\text{water}} = 170 \pm 1.7 \text{ lb}_m/\text{min}$$

$$\text{Air temperature in} = 90^\circ\text{F} \pm .1^\circ\text{F (after collection troughs)}$$

$$\text{Dew point temperature in} = 50^\circ\text{F} \pm .3^\circ\text{F}$$

$$\text{Air temperature out} = 100^\circ\text{F} \pm .1^\circ\text{F}$$

$$\text{Dew point temperature out} = 55^\circ\text{F} \pm .3^\circ\text{F}$$

$$T_{\text{water in}} = 120^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

$$T_{\text{water out}} = 116.4^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

The heat transfer rate for this case as calculated using equation 2-4 is 625.6 BTU/min. The uncertainty according to equation 2-7 is 46.6 BTU/min or 7.5%. To illustrate the dependence of the error in \dot{Q} on the test conditions the following set of conditions is investigated:

$$\dot{m}_{\text{air}} = 150 \pm 8 \text{ lb}_m/\text{min}$$

$$\dot{m}_{\text{water}} = 90 \pm 1.7 \text{ lb}_m/\text{min}$$

$$\text{Air temperature in} = 90^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

$$\text{Dew point temperature in} = 20^{\circ}\text{F} \pm .3^{\circ}\text{F}$$

$$\text{Air temperature out} = 103^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

$$\text{Dew point temperature out} = 33^{\circ}\text{F} \pm .3^{\circ}\text{F}$$

$$T_{\text{water in}} = 140^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

$$T_{\text{water out}} = 132^{\circ}\text{F} \pm .1^{\circ}\text{F}$$

In this case $\dot{Q} = 745.43$ and the uncertainty is 43.83. Even though the water temperature change and the absolute humidity may be more accurately measured for this case, the effect on the energy balance is insignificant. The error in the air flow measurement still predominates, and the uncertainty is essentially unchanged (see Appendix B). Due to an increase in \dot{Q} , however, the percent error is reduced to 5.9%.

Other factors which may influence the accuracy of the data, include the fact that the nine locations may not give a true average over the

cross-section. The hygrometer sampling tube may occasionally remove water droplets with the air, and the pitot tube measurements may be affected by the proximity of other tower components such as the distribution pipes and the drift eliminators. (See Fig. 2-6). The magnitude of these possible errors is difficult to predict analytically.

CHAPTER 3

COMPARISON OF TEST RESULTS AND COMPUTER PROGRAM3.1 Computer Program

The computer program (Appendix H) was the same as given in reference [1], modified to match measured values of the dry plate surface heat transfer coefficient and the wet-to-dry surface area ratio.

The basic equations were taken from a paper by G. Yadigaroglu which was concerned with totally wet towers with flat packing plates. They were then modified to include heat transfer from the dry surface. The solution involves choosing values for the temperatures, water flow rates, absolute humidity and heat transfer rates and solving for the incremental changes. These changes are then added to the initial values and the solution found in a "marching out" or Euler process.

The expressions for the water surface and dry plate heat transfer coefficients h and h_{DP} are taken from the Dittus-Boelter relation as used by Yadigaroglu [2].

$$Nu = .022 Pr^{0.6} Re^{0.8} \quad (3-1)$$

For calculating fin efficiency, the packing plates were modeled as simple plate fins (shown in Fig. 2-3). Fin efficiency was then calculated from the expression:

$$\eta_{fin} = \frac{\tanh \sqrt{Z^2 h/kt}}{\sqrt{Z^2 h/kt}} \quad (3-2)$$

taken from reference [12].

The mass transfer coefficient is derived from the Chilton-Colburn analogy between heat and mass transfer may be written: [1]

$$h_{D_{\text{wet surface}}} = \frac{h_{\text{wet surface}}}{\rho_{\text{mix}} C_{\text{mix}}} (Pr/Sc)^{2/3} \quad (3-3)$$

where ρ_{mix} , C_{mix} and Sc are the density, specific heat and Schmidt number of the air-water vapor mixture. The characteristic length used for these relations is the hydraulic diameter of the air flow channel between the packing plates [12].

Provision was also made to allow the program to run for a completely wet and completely dry surface area. Comparison with previously published results provided a check on the program's validity [1].

Physical properties of air, liquid water and water vapor were approximated by correlating equations [4 and 5] and/or simple curve fits [1], and may be seen in the computer listing (Appendix H).

Overall heat transfer is given by the equation:

$$Q_{\text{tot}} = (T_{\ell_{\text{in}}} - T_{\ell_{\text{out}}}) \dot{m}_{\ell_{\text{out}}} + \dot{m}_a (w_{\text{out}} - w_{\text{in}}) (T_{\ell_{\text{in}}} - T_o) \quad (3-4)$$

And evaporative heat transfer given by:

$$Q_{\text{evap}} = (m_{\ell_{\text{in}}} - m_{\ell_{\text{out}}}) [h_{f_{8T_{\ell_{\text{in}}}}} - C_a (T_{\ell_{\text{in}}} - T_{o_{\text{out}}})] \quad (3-5)$$

Where T_o is the temperature at which the enthalpy of the saturated liquid to be zero, in this case $T_o = 32^\circ\text{F}(0^\circ\text{C})$ so as to remain consistent with published psychrometric charts and tables.

3.2 Determination of Surface Transfer Coefficients

Certain parameters of the packing section were unknown functions of geometry and flow conditions. Two of these, the wet-to-dry surface area ratio and the dry plate surface heat transfer coefficient, were determined in auxiliary experiments and these values used to modify the analytic predictions in the computer analogy.

The determination of $A_{\text{wet}}/A_{\text{dry}}$ was done twice, before and after painting the packing plates. (See Appendix E).

Before painting the deposit buildup was easily visible on the surface and its width could be directly measured with a scale. This was assumed to be the entire extent of the wet surface area due to the "blotter effect" noted in Appendix E. Using the observed deposit width of 0.25 in (0.6 cm) from the trough bottom and the water free surface width as 0.125 in. (0.3 cm) gives a calculated $A_{\text{wet}}/A_{\text{dry}}$ of 11%.

After painting the plates a photographic method was necessary as there was no longer a measureable deposit line. The paint used was non-reflecting and black. A columnated light source was shone down into the trough from the side. Any reflection seen would have to come from the water surface, since the water did not wet the plate and could not "climb" up the side of the trough. The reflection was recorded by a high quality single-lens reflex camera using close-up lenses for magnification. Enlargements of the photographs showed a clear separation between water surface and plate area. These photographs, after correction for depth-of-field, show a water surface area width of 0.134 in (0.34 cm). This corresponds to a wet-to-dry surface area ratio of 4%. This was later increased to 5.5% as it became evident that the same non-wetting characteristics that helped cut the

total wetted surface area were now letting the water streams wander on the upper portion of the packing plates.

This area ratio did not change significantly over the available range of water flow rates.

The dry surface heat transfer coefficient of the packing plates was checked by evaluation of the transient response of the plate temperature to a step change in air temperature. The packing plates were cooled 15 to 20°F (8-11°C) below ambient (intake) air temperature. The apparatus exhaust fan was then started, pulling the warmer ambient air into the packing section at a known rate of flow. Local plate and air temperatures were recorded at 10 second intervals until plate temperature approached intake air temperature.

Analysis (Ref. [8], Chapter 3) of this data (See Appendix J) indicated a dry plate surface heat transfer coefficient equal to $3.3 \text{ BTU/hr-ft}^2\text{°F}$ ($160/\text{kcal/h-m}^2\text{°C}$). This value was approximately 1.5 times higher than the value predicted by equation 3-1 when corrected for entrance effects [16]. This increase was attributed to the highly irregular flow channel and the high inlet turbulence from the flow straighteners and collection channels (Figure 2-1).

Two other parameters, the wet surface heat and mass transfer coefficients could not be experimentally measured but were increased by a factor of 1.5 also. This was done as the wet surface heat and mass transfer coefficients depend on the same flow conditions as the dry heat transfer coefficient (except for the relative Reynolds number between the air and the moving water surface).

3.3 Summary of Test Data and Computer Predictions

The summary charts (Tables 3-1, 3-2) list the major points of comparison between the computer prediction and the model tower test results. For the model tower tests both air side and water side enthalpy change was calculated. On the air side:

$$\Delta H_a = \dot{m}_a [w_{out} (h_{fg}^o + \int_{T_o}^{T_{a,out}} C_v dT) - w_{in} (h_{fg}^o + \int_{T_o}^{T_{air,in}} C_v dT) + \int_{T_{a,in}}^{T_{a,out}} C_a dT] \quad (3.6)$$

Where T_o is the reference temperature where the enthalpy of saturated liquid is taken as zero, and where h_{fg}^o is the latent heat of vaporization at the reference temperature. In practice the air side enthalpy change was evaluated from tabulated values [6] of dry air and water vapor enthalpy. These tables were based on $T_o = 32^\circ\text{F}$ (0°C) and for consistency this value of T_o will be used in all calculations.

On the water side:

$$\Delta H = \dot{m}_\ell \int_{T_{\ell,out}}^{T_{\ell,in}} C_\ell dT - \dot{m}_a (w_{out} - w_{in}) \int_{T_o}^{T_{\ell,in}} C_\ell dT \quad (3-7)$$

Where the second term on the righthand side of the equation represents the enthalpy loss due to mass transfer. In practice, the heat capacity of water was taken to be a constant, 1 BTU/lbm- $^\circ\text{F}$ (1 kcal/kg- $^\circ\text{C}$) and ΔH_ℓ evaluated directly from the measured inlet and outlet conditions.

For energy balance calculations, the error was calculated from the equation:

$$\text{Error} = \frac{\Delta H_a + \Delta H_\ell}{1/2 (\Delta H_a - \Delta H_\ell)} \quad (3-8)$$

where $1/2(\Delta H_a - \Delta H_\ell)$ is the average of the air and water side enthalpy changes and appears in Tables (3-1, 3-2) as Q_{total} for the model tower tests.

Q_{evap} was evaluated for the model tower tests from the inlet and exhaust conditions based on the thermodynamic relations:

$$Q_{evap} = \dot{m}_a (w_{out} - w_{in}) [h_{fg}^o + \int_{T_o}^{T_{a_{out}}} C_v dT - \int_{T_o}^{T_{water_{in}}} C_\ell dT] \quad (3-9)$$

Assuming constant C_v and rearranging this can be written as:

$$Q_{evap} = \dot{m}_a (w_{out} - w_{in}) [h_{fg}^{T_{\ell_{in}}} - C_v (T_{\ell_{in}} - T_{a_{out}})] \quad (3-10)$$

Where C_v was taken to be 0.4458 BTU/lbm°F (0.4458 kcal/kg°C) and $h_{fg}^{T_{\ell_{in}}}$ was taken from tabulated values [6].

The values of Q_{evap} and Q_{total} for the computer model were calculated from the exhaust conditions predicted by the program and are presented in the tables for each test run. A specific discussion of each table follows below.

The energy balance ($\Delta H_\ell + \Delta H_a = 0$) provides an indication of the validity of the experimental process. Analysis (see Section 2.5) of this particular experiment showed a maximum possible error of 15% (as defined above) based on individual instrument repeatability. In practice this limit was exceeded on several occasions. These discrepancies have been attributed to changing flow conditions. Data measurements for a typical run required approximately one hour. Water temperatures were observed to remain fairly constant during this time as were, to a lesser extent air temperatures. The greatest problem was encountered in the humidity

data. The optical dewpoint hydrometer, while highly repeatable, was slow reading and as much as 20 to 30 minutes could elapse between beginning to scan the inlet section and finishing the scan of the outlet section. Water sprays, steam jets and other watery experiments in the laboratory had a noticeable effect on the air humidity content. Efforts to reduce this interference included running at odd hours and attempting to control local sources of humidity. Reasonable energy balances were thus obtained with the larger errors blamed on unobserved humidity and air temperature transients.

This summary is divided into two parts, 5 data runs completed before painting the packing plates and 4 runs done since that time (see Appendix E). Program inputs for the first part consist of ambient inlet conditions to the model tower packing section plus these approximate parameters (see Section 3-2):

$$A_{\text{wet}}/A_{\text{dry}} = 11\%$$

$$h_{\text{dry plate}} = 3.3 \text{ BTU/hr-ft}^2\text{ }^{\circ}\text{F} \text{ (16.1 kcal/h-m}^2\text{ }^{\circ}\text{C)}$$

$$h_{\text{wet surface}} = 3.6 \text{ BTU/hr-ft}^2\text{ }^{\circ}\text{F} \text{ (18.0 kcal/h-m}^2\text{ }^{\circ}\text{C)}$$

$$h_{\text{D wet surface}} = 240 \text{ ft/hr (75 m/h)}$$

As can be seen from the summary chart (Table 3-1), only once does the predicted evaporative heat transfer rate differ by more than 10% of the measured rate and never does the total heat transfer rate fall outside the measured air and water side heat transfer rates.

Corrosion noted on the plates after the initial series of data runs eventually necessitated the coating of the packing plates with an acrylic protective paint (App. E). After painting the plates the $A_{\text{wet}}/A_{\text{dry}}$ ratio

was found to be 5.5% (Section 3-2). An order of magnitude analysis shows that the resistance to heat transfer added by painting the surface is very small when compared to the convective heat transfer coefficient. Since the wet surface characteristics have been assumed to remain the same as before the painting, (except for the surface area ratio) all the transfer coefficients have been left as in the previous runs.

Referring to the summary chart, Table 3-2, the four runs produced good energy balances, however the correlation between measured and computer-predicted total heat transfer is not as close as in the previous series. The experimentally measured heat transfer rate is consistently low in each of the four cases, but moves closer to the computer-predicted value in consecutive runs. By the last run (on 6/24) the difference is much less than 10% of the measured value and the predicted value well within the measured air and water side heat transfer rates.

It was speculated that a temporary resistance to heat transfer was caused by incomplete wetting of the newly painted plates. The approach of measured to predicted heat transfer can be a "wearing in" period during which the non-wetting characteristic declined. There was no noticeable change in the wet-to-dry surface area ratio.

As Table 3-2 shows, the evaporative heat transfer predictions match closely with the measured values in each run.

From these results it was concluded that the agreement between the analytic (computer) model and the heat transfer model tower test results were good enough to permit the use of the computer program (listed in Appendix H) to generate data for the comparisons in Chapter 4.

TABLE 3-1

Comparison of Computer Program and Model Tower Test Results -
 Unpainted Packing Plates $A_{wet}/A_{dry} = 11\%$, Galvanized Plates

| Run Date | ITD (F) | Q_{total} BTU/min | Q_{evap} BTU/min | Air Side Heat Transfer (BTU/min) | Water Side Heat Transfer (BTU/min) | $\frac{Q_{tot}}{Q_{evap}}$ (%) | Energy Balance Error (%) |
|---------------------------------|---------|---------------------|--------------------|----------------------------------|------------------------------------|--------------------------------|--------------------------|
| 3/19 measured computer | 43.7 | 839 875 | 405 460 | 738 | 939 | 48 53 | 24 |
| 3/22 measured computer | 36.7 | 677 700 | 336 358 | 638 | 727 | 50 51 | 13 |
| 3/23 am measured computer | 38.1 | 759 765 | 427 402 | 750 | 768 | 56 53 | 2 |
| 3/23 pm measured computer | 43.3 | 935 955 | 490 526 | 835 | 1017 | 52 55 | 18 |
| 3/25 measured computer | 44.5 | 877 900 | 463 481 | 844 | 909 | 55 54 | 7 |

TABLE 3-2

Comparison of Computer Program and Model Tower Test Results -
Painted Packing Plates

$A_{\text{wet}}/A_{\text{dry}} = 5.5\%$, Plates coated 0.001 in Crylon^R paint

| Run Dates | ITD (F) | Q_{total} BTU/min | Q_{evap} BTU/min | Air Side Heat Trans- fer(BTU/min) | Water Side Heat Trans- fer(BTU/min) | $\frac{Q_{\text{total}}}{Q_{\text{evap}}}$ (%) | Energy Balance Error(%) |
|------------------------------|------------|-------------------------------|------------------------------|---|---|---|-------------------------------|
| 5/6 measured computer | 42.3 | 544 654 | 279 278 | 518 | 571 | 51 43 | 10 |
| 5/11 measured computer | 53.7 | 755 366 | 440 371 | 812 | 697 | 58 43 | 15 |
| 5/25 measured computer | 52.8 | 701 771 | 316 305 | 653 | 749 | 45 40 | 14 |
| 6/24 measured computer | 28.0 | 380 400 | 195 171 | 413 | 346 | 51 43 | 17 |

CHAPTER 4

PERFORMANCE COMPARISON FOR VARIOUS COOLING TOWER COMBINATIONS

This chapter compares the performance for a given power plant of the two most widespread types of cooling towers used today, the evaporative (wet) and the non-evaporative (dry), with that of the V-trough packing design developed in this project (wet-dry) (see Section 2-1). To generate the data necessary for these comparisons an idealized design has been made of each tower type (Appendix D). Each of these designs is sized to give the same hot water temperature drop and total heat transfer rate for a set of fixed design inlet conditions. The designs have then been evaluated by computer analogy under a number of differing inlet conditions and the results (evaporation rate, heat transfer rate, exit air conditions) used as a basis for the following comparisons.

Also included are the combination-type towers used by some designers to overcome the disadvantages of single wet and dry systems. Studies of these [13] systems are very promising and the main thrust of this chapter is directed toward the analysis of these systems.

4.1 Description of Combination-Type Systems

A combination tower design consists of two individual cooling towers in parallel or series connection, each bearing a share of the heat load requirement. Towers can be inter-connected in several different ways (Figure 4-1), but for simplicity it was assumed that each tower acted independently and received its share of the hot water to be cooled at the same temperature as in Figure 4-1C. In this analysis, the designs use a dry tower sized so that it alone can handle the required heat load

Fig. 4-1 A

Flow Configurations for
Combination Cooling Towers

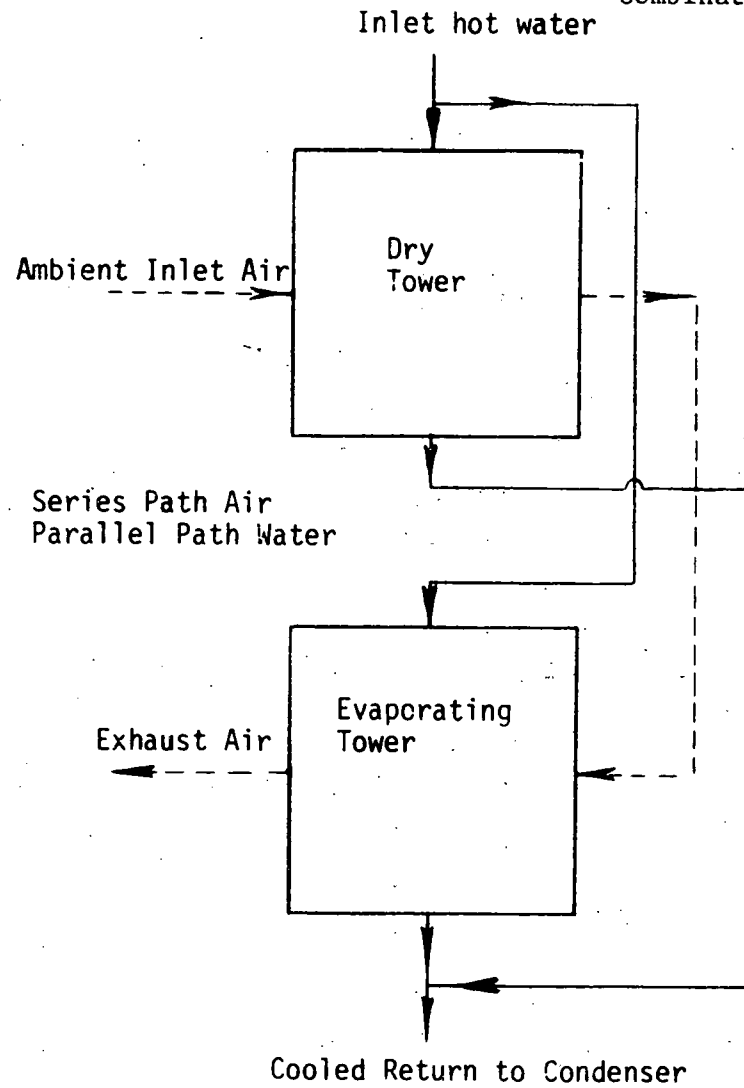


Fig. 4 -1 B

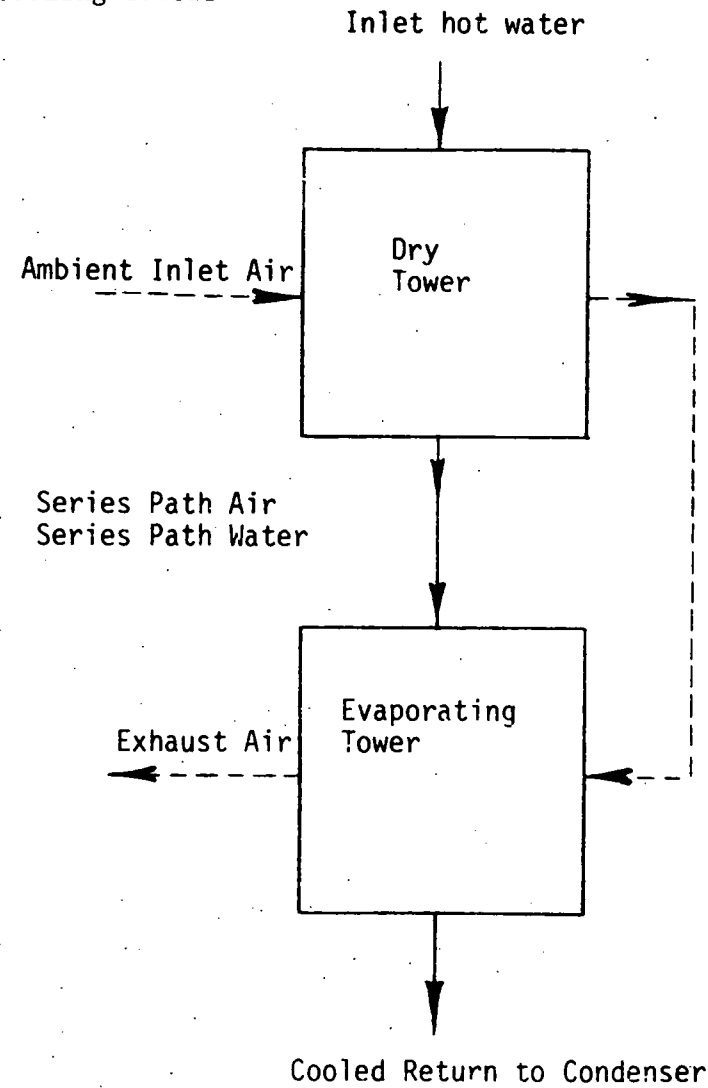


Fig. 4-1 C

Flow Configurations for
Combination Cooling Towers

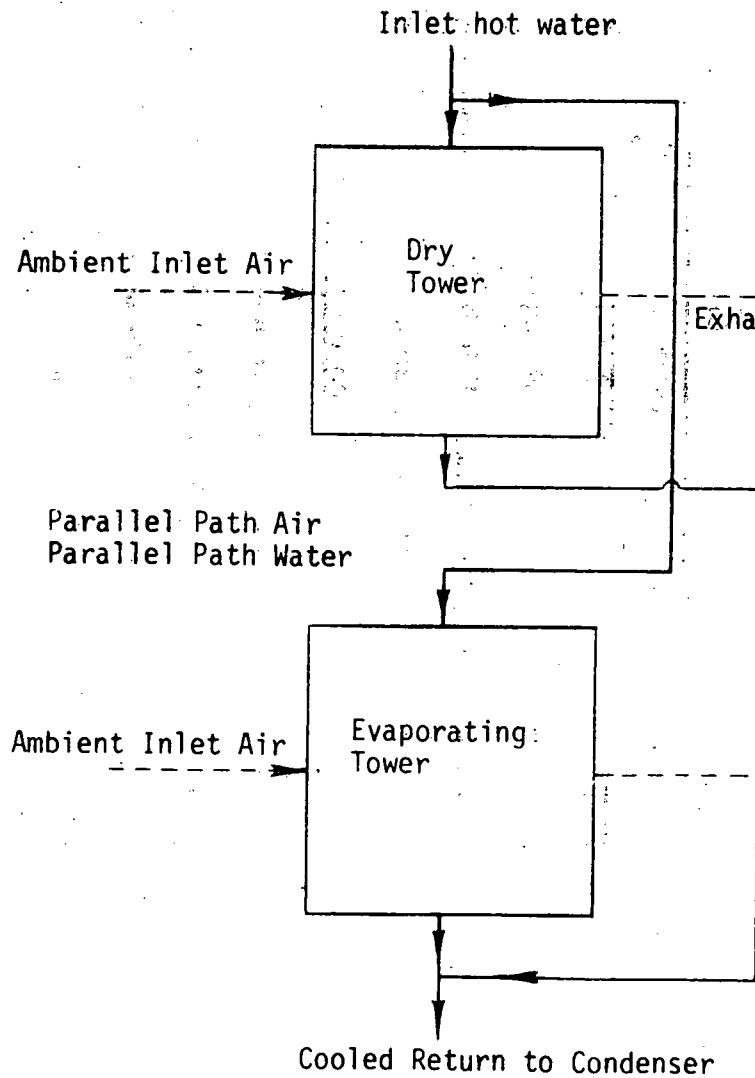
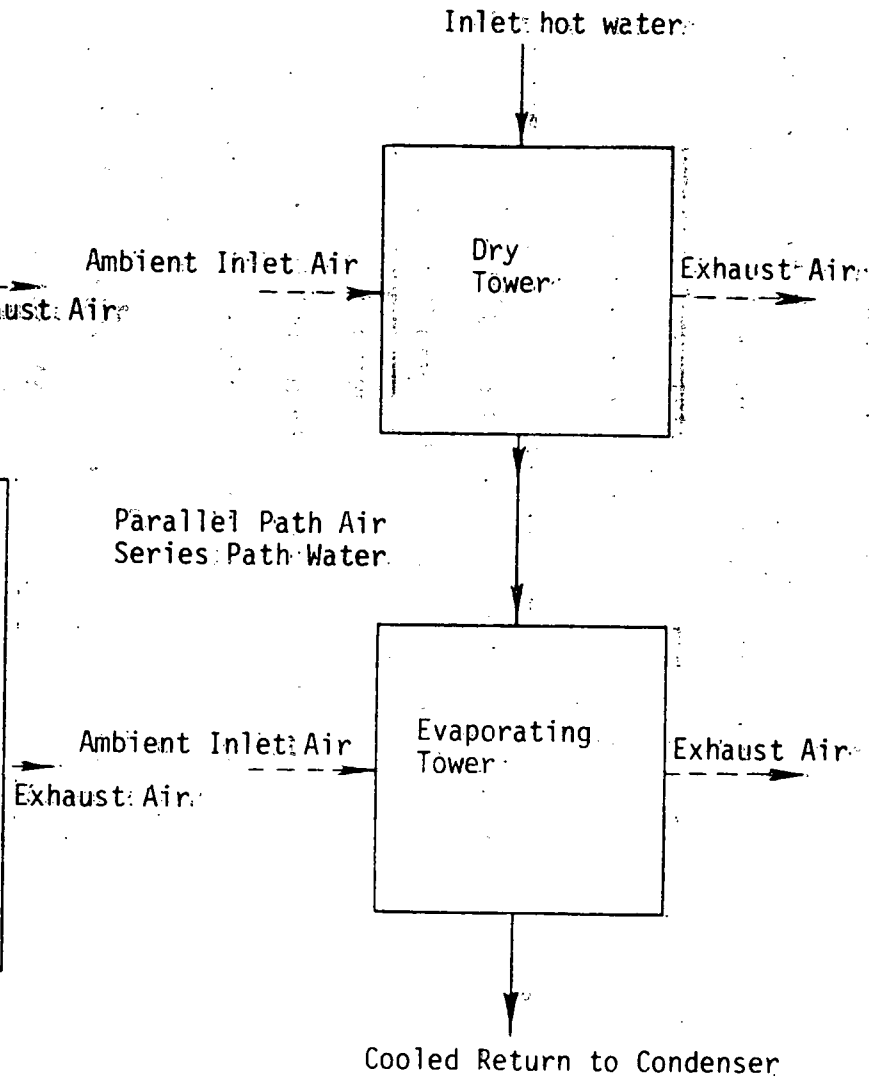


Fig. 4-1 D



below a certain ambient dry-bulb temperature, and a wet tower which is added only at higher ambient temperatures when the dry tower is unable to carry the entire heat load alone. This design has the advantages of being much lower in cost than a single dry tower sized to handle the fixed heat load at all operating conditions [13] while evaporating much less water than a single, equivalent wet tower.

The advantages of this concept are such that it was decided to include combinations of dry and wet-dry concept towers (described in this work) which would also reflect some of the advantages of the conventional dry and wet tower combination systems.

For the purposes of this comparison, it was assumed that the basic component towers (dry, wet, wet-dry) could each be linearly employed; that is, if a certain tower could transfer heat at a rate Q under one set of ambient conditions, and at a rate $2Q$ at some other ambient condition, it would be possible to simply take one-half of the tower out of service to keep the heat transfer rate constant at this second set of conditions. The fan power and water loss would also be reduced by the same fraction. This assumption is more accurate when applied to large cooling tower systems which consist of many controllable units than when applied to a single tower.

The following eight representative combinations will be compared and referred to by the names listed here.

- 1) DRY tower: completely non-evaporative, this tower transfers all heat by forced convection;
- 2) WET tower: deluge-type packing, this tower transfers approximately

85% of its heat load through latent heat of evaporation with the remaining heat transfer due to convection from the water surface;

- 3) WET/DRY tower: V-trough type packing with a wet-to-dry surface area ratio of 5%. This configuration is similar to the model tower which was built and evaluated at M.I.T. within the last year.
- 4) WET + DRY @ 80°F: a DRY tower sized to handle the design heat load below 80°F dry-bulb (27°C) inlet air and a WET tower to help carry the design heat load at higher ambient temperatures.
- 5) WET + DRY @ 60°F: similar to (4), with the DRY tower sized at 60°F (16°C) inlet air and the WET tower added at higher ambient temperatures.
- 6) WET + DRY @ 40°F: similar to (4), with the DRY tower sized at 40°F (4°C) inlet air and the WET tower added at higher ambient temperatures.
- 7) WET/DRY + DRY @ 60°F: a DRY tower sized to handle the required heat load at 60°F (16°C) inlet air and a WET/DRY tower to help carry the design heat load at higher ambient temperatures.
- 8) WET/DRY + DRY @ 40°F: similar to (7), with the DRY tower sized at 40°F (4°C) inlet air and the WET/DRY tower added at higher ambient temperatures.

Performance for each of these configurations was evaluated by means of the computer program listed in Appendix H using the individual design parameters given in Appendix D.

4.2 Performance Predictions

The following sections and figures compare various aspects of performance for the combination cooling towers described above. For the most part these comparisons are based on a constant rate of heat rejection and a constant inlet hot water temperature for each system being evaluated. At high ambient temperatures both components of a combination system share the heat load, with the DRY component operating at its maximum capacity for those conditions and the evaporating (WET or WET/DRY) component making up the rest of the requirement.

As the ambient temperature decreases, the DRY component carries an increasing portion of the fixed heat load as the evaporating tower is cut back to hold the system heat rejection rate constant. When the DRY component sizing temperature is reached, the DRY tower carries the entire heat load and the WET component is completely shut down. Only for temperatures below this point is the DRY component capacity reduced in order to maintain a constant system heat rejection rate.

Use of these graphs must be tempered with the knowledge that each basic tower type (DRY, WET, WET/DRY) has a different mechanism for heat transfer. Usual heat exchanger performance comparison parameters may be misleading when applied to a WET or WET/DRY tower due to the portion of heat transferred by evaporation. For this reason, no attempt has been made to consolidate the findings into a single figure or section, but rather many views of system performance are presented, with overall conclusions appearing below.

4.2.1 Heat Transfer Rate

Figure 4-2 shows the variation of the overall heat rejection capabilities of the three basic designs with inlet dry bulb temperature. These and the following performance curves were taken from data calculated by the computer program listed in Appendix H.

DRY tower heat transfer rate under these conditions is linearly dependent on the initial temperature difference (ITD) between the ambient air and the inlet hot water. Thus the total heat rejection rate at $T_{amb} = 90^{\circ}\text{F}$ (32°C), $\text{ITD} = 40^{\circ}\text{F}$ (22°C), would be half that at $T_{amb} = 50^{\circ}\text{F}$ (10°C), $\text{ITD} = 80^{\circ}\text{F}$ (44°C), etc.

The WET tower has two heat transfer mechanisms which must be taken together to determine the overall heat transfer rate. One, dry convection from the hot water surface is essentially the same as that of the dry tower above, decreasing linearly to zero as $T_{amb} = T_{l,in}$. This mechanism, however, makes up only 15% of the total heat rejection on the average and thus has small influence on the total performance of the tower. The major portion of the heat load is transferred by evaporation which varies linearly as to the difference of the partial water vapor pressures of the ambient air and inlet hot water. For most of the operating range of the WET tower, this difference is a strong function of the inlet hot water temperature due to the non-linearity of the temperature-saturation pressure curve for water. Only at ambient temperatures close to the inlet hot water temperature, does the moisture content of the ambient air become an important parameters (Figure 3-3).

The combined result of these two mechanisms gives the WET tower a fairly flat performance curve over most of its operating range.

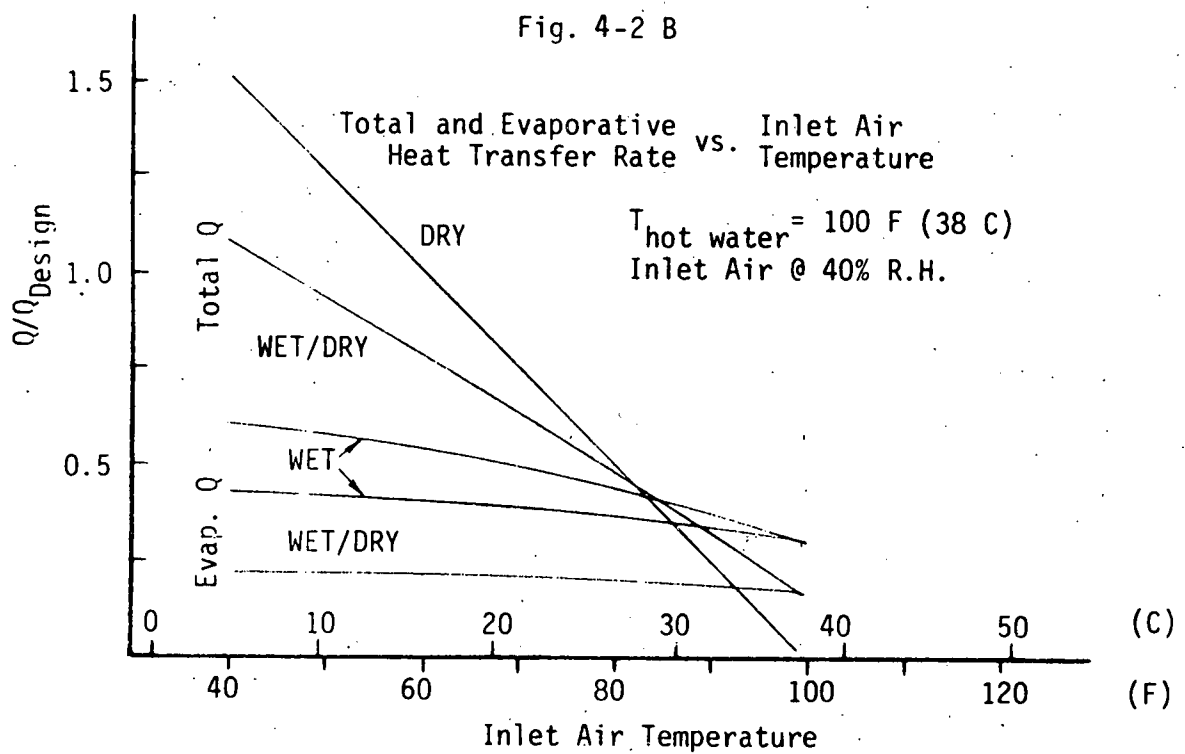
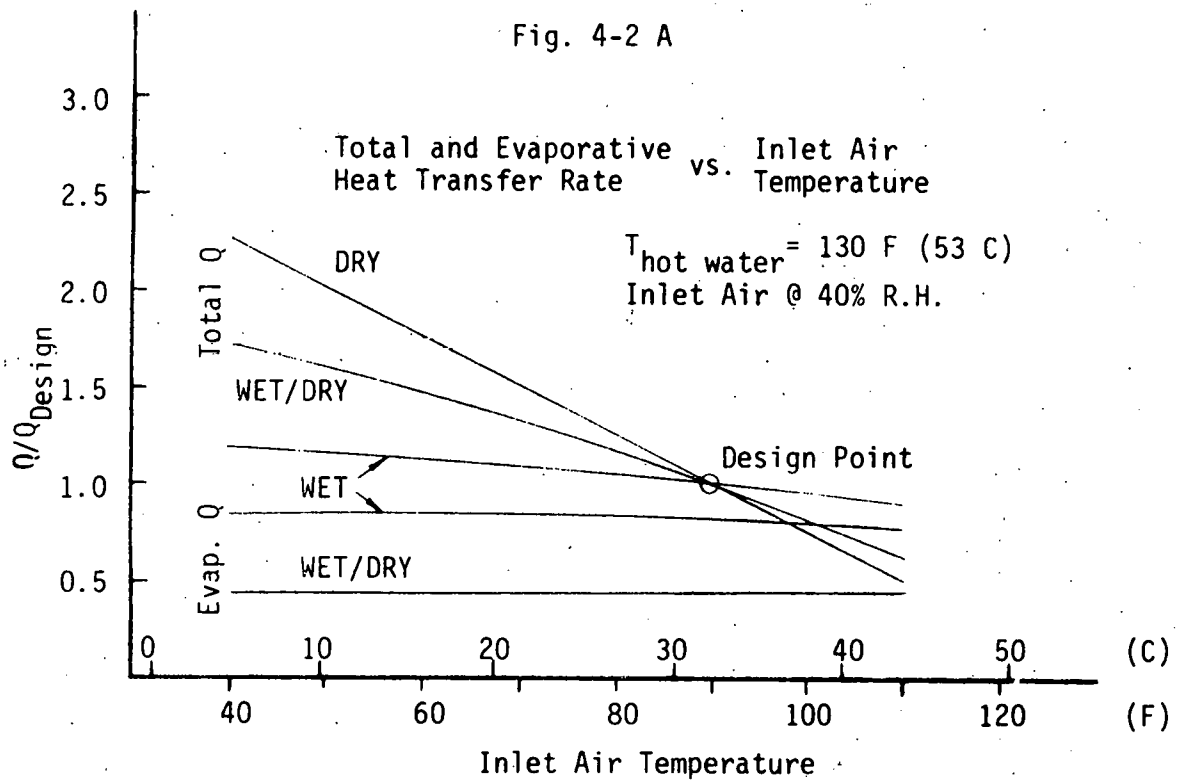


Fig. 4-2 C

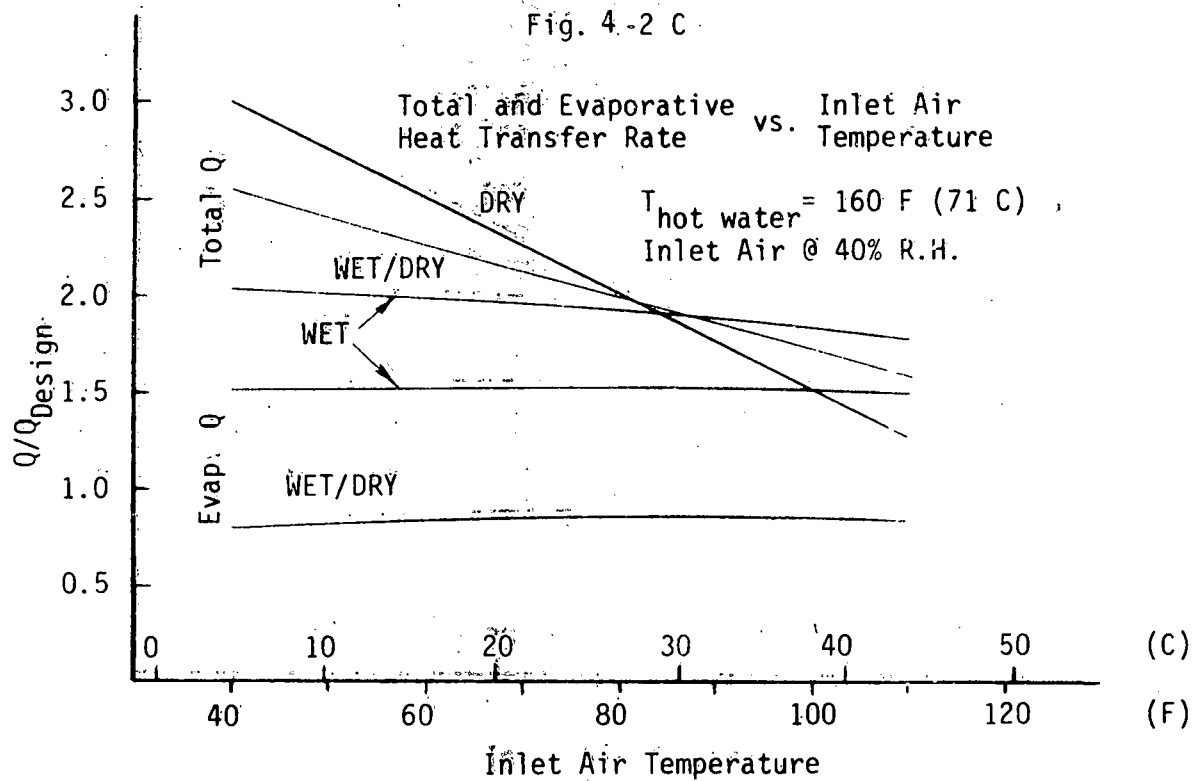
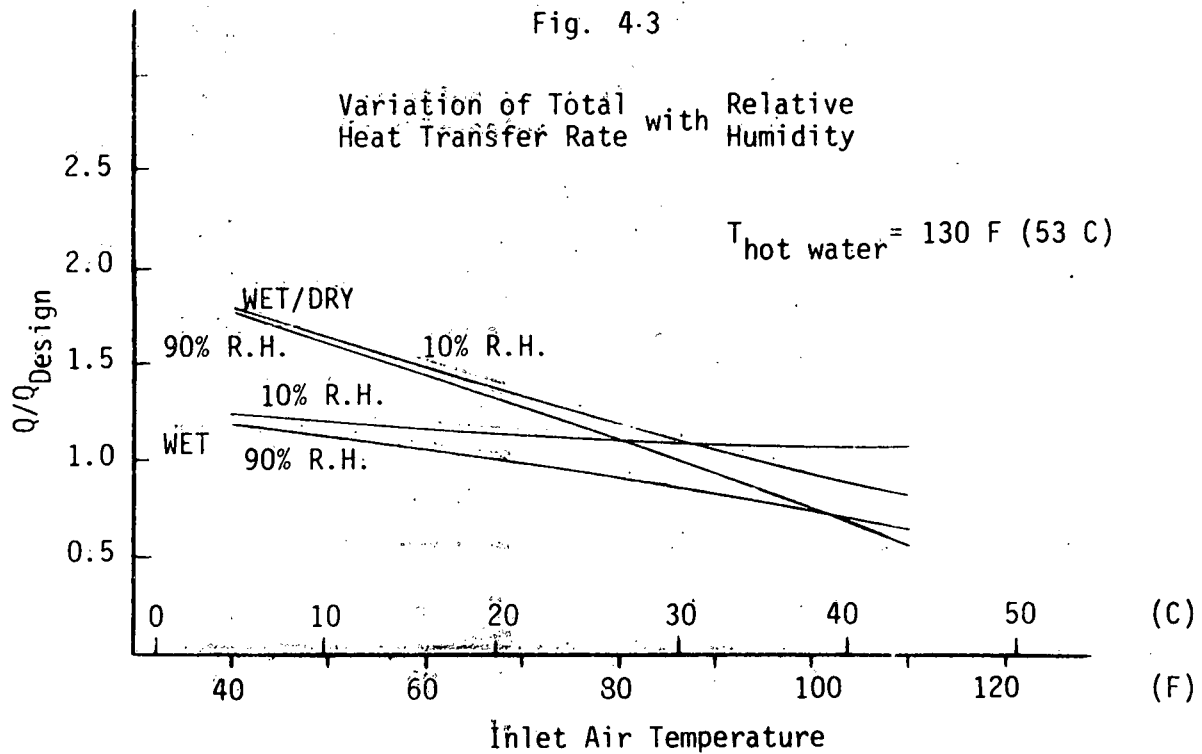


Fig. 4-3



For the WET/DRY tower, the same two mechanisms for heat transfer are also operating, but their proportional share of the heat load has been changed. On the average about 60 to 70% of the heat load is rejected by dry convection with the remaining 30 to 40% rejected through evaporation. Thus the WET/DRY tower seems to behave much more like the DRY tower, where overall capacity increases with the ITD. The WET/DRY tower is also less affected than the WET tower by ambient air moisture content at high ambient air temperatures (Figure 4-3).

4.2.2 Air Flow Requirements

Figure 4-4 compares the required air mass flow rates of the various designs using a purely WET tower as a basis. Heat load is held constant here at Q_{design} and the combination towers are employed as described in Sec. 4.1. Air mass flow is proportional to fan equipment and power consumption as well as overall tower size.

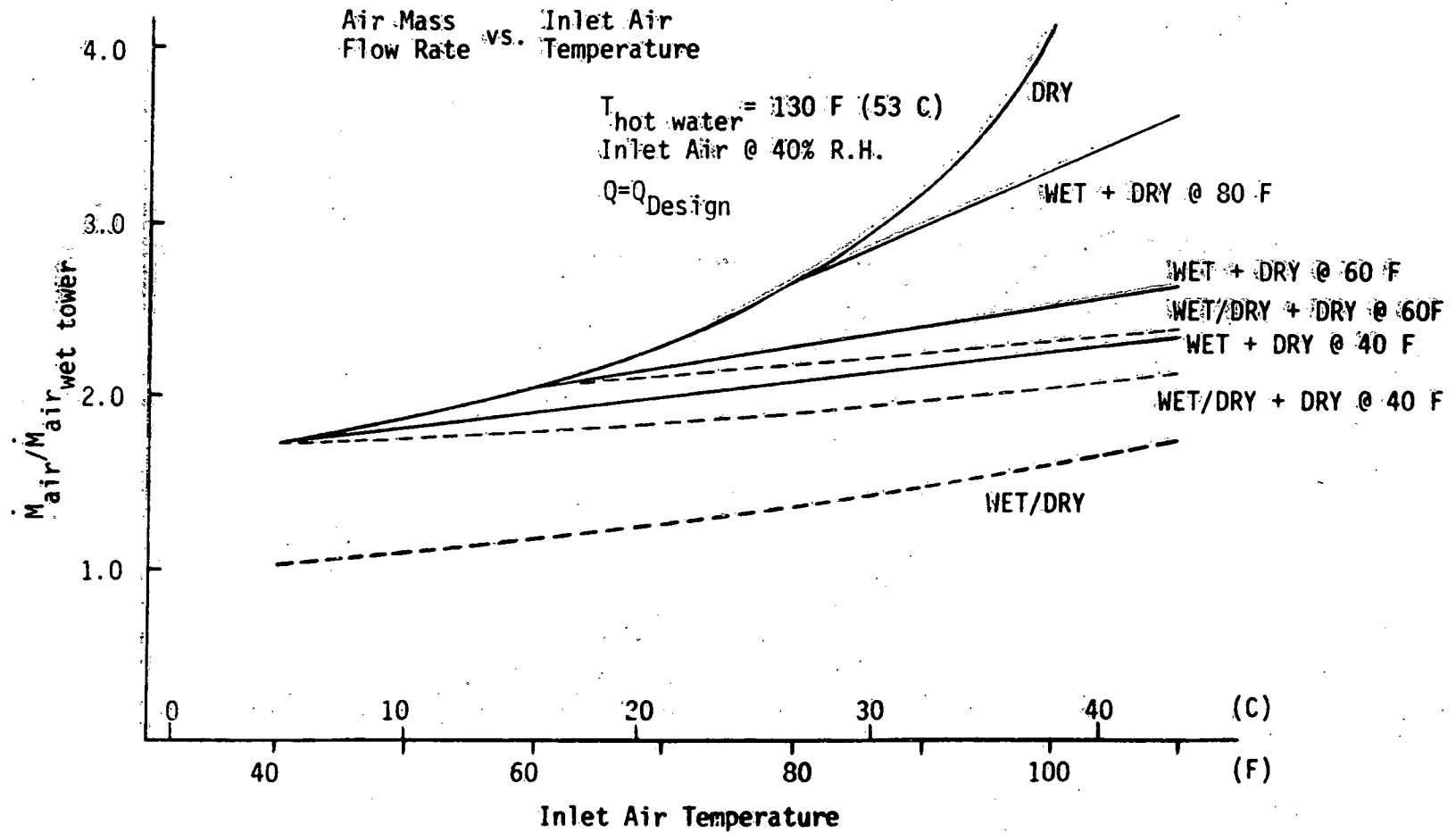
For a DRY tower the necessary air mass flow increases rapidly as the initial temperature difference is reduced.

The WET + DRY combination towers behave as a purely DRY tower until they reach the temperature for which the DRY portion was sized. Above this design point the rate of increase of the air flow rate is reduced as the WET portion now takes a growing share of the heat load.

The WET/DRY tower operated above the WET tower flow rate, but well below those of the WET + DRY towers in all cases. It also exhibits much less of a tendency to rise at higher temperatures than does the DRY tower.

The WET/DRY + DRY towers require slightly more air flow than the WET + DRY towers with the same all-dry design temperature. This was due to the reduced evaporative share of the heat load and the higher air flow requirements of the WET/DRY tower over a WET tower.

Fig. 4-4



This graph does not take into account air pressure drop across the tower which is an essential parameter of pumping power costs.

4.2.3 Water Consumption

Figure 4-5 compares the rate of water consumption of the various towers for a fixed heat rejection rate using a purely WET tower continuously operating at 90°F dry-bulb (32°C), 40% R.H. for comparison. Inlet hot water for all towers was assumed to be constant at 130°F (54°C).

The combination towers of both WET + DRY and WET/DRY + DRY configurations consume no water until inlet conditions rise above the DRY component design temperature at which time the evaporating component must be brought into service. The water consumption rates for all designs then rise to meet where $T_a = 130^\circ\text{F}$ (54°C), at which point all heat transfer must be done by evaporation.

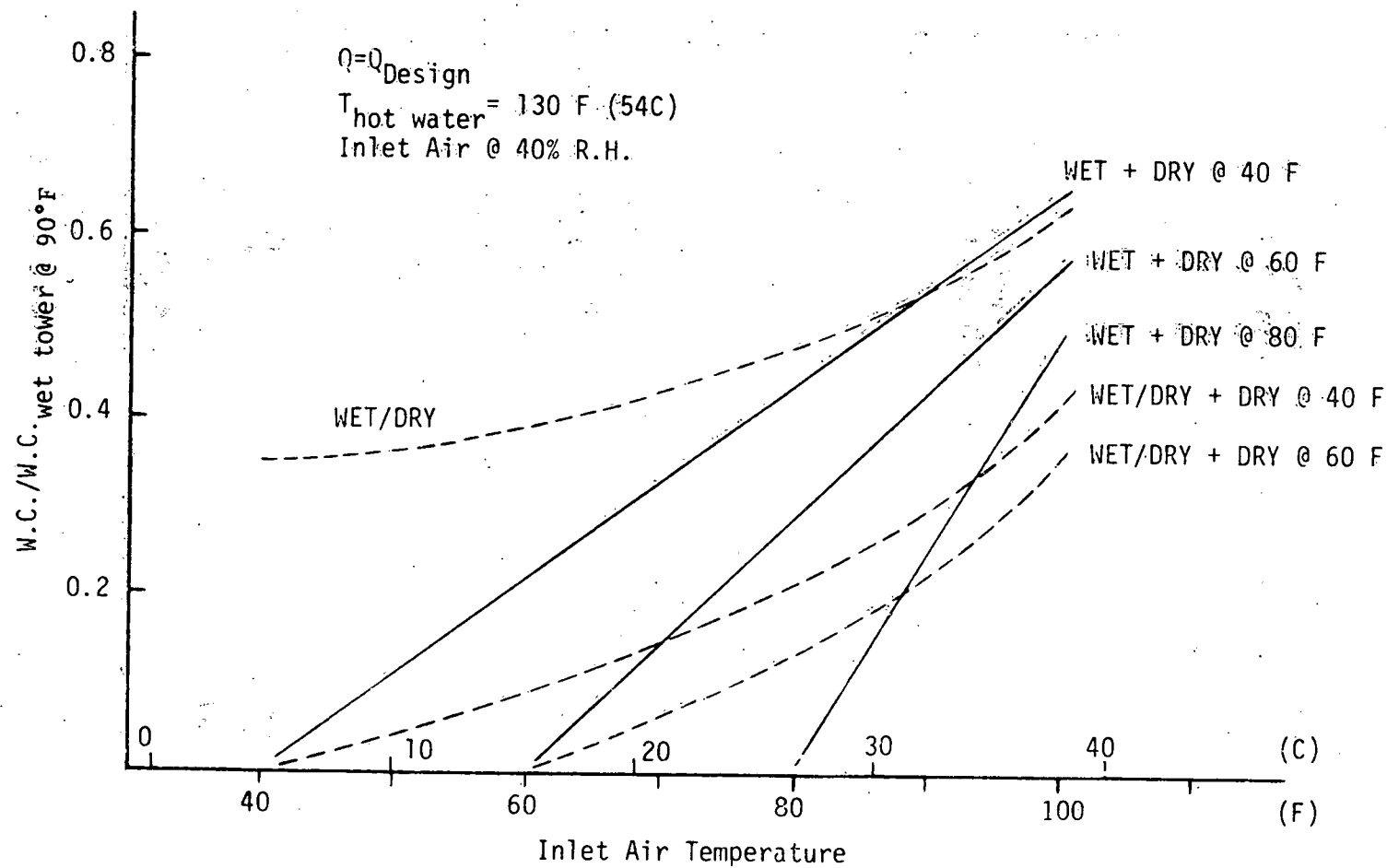
The WET/DRY tower has a relatively constant consumption rate in the lower operating ranges, but begins to consume more water as inlet air temperature increases and the convective heat transfer rate is reduced.

All the combination towers and the WET/DRY tower consume much less water than a WET tower under the same conditions. All WET/DRY + DRY systems consume far less water than similar WET + DRY systems having the same DRY tower design temperature.

The variation of water consumption rate with ambient and inlet hot water temperature was investigated as one means of optimizing the wet-dry design. The WET/DRY tower used in the previous comparisons was evaluated

Fig. 4-5

Water Consumption Rate vs. Inlet Air Temperature
for Combination Towers



with varying inlet temperatures and fixed heat load. The results appear in Figure 4-6.

As an example, for the $T_{\ell_{in}} = 130^{\circ}\text{F}$ (54°C) curve, water consumption is seen to rise with ambient air temperature as more of the tower comes on line and evaporation takes over a growing share of the heat load. Finally, at 90°F (32°C), the WET/DRY tower is operating at full capacity. Any further increase in ambient air temperature would cause the tower heat transfer rate to drop below the fixed heat load consumption of the WET/DRY tower for fixed hot water inlet temperature of 130°F (54°C) over a range of inlet air temperatures.

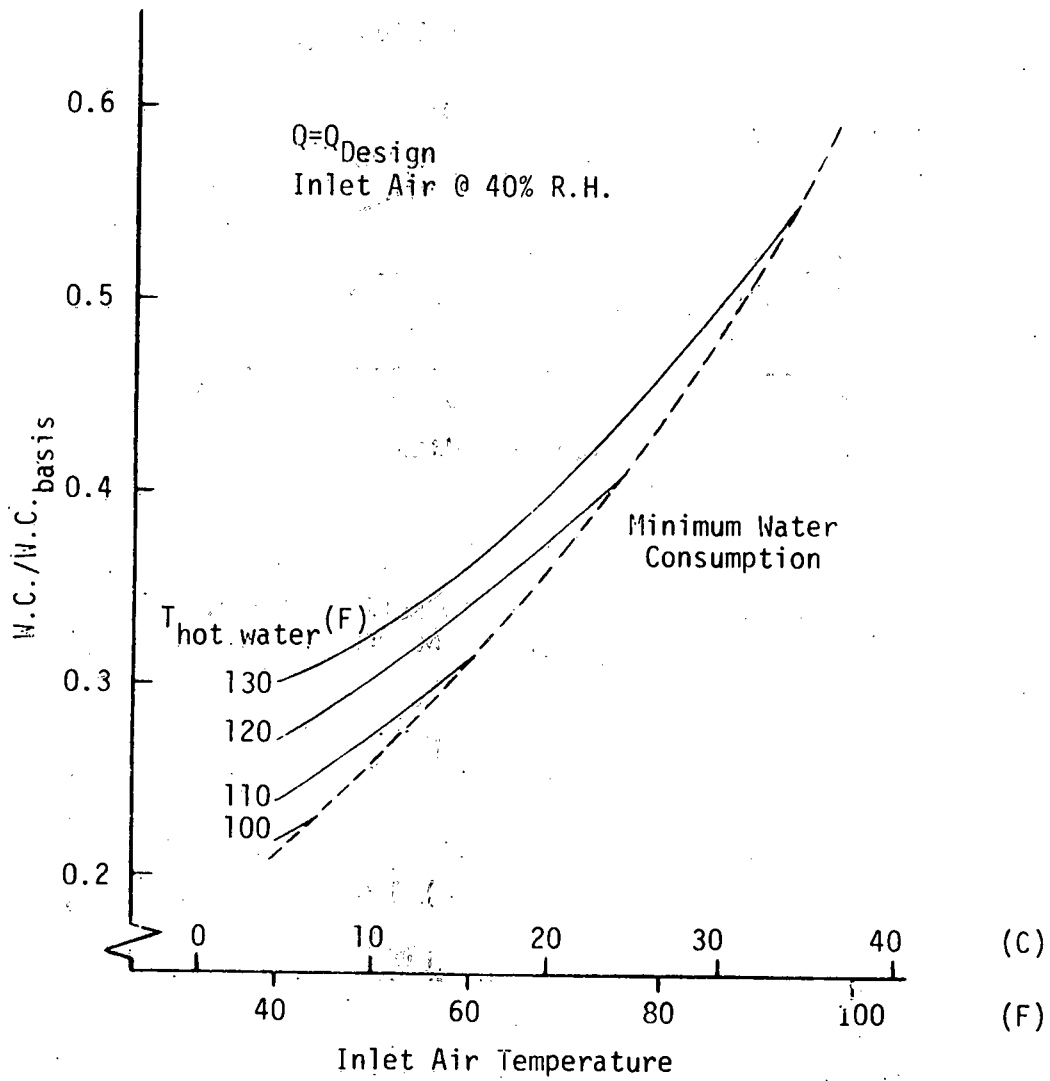
The same procedure was followed for $T_{\ell_{in}} = 120^{\circ}\text{F}$ (49°C), 110°F (43°C), and 100°F (38°C) with the same tower design. These curves also appear in Figure 4-5. The endpoints and the dashed line represent water consumption and inlet air temperature when the design is allowed to run continuously at full capacity under a fixed heat load.

A WET/DRY tower system, optimized for minimum water consumption, would therefore be run at maximum capacity whenever possible, and only cut back when $T_{\ell_{in}}$ (and thus turbine back pressure) falls below an acceptable level. This also minimized turbine back pressure under all operating conditions and increases power plant gross output.

Running the tower system at full capacity also means higher operating and maintenance costs, but these should be more than offset by reductions in power generating costs.

Fig. 4-6

Variation of Water Consumption Rate
with Inlet Air and Water Temperature



Figures 4-7 A,B,C,D compare the monthly water consumption rates of the various systems for locations around the country. All systems had a constant heat load, inlet hot water of 130°F (54°C), and were again compared using a WET tower working at 90°F (32°C) for a basis.

Water consumption for all designs in general climbs during the summer months and declines during the winter. It can also be seen that the WET + DRY @ 60°F and the WET/DRY + DRY @ 40°F designs have approximately equal water consumption in each location.

Climatic data were summarized from reference [14] and individual configuration performance under various conditions (evaporation rate, heat transfer rate, etc) taken from the preceeding figures. This use of hourly temperature distributions gives an accurate description of the weather conditions under which towers would be operating and makes possible this sort of analysis.

Please note that each value is a monthly consumption, independent of the preceeding and following months. For clarity the curves are continuous rather than stepped.

Figures 4-8 A,B represent the yearly totals of the monthly water consumption values shown in figures 4-7 A,B,C,D. A WET tower operating at 90°F (32°C) was again the basis.

The WET/DRY tower is shown to consume approximately 60% less water than a WET tower under the same conditions. The consumption totals for the WET + DRY @ 40°F are also fairly close for most locations, and about 90% lower than a comparable WET tower.

Fig. 4-7 A

Seasonal Variation of Water Consumption Rate
for Albuquerque, N.M.

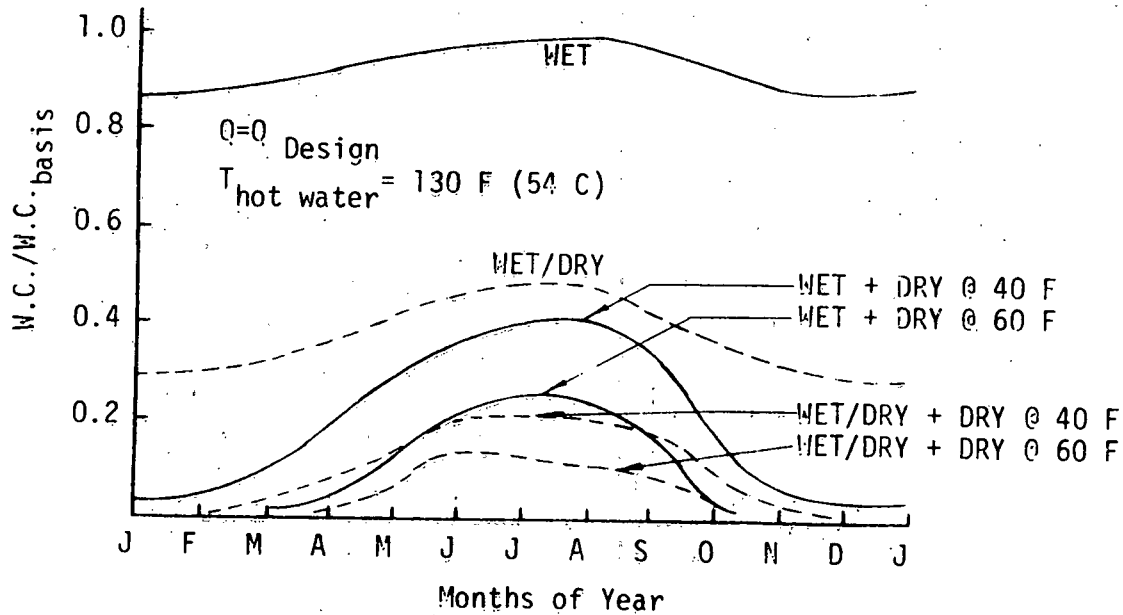


Fig. 4-7 B

Seasonal Variation of Water Consumption Rate
for Boston, Mass.

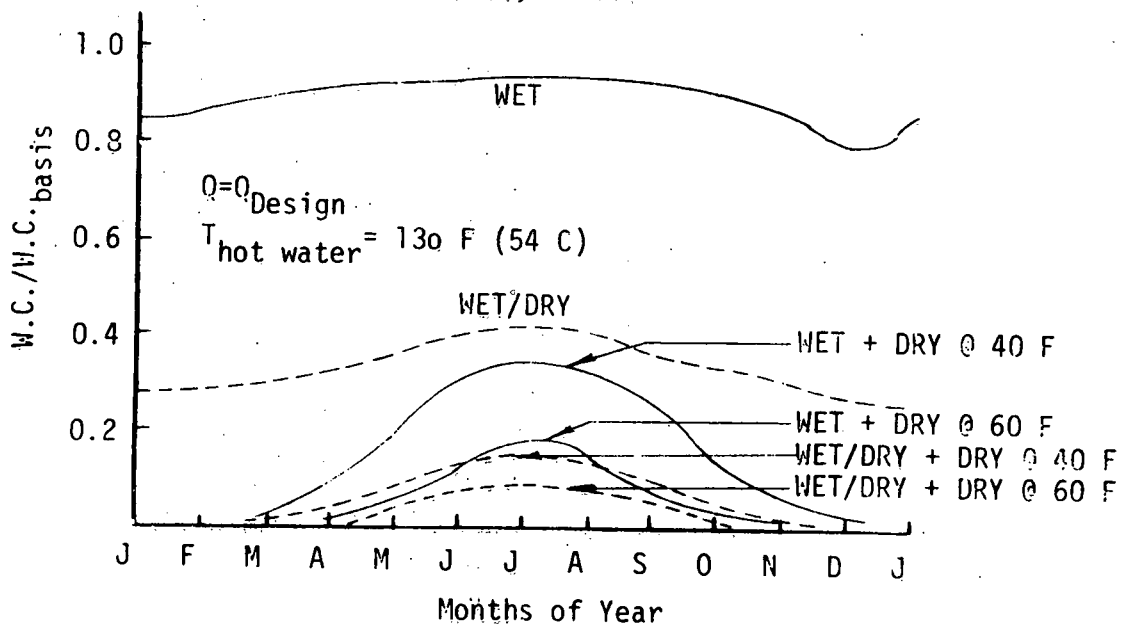


Fig. 4 -7 C

Seasonal Variation of Water Consumption Rate
for New, N.Y.

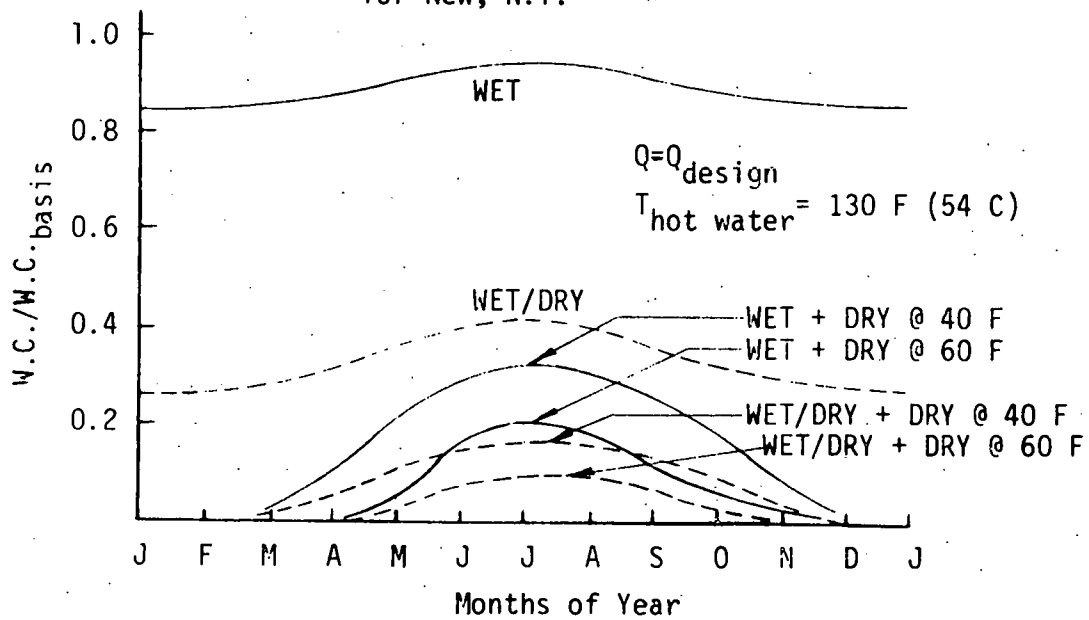
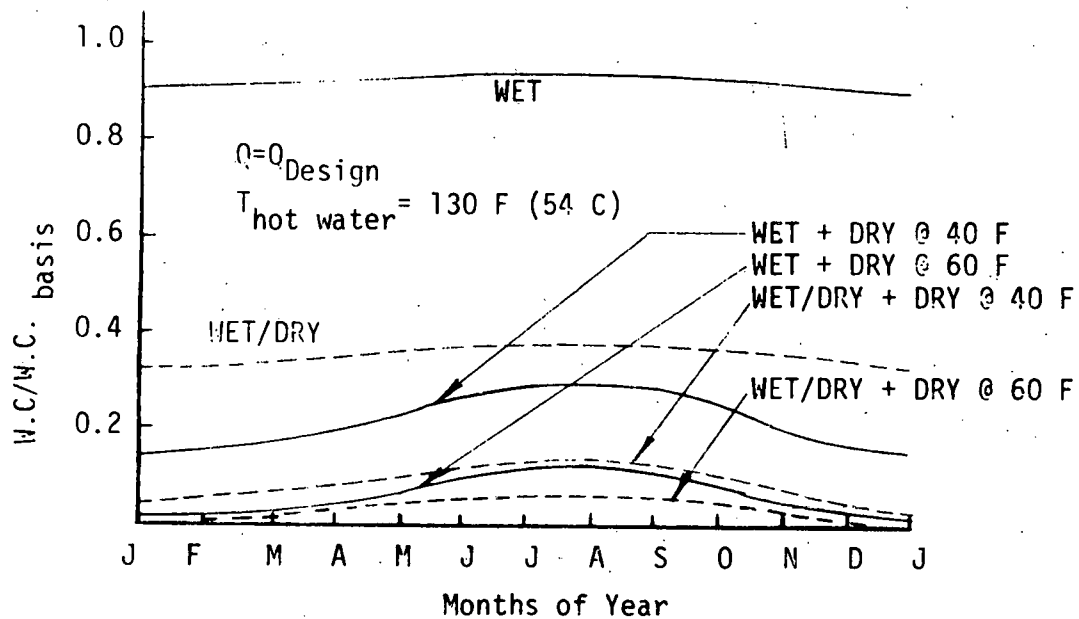
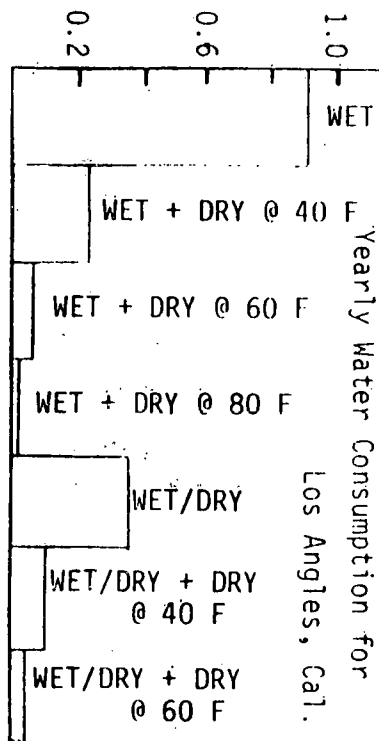


Fig. 4 -7 D

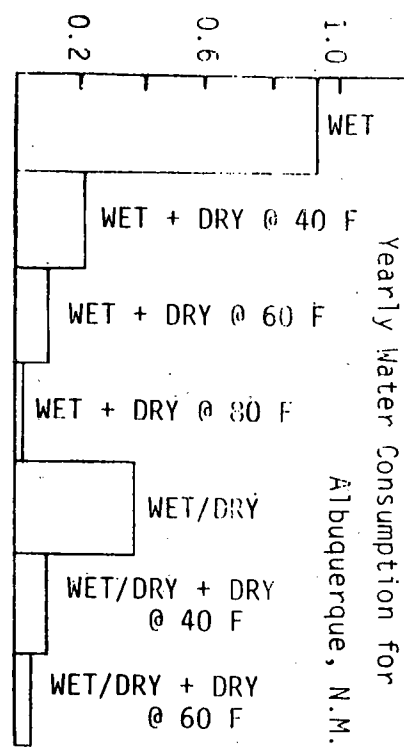
Seasonal Variation of Water Consumption Rate
for Los Angeles, California



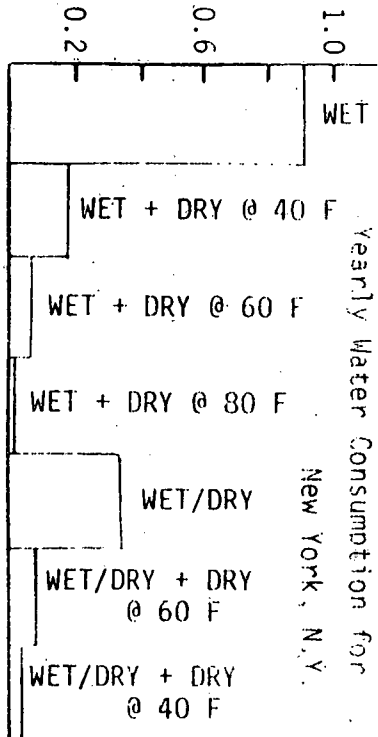
W.C./W.C. wet tower @ 90F



W.C./W.C. wet tower @ 90F



W.C./W.C. wet tower @ 90F



W.C./W.C. wet tower @ 90F

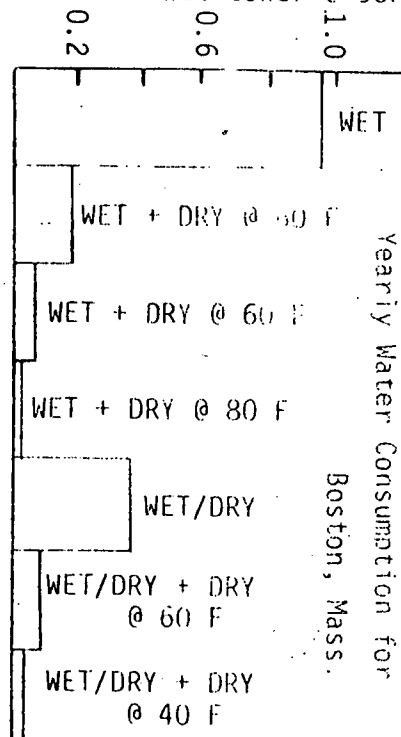


Fig. 4-8

4.2.4 Minimum Turbine Back Pressure

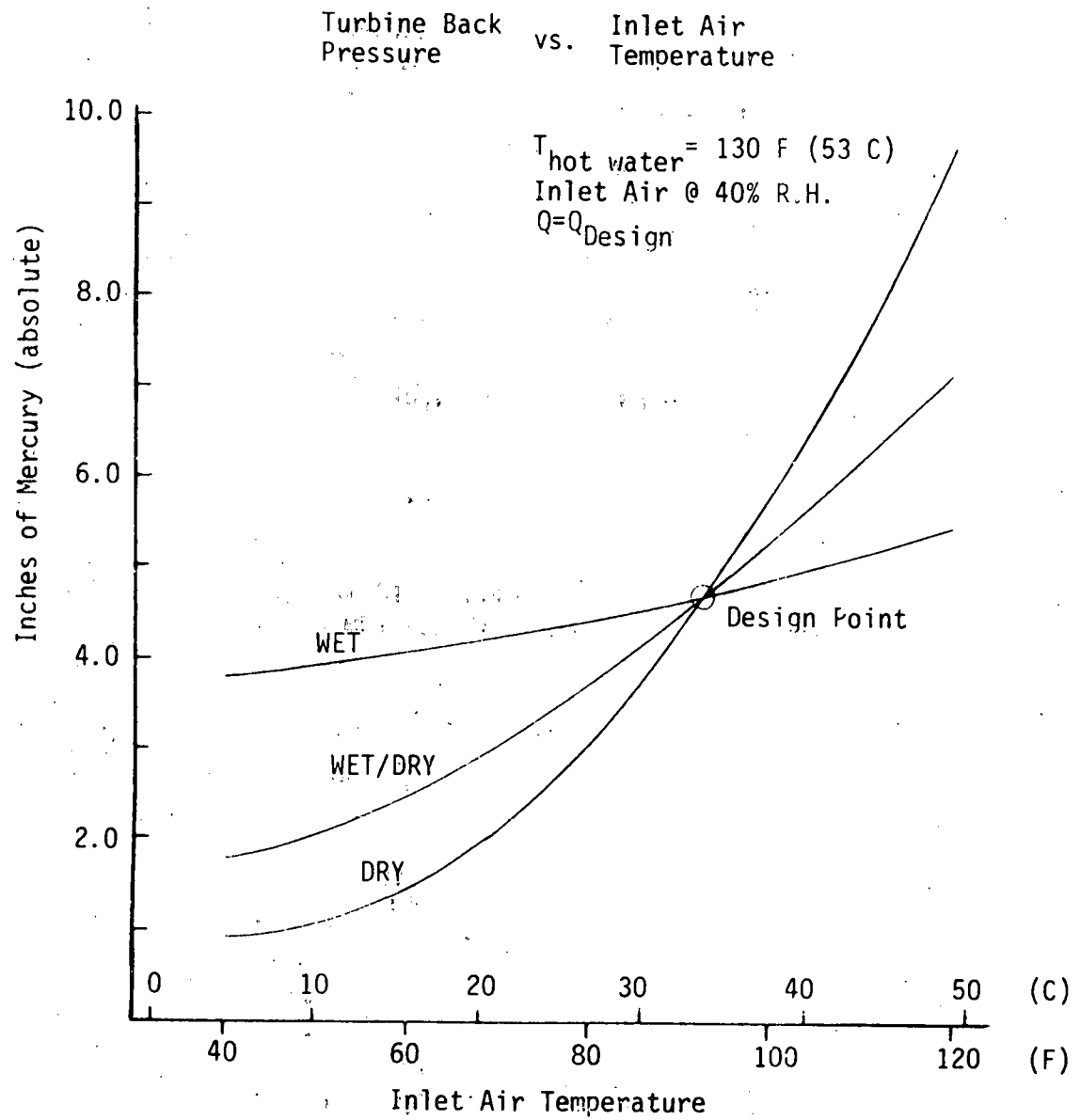
Figure 4-9 shows the variation of condenser pressure with inlet air temperature for the three basic component towers. At lower ambient temperatures it is possible, by running the towers at full capacity, to reduce the hot inlet water temperature while maintaining the design heat rejection rate.

The inlet hot water temperature is related to the turbine back pressure and overall power plant efficiency. Modern turbines can have an operating range from 1.5 up to 15 in. of Hg ($300\text{--}1300\text{ N/m}^2$) back pressure with efficiency decreasing and capital costs increasing very rapidly for designs above 5 in. of Hg. (600 N/m^2). Most turbines now being installed are low back pressure designs.

Note that the DRY tower is extremely sensitive to ambient temperature and requires very hot water above the design point of 90°F (32°C) inlet air. The WET/DRY and WET towers are respectively less sensitive and could continue to deliver full service at higher temperatures.

In practice, the inlet hot water temperature would be allowed to "float" with ambient conditions, rising or falling until the turbine design limit is reached. At low ambient temperatures where the turbine back pressure is likely to drop below the design limit, the tower system capacity would be reduced to maintain the minimum back pressure. At high ambient temperatures where maximum turbine back pressure would be exceeded, the generating turbines would be throttled back to reduce the total heat load requirement to within the capacity of the tower system under the prevailing ambient conditions.

Fig. 4-9



4.2.5 Incidence of Fogging

Figures 4-10,11 show the exhaust air conditions (temperature, for each relative humidity, specific humidity of the three basic tower designs.

Inlet conditions (ambient air temperature and hot water temperature) were varied over the normal operating range of a cooling tower system, while ambient air relative humidity was set at 90%.

Except at high ambient temperature conditions, the WET tower exhausts air at over 100% relative humidity, i.e. with entrained air droplets. This indicates the immediate presence of a fog plume. At air exhaust conditions of less than 100% R.H. fogging may occur depending on the mixing conditions of the ambient air and the tower exhaust. If the mixture passes through the saturation region for water vapor during mixing, fog will form in the area around the tower.

The WET/DRY tower, with its reduced evaporative heat transfer, does not produce a fog plume even under the most unfavorable ambient conditions.

Some types of WET+DRY combination tower systems mix the exhaust air from each component before discharging it to the atmosphere. This will reduce the incidence of fogging by allowing the wet tower exhaust to pre-mix with hot air at ambient humidity before being discharged.

However, due to cost and land use considerations, the more recent trend in combination systems has been to build two separate towers, each exhausting separately into the atmosphere. In this case the individual towers would be described as in figures 4-10 and 4-11.

Fig. 4-10 A

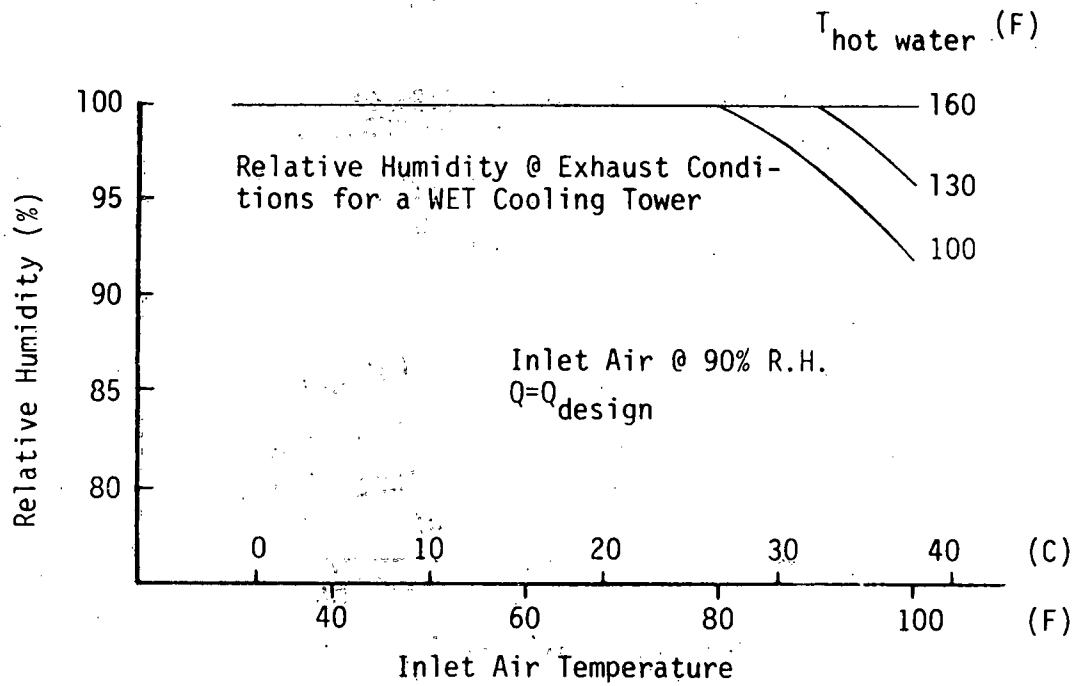


Fig. 4-10 B

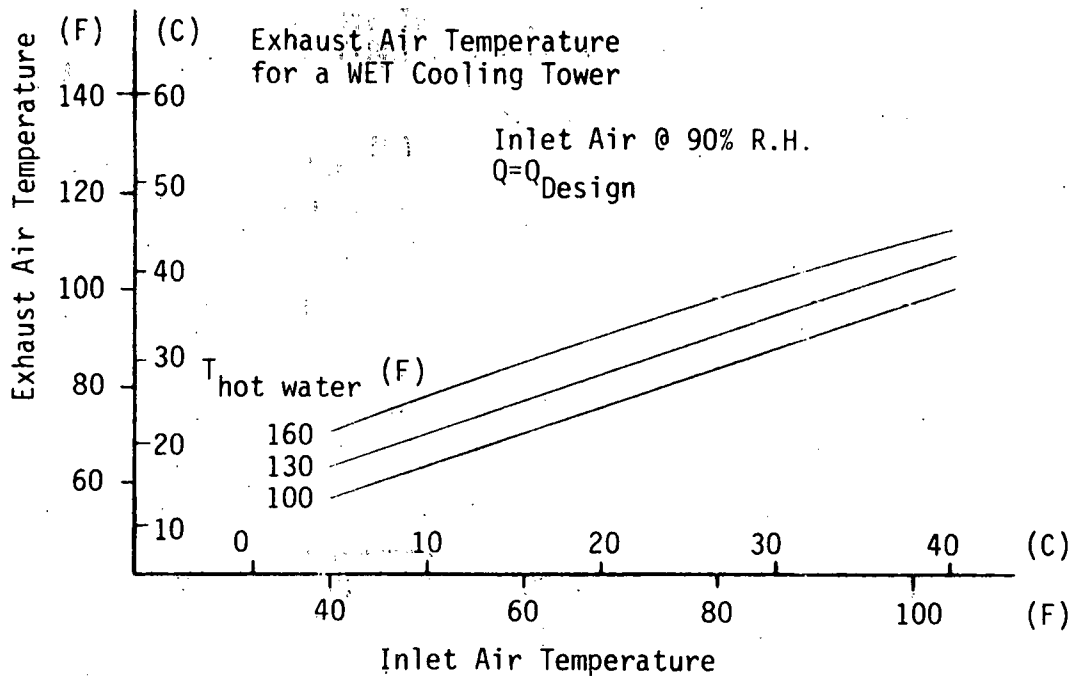


Fig. 4-10 C

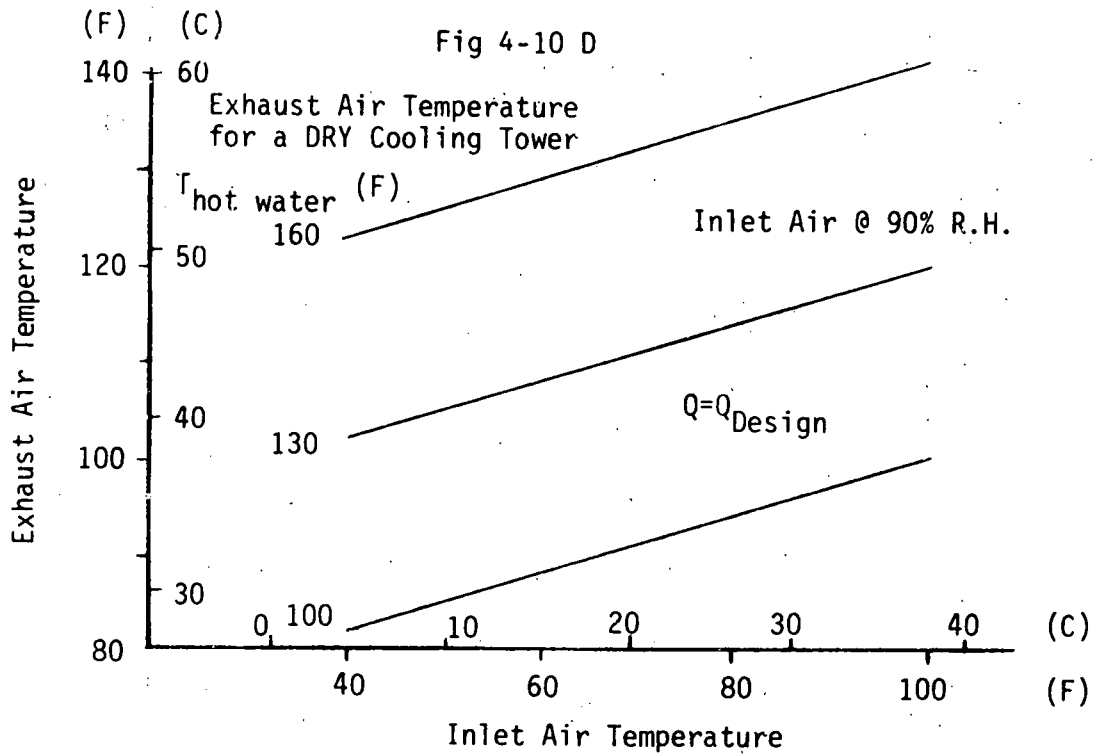
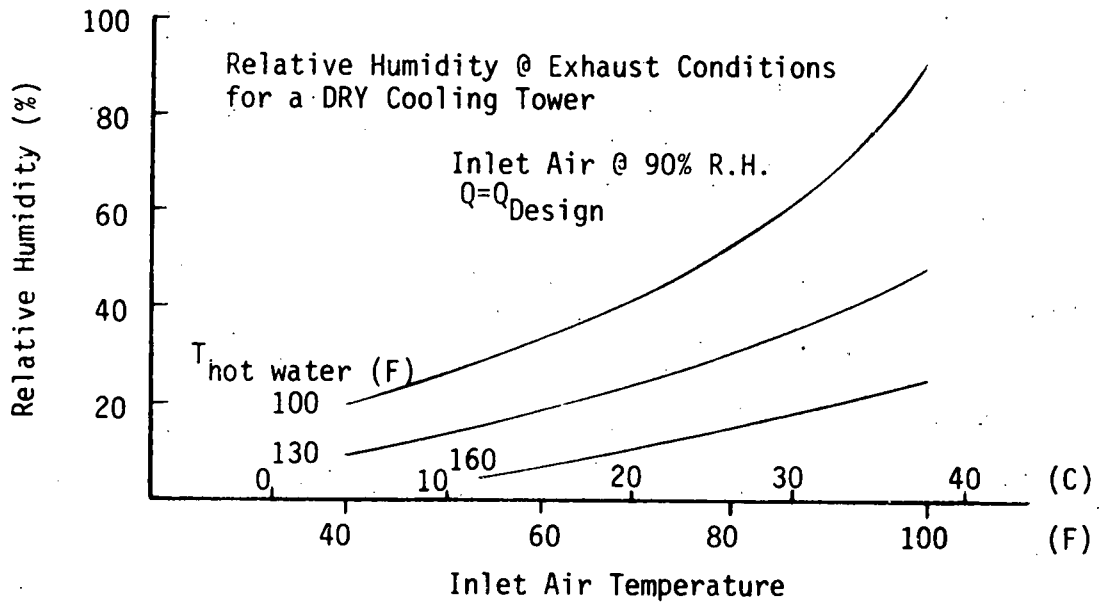


Fig. 4-10 E

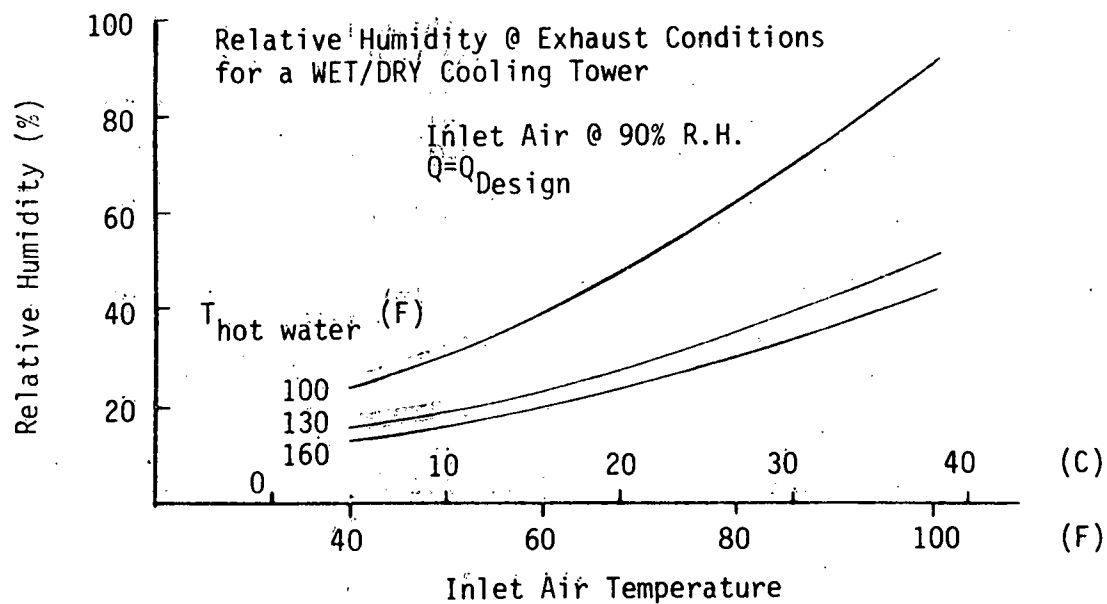


Fig. 4-10 F

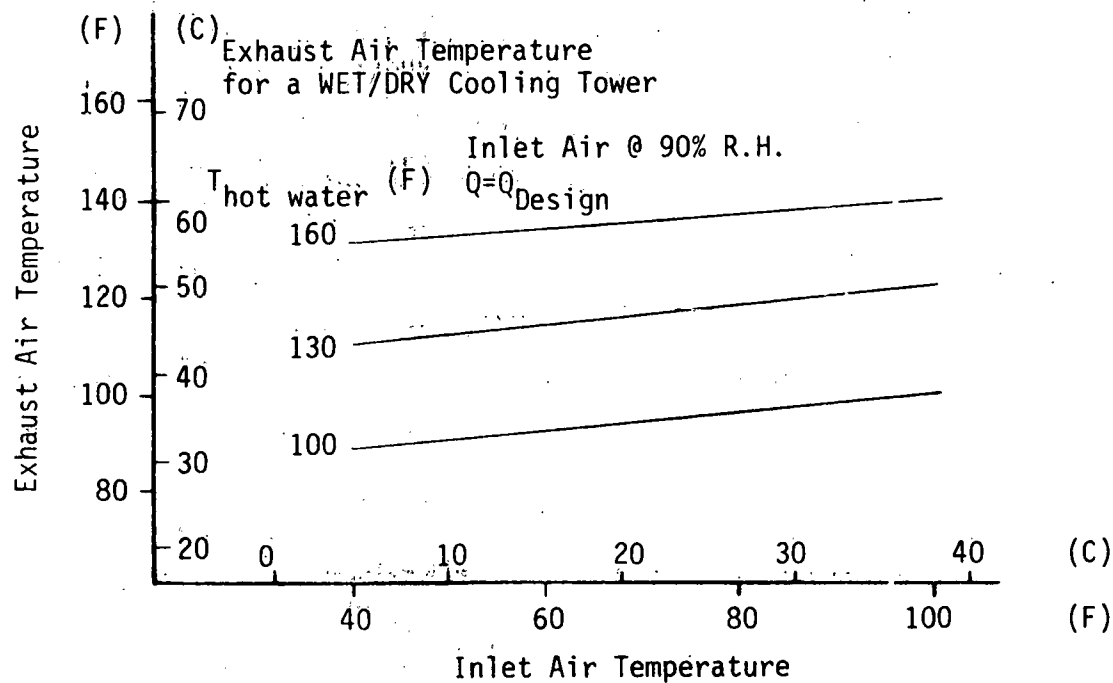


Fig. 4-11 A

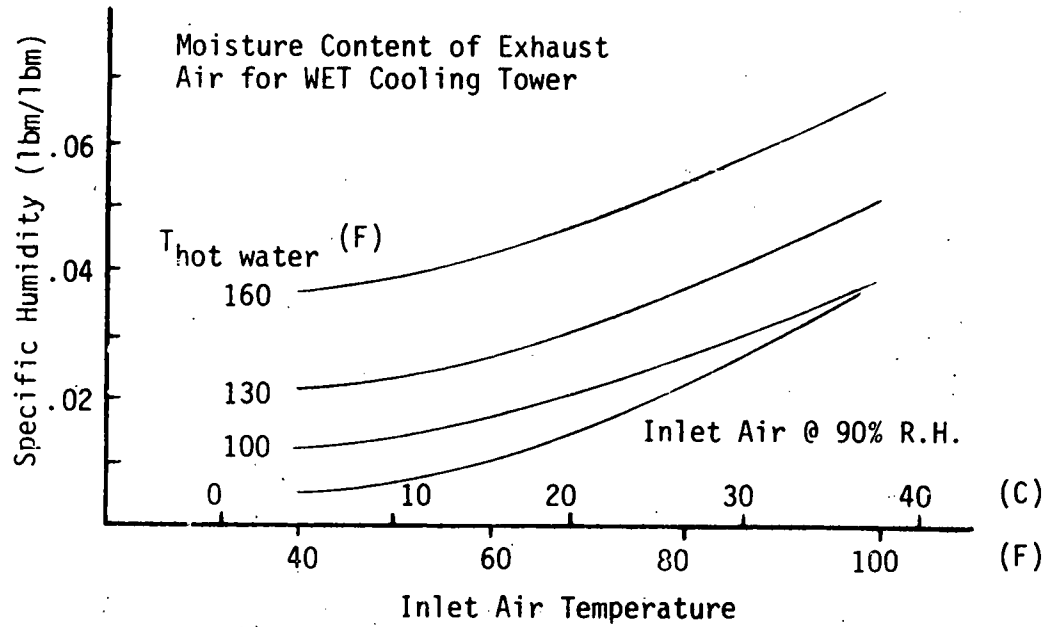
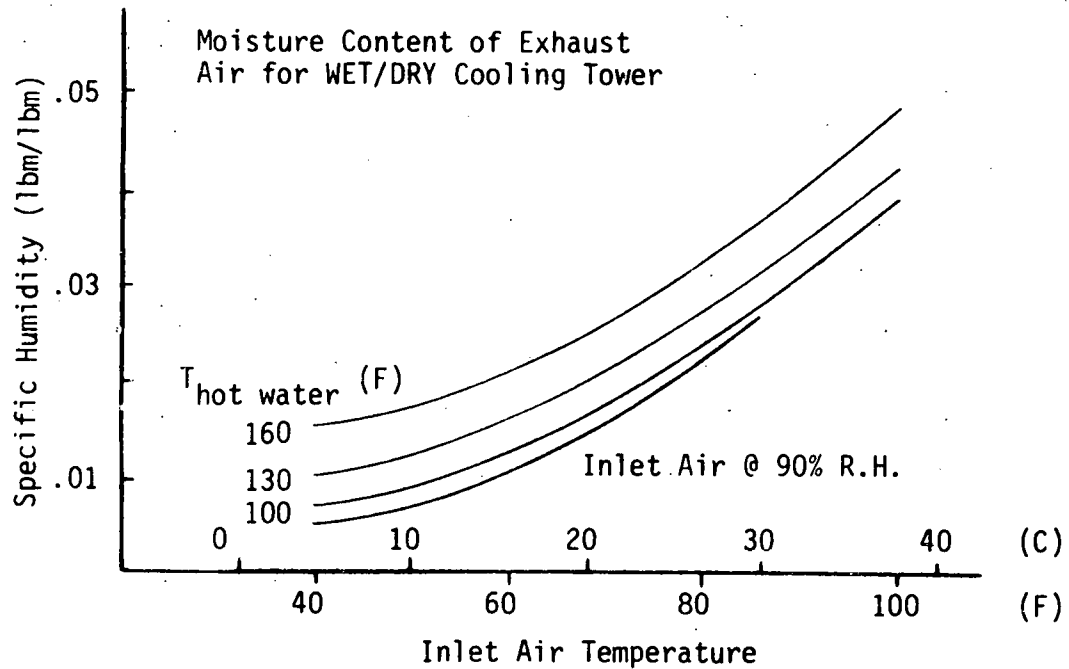


Fig. 4-11 B



CHAPTER 5

CONCEPTUAL DISCUSSION OF CROSSFLOW CONFIGURATION5.1 Need for a Crossflow Design

Early in the project it was recognized that the counterflow configuration would pose difficult design questions which would have to be answered before the concept could be used in practice.

The present design has the cool air entering at the bottom of the packing section and exiting at the top [1]. The hot water is distributed over the packing plates at the top of the section and collected at the bottom, hence the designation counterflow. Since the distribution and collection of the water is done in the airstream, great care had to be taken in the original design to reduce the amount of evaporation which would take place.

This was eventually done by distributing the hot water by rows of copper pipes with evenly spaced holes drilled along their length. These were arranged so as to provide one stream of water for each channel on each packing plate. Collection was accomplished by large sheet metal gutters at the bottom of each packing plate which would catch the cooled water and channel it quickly aside out of the air flow (Figure 2-1).

On an experimental level, this design presented several problems. Alignment of both the pipes and gutters was difficult and had to be done very closely to avoid splashing and dripping of water in the airstream. The distribution pipes became clogged with impurities and

had to be individually cleaned [1]. The collection gutters were large enough to warm the incoming air about 10°F (6°C) or more before it reached the packing section, thus precluding low air temperature tests. Despite the above precautions, in some tests the humidity change across the distribution system matched the change across the entire packing section.

For a full-size tower the size of the pipes and gutters would have to be increased to account for the greater width of the tower. Clogging and alignment would be an even greater problem, while the larger pipes and gutters would decrease the airflow area and require more fan power for an equivalent rate of heat transfer. Evaporation in the distribution section would continue at what is considered an unacceptable level.

The solutions to each of these problems of the counterflow design would be costly and complicated. In an effort to keep the design both simple and effective, it was decided to investigate another configuration.

Instead of having the air flow from the bottom of the packing section to the top, the proposed design would have the air flowing across the plates horizontally, thus "crossflow".

This idea has several advantages:

- 1) It is possible to distribute and collect the hot water in an enclosed space, out of the airflow. This would eliminate most of the unwanted evaporation.
- 2) The distribution and collection are simplified. It would be possible to distribute the water with large spray nozzles

and collect it in a simple catch basin, thus cutting both fabrication and maintenance costs.

- 3) The elevation profile is reduced. There would be no need for an air inlet space under the packing section or for exhaust fans or ducting mounted above the packing section.
- 4) Air pressure drop can be adjusted by varying the width of the plates in addition to plate length and plate spacing. This would help in the optimization of the final design.

Disadvantages of this design include:

- 1) Reduced theoretical efficiency in heat transfer due to the change from counterflow to crossflow configuration.
- 2) Lack of knowledge of the heat and mass transfer characteristics of the V-trough packing plate under these conditions.
- 3) The need for major modification of the model tower presently in use if this configuration is to be tested.

5.2 Feasibility and Computer Program

In order to provide an estimate of the reduction in heat transfer rate caused by the change to crossflow configuration, a new computer program (listed in Appendix I) has been written. This program follows the analogy developed for the counterflow program (Appendix H, [1]) except for some minor bookkeeping changes.

The crossflow packing is modeled by the program as a grid of series exchangers each of which has a small enough property change to be approximated by a counterflow heat exchanger. The program starts in the upper corner of the grid where both air and water inlet conditions are known. Using the method described in Chapter 3, the outlet conditions of this section are calculated and the results used as inlet conditions for the grid sections below and directly to the side. By moving downward one column at a time, using the previously calculated outlet

conditions from the adjoining grid squares it is possible to calculate the performance of the entire packing section.

The accuracy of this calculation depends on the number of grid squares employed. Trial runs for completely dry packing sections give results within 5% of published [15] solutions when a 25 x 25 grid is used. Further increase in the number of grid squares does give some better agreement, but the calculating time is greatly increased.

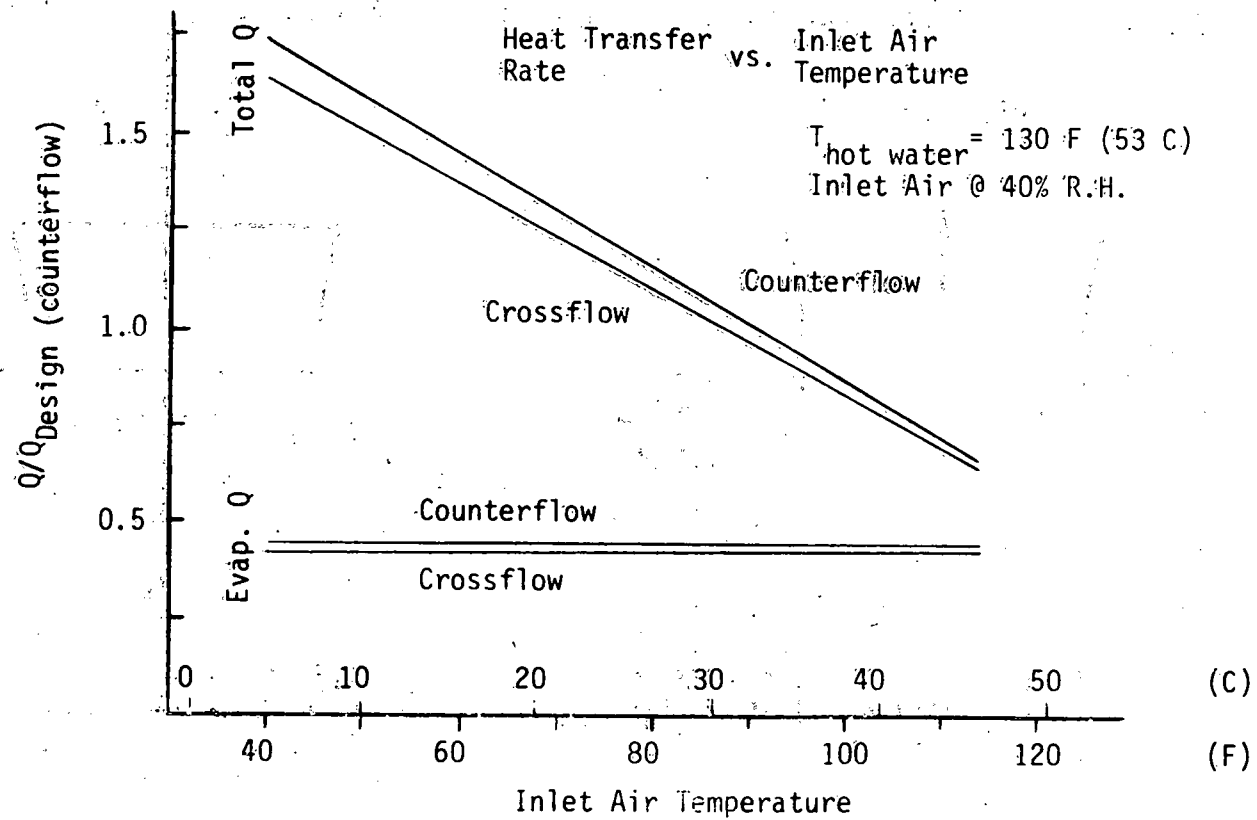
The program also contained provision for mixing the cooling air as it passed through the packing section. However the difference in heat transfer rate between well-mixed and unmixed airflow was found to be very small (5%) for the operating range of an atmospheric tower.

Using the program described above a wet-dry crossflow tower was sized in the same way as the component towers used for the comparison in Chapter 4. This crossflow wet-dry required about 6% more surface area than the comparable counterflow WET/DRY tower where both were sized to carry the same heat load under 90°F (32°C), 40% R.H. conditions.

Performance curves under varying ambient air conditions for the tower designs are quite close and are shown in Figure 5-1.

Although these initial estimates are very rough and were not optimized in any way, they do indicate that the necessary increase in heat transfer surface for the crossflow design is not unreasonable. The savings in design and production costs would well offset the cost of the additional surface area.

Fig. 5-1



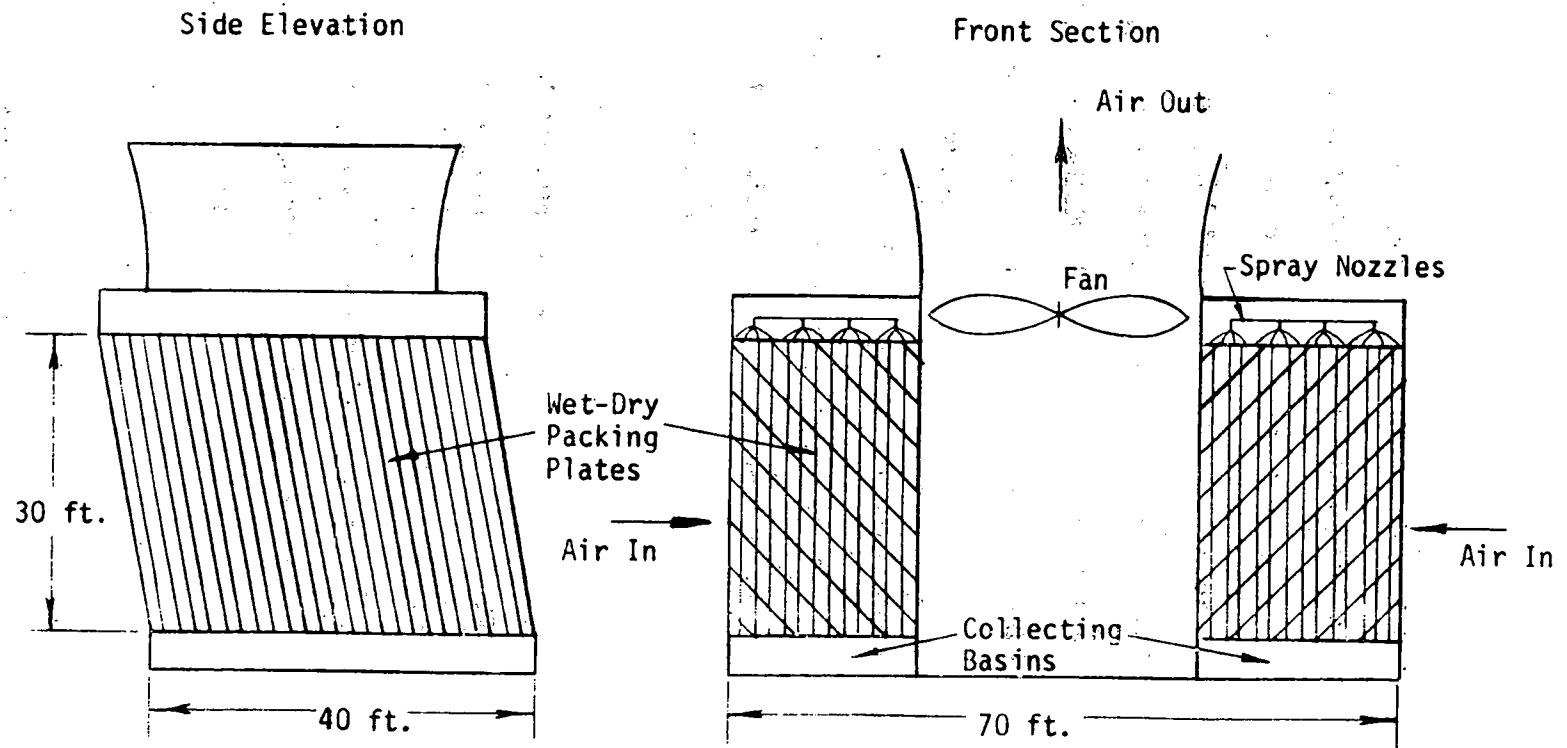
5.3 Visualization of Large Crossflow Towers

The full size crossflow module shown here (Figure 5-2) is meant to give the reader a feel for the general size and shape of such a system. The module is approximately the same size as a mechanical draft wet tower module, with the packing section about 30 ft (9.19 m) high and 20 ft (6.1 m) thick. The air intake area is about 4300 ft² (400 m²) and requires a fan of about 160 hp. Approximately 60 of these modules would be required to cool a 1000 MWe fossil fuel plant.

Part 2 of this report, printed under separate cover, will discuss the construction, instrumentation, testing and analysis of a crossflow model. Also included are project conclusions and recommendations for future investigation.

Fig. 5-2

Visualization of Full-Size Crossflow Module



CHAPTER 6: PRELIMINARY COST COMPARISON

This chapter presents a preliminary cost estimate of the wet-dry cooling tower concept and compares it with estimates for non-evaporative (dry) and evaporative (wet) cooling systems. Due to lack of operating experience with the wet-dry concept this analysis will rely heavily on previous economic comparisons [16] of cooling systems from which the wet-dry costs will be approximated.

Please note that the wet, dry and wet-dry performance models summarized in Sec. 6.2 are not the same as those presented in Appendix A and used in Chapter 4. The dry and wet system design parameters and total costs have been taken directly from the WASH-1360 report [16]. These designs are optimized for a 1000 Mwe fossil fuel plant at a fictional location designated as Middletown, USA. The wet-dry performance module has been designed to match certain parameters (airflow, water loading, exterior dimensions) of the wet system module, roughly optimized for minimum capital costs.

Section 6.1 will describe the economic models developed in Ref. [16] and used here for the calculation of capital and penalty costs for each type of cooling system. Section 6.2 summarizes the design parameters of the wet-dry design and sketches its performance at a few inlet conditions. A step-by-step description of the wet-dry cost evaluation and a short discussion of results will be found in Sec. 6.3 and 6.4, respectively. A more complete discussion of the cost evaluation method and optimization procedure for the dry and wet systems can be found by consulting Ref. [16].

6.1 Economic Model

The cooling system of a power plant determines the heat rejection temperature of the thermodynamic cycle and hence overall efficiency [12]. Cooling system performance is a function of a wide variety of factors including geographic location, ambient weather conditions, cooling system type and capacity, etc. Thus the determination of the true cost of a cooling system must include not only the initial and maintenance costs, but also an assessment of costs based on the performance of the cooling system.

The economic model consists of 1) the capital costs of the cooling system, 2) various economic penalties to account for the effects of changes in ambient conditions and other variables of cooling system and power plant operation, and 3) a total evaluated cost of the cooling system.

6.1.1 Capital Cost of Cooling System

The capital cost of a cooling system includes all expenditures for parts and labor on the system. The major equipment for the systems under consideration here includes condensers, circulating pumps, piping, makeup and blowdown equipment, and the terminal heat sink, e.g. the cooling tower structure. Indirect costs include engineering and contingency charges.

6.1.2 Economic Penalties of Cooling Systems

The method of placing a dollar value on the performance of a

cooling system is to assign economic penalties to this cooling system [16]. Two major penalties are the loss of capacity and the cooling system auxiliary power requirement. In assessing the capacity penalty, the actual capacity of a plant and the corresponding energy produced by the plant are compared to a base capacity and its corresponding energy. Deviations below base values are charged to the cooling system as penalties, whereas deviations above the base values are taken as credits. The auxiliary power requirement is charged to the system according to the costs of the energy consumed.

Six penalties, in capitalized dollars, have been assigned to the economic models. They are as follows [16]:

- P1 = Capacity penalty cost due to highest turbine back pressure
- P2 = Replacement energy cost due to turbine back pressure variation
- P3 = Cost for operating circulating pumps
- P4 = Cost of supplying makeup water to the cooling system
- P5 = Cost for operating the terminal heat sink
- P6 = Cost of operating and maintaining the cooling system

The equation used to evaluate the capacity penalty is:

$$P1 = (K) (\Delta KW_{\max}) \quad (6-1)$$

where: K = Capacity penalty charge rate (\$/kwe)

ΔKW_{\max} = Maximum loss of capacity at the worst ambient condition as compared to base plant capacity (kwe)

This penalty is treated as a capital expenditure and represents the cost of supplementing the capacity loss as compared to the base capacity, perhaps by the addition of gas turbine generating units.

The base capacity discussed here is arbitrarily defined. This capacity was chosen to correspond to an exhaust pressure of 1.5 in. of Hg (5080N/m^2), a common design point for a power plant using a once-through cooling system [16].

A levelized capacity factor of 0.75 is introduced into the maintenance and operating cost equations. This factor assumes the plant will run at maximum power for 75% of the year and otherwise be at zero load. This is believed to be an adequate representation of central station base load power plants which are usually run at full capacity due to their low operating costs.

The replacement energy penalty, P2, and the cooling system auxiliary requirement energy penalties P3 and P5 are evaluated as follows

[16]:

$$P2 = \text{Cap} \left(\frac{1}{\text{afcr}} \right) \int_0^{8760} (R) [\Delta KW(T)] dt \quad (6-2)$$

$$P3 = (K) (HP_w)_{\max} + \text{Cap} \left(\frac{1}{\text{afcr}} \right) \int_0^{8760} (R) [HP_w(T)] dt \quad (6-3)$$

$$P5 = (K) (HP_t)_{\max} + \text{Cap} \left(\frac{1}{\text{afct}} \right) \int_0^{8760} (R) [HP_t(T)] dt \quad (6-4)$$

where:

afcr = Annual fixed charge rate

Cap = Levelized capacity factor

R = Replacement energy charge rate (\$/kw-hr)

KW(T) = Loss of capacity due to variation in ambient conditions
i.e., (plant capacity at ambient conditions) -
(Base plant capacity) (kwe)

$(HP_w)_{\max}$ = Maximum power requirement for pumping cooling water (kwe)

$HP_t(T)$ = Power requirement for operating the terminal heat sink at ambient condition (kwe)

$(HP_t)_{\max}$ = Maximum power requirement for operating the terminal heat sink at the worst ambient condition (kwe)

t = Time (hrs)

Note that P3 and P5 both have two components, i.e., the capital expenditure of additional generating equipment and the capitalized replacement energy costs.

Penalty P4 is the makeup water costs for the system given by:

$$P4 = (G_m) (C_w) (1/afcr) \quad (6-5)$$

where;

G_m = Yearly makeup requirement (gal/yr)

C_w = Cost of makeup water (\$/gal)

The annual operation and maintenance cost P6 of the system is based on the total capital cost and the amount of rotating machinery.

6.2 Performance Model

The cost of heat rejection from a 1000 Mwe fossil fuel generating plant will be considered. Figure 6-1 shows the heat rate correction for this type of plant using low exhaust pressure turbines.

The wet and dry systems compared in the following section (6.3) are taken directly from the WASH-1360 report. These two systems are the results of an extensive cost and performance optimization procedure based on the economic model described in Sec. 6.1. The wet-dry system used in this economic comparison is the result of a very rough optimization procedure which is described below.

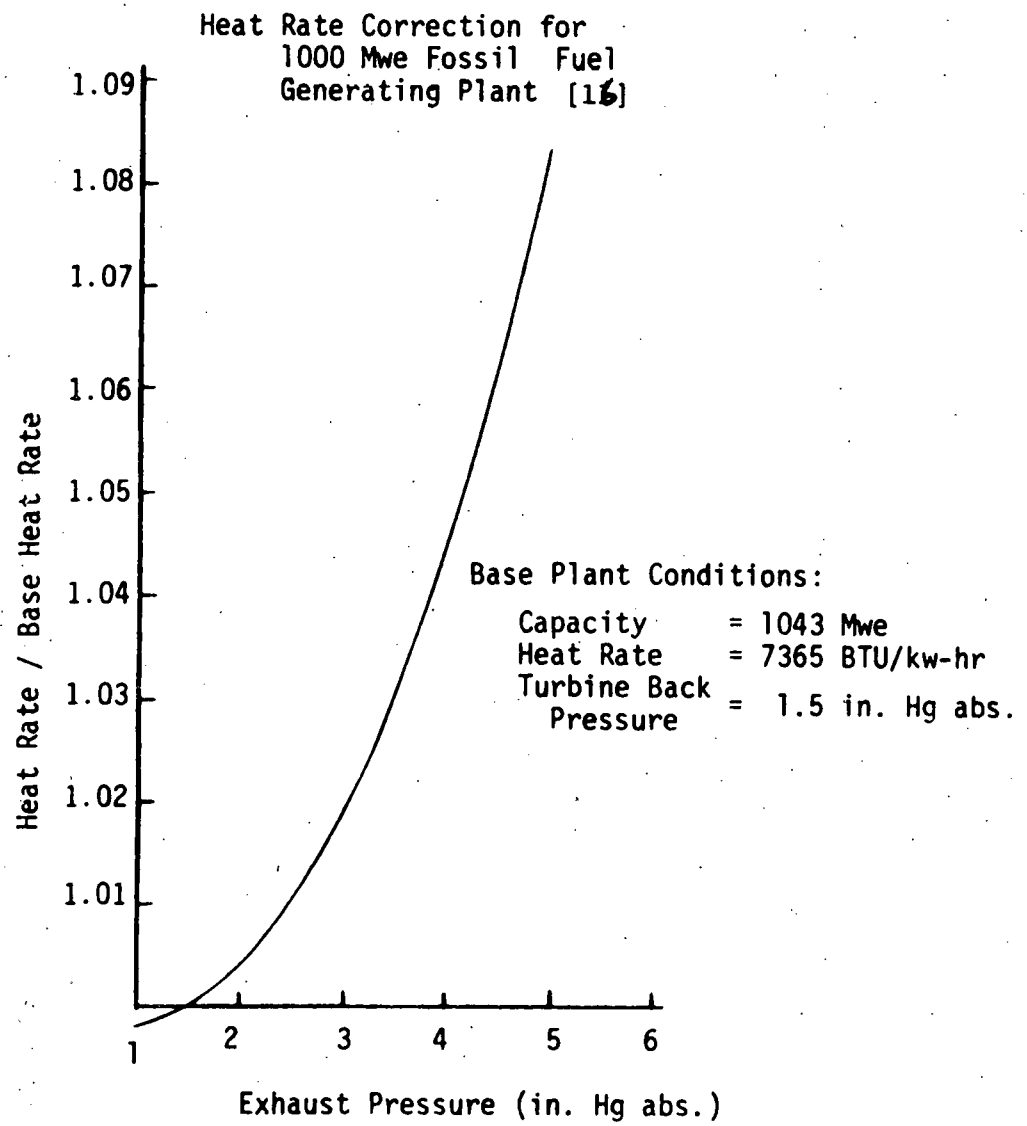
6.2.1 Optimization of the Wet-Dry System

Due to the large number of variable parameters in the wet-dry concept, a few major assumptions have been made about the final design model, and an optimum found for the remaining variable.

These major initial assumptions include:

- 1) Air flow per system module equal to $1.5 \times 10^6 \text{ ft}^3/\text{min}$ ($710 \text{ m}^3/\text{s}$), based on fan limitations.
- 2) Pressure drop of 0.5 in H_2O (125 N/m^2) across packing section, based on fan limitations.
- 3) Individual water channel loading of $1 \text{ lb}_m \text{ H}_2\text{O}/\text{min}$ (0.008 l/s), based on observed model tower performance.
- 4) Packing plate spacing of 1.5 in (3.8 cm), based on consideration of fabrication limitations.

Fig. 6-1



- 5) Plate length in direction of water flow of 30 ft (9.1 m) based on comparably sized wet modules.
- 6) Air and water flow streams in crossflow configuration for design considerations discussed in Chapter 5.

As in the case of both wet and dry systems, the wet-dry cooling system is made up of a number of modules, all identical in construction and performance. The number of modules necessary for a given cooling load is then determined by the heat rejection capacity at a given design temperature. The optimization presented here first determines the remaining design variable, packing surface area per module, then seeks an optimum design point for sizing the final tower system.

Using the above assumptions tower modules with packing widths of 10, 20 and 30 ft were optimized on the basis of minimum capital investment in the terminal heat sink. The cost of each module is broken down into three groups, fans and motors (identical in cost for all designs), structure and piping (here based on data from Ref. [16] and the ground area of individual modules) and packing material cost (based on packing surface area and material and fabrication costs discussed in Sec. 6.3.2). For determining the number of modules for a full size system a design point of 93 F (34 C) ambient air temperature and 121 F (40 C) hot water temperature has been assumed. The following costs in thousands of dollars can now be estimated for each module.

TABLE 6 -1 MODULE COST OPTIMIZATION

| Component Cost in thousands of dollars | Packing Width | | |
|---|---------------|-------|-------|
| | 10 ft | 20 ft | 30 ft |
| Fans and Motors | 19.2 | 19.2 | 19.2 |
| Structure and Piping | 35.7 | 52.8 | 61.0 |
| Packing Material and Fabrication | 37.4 | 106.6 | 194.4 |
| Cost per module | 92.3 | 179.0 | 276.0 |
| Number of modules needed for 1000 MWe facility | 138 | 65 | 46 |
| Total cost of Terminal Heat Sink | 12.7 | 11.6 | 12.7 |
| (millions of dollars) | | | |

As shown in Table 6-1, the 20 ft wide packing section design is the optimum. Next the design inlet hot water temperature was optimized. A higher design inlet hot water temperature would require fewer modules at higher ambient temperatures to reject the same heat load as a system sized to a lower design water temperature. The reduced capital costs would be offset, however, by an increase in the operating penalty costs associated with the resulting higher turbine back pressure. The actual functional relationships are discussed in Sec. 6.1 and 6.3.1 and will not be described here in detail.

The results of this check show that savings in overall system costs are achieved by sizing the wet-dry system for a high inlet hot water temperature. This improvement is limited by the maximum turbine back pressure of 5 in. Hg. ($17,000 \text{ N/m}^2$). Higher inlet water temperatures would require the use of so-called high back pressure turbines. These proposed designs can accommodate back pressures of up to 15 in. Hg. ($51,000 \text{ N/m}^2$) thus allowing a major reduction in the number of necessary modules. Inefficiency penalties are substantial, but in the case of a dry cooling system [16] proved to be the best alternative. For the wet-dry system (Table 6-4) costs were significantly increased.

6.2.2 Wet-Dry Module Design

Table 6-2 shows the results of the wet-dry optimization. The performance of this module design was calculated under a wide range of operating conditions by the computer analogy listed in Appendix I and the results summarized for both high and low back-pressure turbines in Table 6-3.

An estimate of the possible savings which could be produced by relaxing some of the initial design criteria follows. These figures are meant to point out possible directions for future investigation, not as definite savings.

With this in mind, the effect of increased water loading per channel was investigated. By re-shaping the water channel it may be possible to double or triple the water flow over a given plate surface area. Further assuming that this could be done with little or no in-

TABLE 6-2

DESIGN PARAMETERS OF WET-DRY MODULE

| | |
|--------------------------------------|--|
| Configuration | crossflow (Fig. 6-2) |
| Surface Heat Transfer Coefficient | 5.8 BTU/hr-ft ² -F (28 kcal/h-m ² -C) |
| Wet-to-Dry Surface Area Ratio | 0.05 |
| Packing Height | 30 ft (9.1 m) |
| Packing Width | 20 ft (6.1 m) |
| Spacing of Packing Plates | 1.5 in (3.8 cm) |
| Packing Plate Material | galvanized steel |
| Plate thickness | 0.025 in (0.64 mm) |
| Number of plates per module | 620 |
| Water Channels per plate | 80 |
| Air flow per module | 1.5 x 10 ⁶ ft ³ /min (710 m ³ /s) |
| Water flow per module | 6000 gal/min (380 l/s) |
| Pressure Drop across Packing Section | 0.5 in H ₂ O (125 N/m ²) |
| Air Velocity in Packing Section | 10.8 ft/sec (3.3 m/s) |
| Fan Horsepower | 186 hp (140 kw) |
| Number of modules in System: | |
| Low Back-Pressure Design | 59 |
| High Back-Pressure Design | 27 |

FIGURE 6-2 CROSSFLOW PACKING PLATE

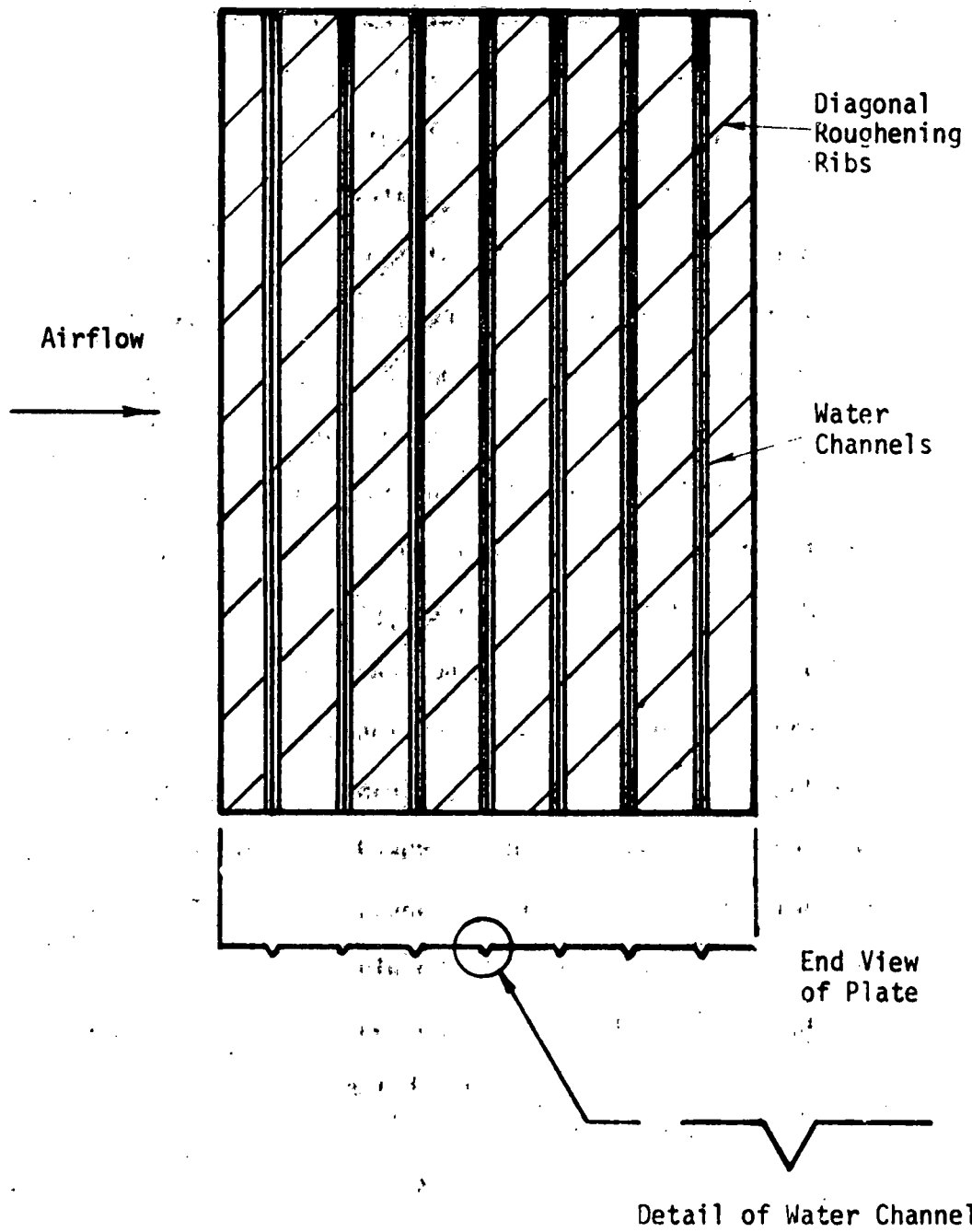


TABLE 6 -3 SUMMARY OF WET-DRY SYSTEMS PERFORMANCE

| Ambient Air Temperature F (C) | Low Back-Pressure Design | | | | High Back-Pressure Design | | | |
|-------------------------------------|-----------------------------------|--|---------------------|--------------------------------|-----------------------------------|--|---------------------|--------------------------------|
| | Inlet Hot Water Temperature | | Capacity Penalty | System Water Consumption | Inlet Hot Water Temperature | | Capacity Penalty | System Water Consumption |
| | F (C) | | kwe | gal/hr (l/s) | F (C) | | kwe | gal/hr (l/s) (x105) |
| 100 (38) | 129 (54) | | 58,500 | 3.18×10^5 (23) | 170 (77) | | 132,000 | 3.28 (24) |
| 90 (32) | 123 (51) | | 36,500 | 2.77×10^5 (20) | 166 (74) | | 116,000 | 3.04 (22) |
| 80 (27) | 118 (48) | | 23,500 | 2.49×10^5 (18) | 162 (72) | | 110,000 | 2.82 (21) |
| 70 (21) | 111 (44) | | 12,500 | 2.10×10^5 (15) | 158 (70) | | 103,000 | 2.59 (19) |
| 60 (16) | 105 (41) | | 6,500 | 1.79×10^5 (13) | 154 (68) | | 96,000 | 2.37 (17) |
| 50 (10) | 98 (37) | | 2,000 | 1.48×10^5 (11) | 149 (65) | | 90,000 | 2.12 (16) |
| 40 (4) | 92 (33) | | 0 | 1.21×10^5 (8.9) | 144 (62) | | 86,000 | 1.87 (14) |
| 30 (-1) | 84 (29) | | -1,500 | 0.94×10^5 (6.9) | 140 (60) | | 83,000 | 1.68 (12) |
| 20 (-7) | 80 (27) | | -2,000 | 0.84×10^5 (6.2) | 135 (57) | | 80,000 | 1.48 (11) |

Yearly total water consumption for Middletown, U.S.A.

Low Back - Pressure Design - 1.39×10^9 gal (3.67×10^8 l)

High Back - Pressure Design - 1.90×10^9 gal (5.01×10^8 l)

crease in the wet-to-dry surface area ratio would produce an increase in the total module design heat transfer rate of 22% for 2X water loading and 32% for 3X water loading. This occurs due to the reduced water side effectiveness thus raising the log mean temperature difference [12].

In cost figures, if similar performance to the original module is assumed, capital cost reductions of 8% and 11% and total evaluated system cost reductions of 5% and 6% will be realized for the doubled and tripled water loading designs, respectively.

Continuing the process, should the plate height and spacing restrictions now be relaxed, the plates lengthened from 30 ft. to 45 ft. (9 to 14 m), plate spacing decreased to 1.25 in (3.2 cm), and normal water loading restored, the module design heat transfer will rise an estimated 20%. This translates into an estimated decrease of 7% in capital cost and 4% in total evaluated cost for the entire system. In this particular design, however, the lengthened plates also increase water consumption for a given amount of total heat transfer. Initial calculations show that this could be a 20% increase in system yearly water consumption. Further study into this type of optimization will be necessary for the final design process.

6.3 Cost Estimation

This step-by-step calculation presents only the values for the low back pressure design. The final figures for the high back pressure design are listed in Table 6-4.

6.3.1 Penalty Costs

Following the procedure described in Sec. 6.1.2,

Capacity Penalty Cost

$$P_1 = (K) (\Delta KW_{\max}) \quad (6-1)$$

$$\Delta KW_{\max} = \Delta KW @ 100F = 58,500 \text{ kwe}$$

$$K = \$150/\text{kwe}$$

$$P_1 = \$8.78 \times 10^6$$

Replacement Energy Cost due to Turbine Back Pressure Variation

$$P_2 = \text{Cap} (1/\text{afcr}) \int_0^{8760} (R) [\Delta KW(T)] dt \quad (6-2)$$

$$\text{Cap} = 0.75$$

$$\text{afcr} = 0.15$$

$$R = \$8.5 \times 10^{-3} / \text{kw-hr}$$

$$\int_0^{8760} \Delta KW(T) dt = 5.19 \times 10^7 \text{ kw-hr}$$

$$P_2 = \$2.21 \times 10^6$$

The integral $\int_0^{8760} \Delta KW(T) dt$ has been evaluated from Table 6-2 and climatic data for Boston, Mass. (Table G-3). This data is comparable to the composite climate of Middletown, U.S.A.

Cost for Operating Circulating Water Pumps

This cost has been set equal to that of the mechanical draft wet tower of Ref. [16].

$$P_3 = \$3.83 \times 10^6$$

This is felt to be accurate due to the similar water flow rate and pumping head requirements of the two designs.

Cost of Makeup Water and Water Treatment for the Cooling System

Due to the 40% lower water consumption of the wet-dry design (see Fig. 4.8), water consumption costs should be likewise reduced.

Thus:

$$P_4 = \$0.07 \times 10^6$$

Cost for Operating the Terminal Heat Sink (fan power)

$$P_5 = (K)(HP_t)_{\max} + \text{Cap} \left(\frac{1}{\text{afcr}} \right) \int_0^{8760} (R) [HP_t(T)] dt \quad (6-4)$$

Assuming that all the module fans run constantly throughout the operating year,

$$(HP_t)_{\max} = HP_t(T) = 11.8 \times 10^3 \text{ hp}$$

Thus:

$$P_5 = \$1.23 \times 10^6 + \$4.06 \times 10^6$$

$$P_5 = \$5.29 \times 10^6$$

Cost of Operating and Maintaining the Cooling System

This cost is based (as described in Sec. 6.1.2) on the amount of rotating machinery in the tower system. Assuming this is roughly proportional to the number of modules in the cooling system

and using the cost of $\$1.21 \times 10^6$ for a wet tower system with 23 modules, and $\$2.61 \times 10^6$ for a dry system with 94 modules, then for a wet-dry system of 59 modules:

$$P_6 = \$1.92 \times 10^6$$

Adding up all of these capitalized penalties gives a total penalty cost of $\$22.1 \times 10^6$. Individual penalties assessed to each system (wet, dry and wet-dry) have been tabulated for comparison in Table 6-3.

6.3.2 Capital Costs

The capital costs of the wet-dry tower system are very dependent on the system design parameters. However, all tower systems will have costs that will be relatively fixed for any given size system (water pumps, piping, electrical work, condensers), costs that will be proportional to the number of tower modules (fans, structure, basins) and costs that will be unique to each design (packing, fin-tube units, etc.).

Using the capital cost breakdown of Ref. [16] for mechanical draft wet and dry systems and the wet-dry module design of Sec. 6.2, the first two groups of these costs have been evaluated for the wet-dry systems. As the major difference between a wet and a wet-dry module is in the packing section, this cost for a wet-dry tower system will be based on the material requirement.

Using $\$0.143$ per ft^2 of 0.025 in. (0.06cm) thick galvanized sheet (a June 1976 U.S Steel quote adjusted for inflation) and the

total cooling system requirement of $21.9 \times 10^6 \text{ ft}^2$ of packing material gives a material cost of $\$3.11 \times 10^6$. One rule of thumb for estimating fabrication costs is to set them roughly equal to material costs for a final cost of $\$6.21 \times 10^6$ for packing plates alone. Adding this to the fixed and proportional costs gives a capital cost of $\$26.6 \times 10^6$ for the low-back pressure wet-dry system.

A more complete breakdown of the capital cost estimation for the wet, dry and wet-dry design is presented in Table 6-4.

TABLE 6-4

SUMMARY OF COST COMPARISONS

(millions of dollars)

(Year of printing 1973)

| Category | Dry Tower | Wet Tower | Wet/Dry Towers | |
|---|---------------|---------------|---------------------------------|--------------------------------|
| | (WASH-1360) | (WASH-1360) | High Back Pressure Design | Low Back Pressure Design |
| Capital Costs | | | | |
| Circulating Water Structure | 0.550 | .710 | .710 | .710 |
| Circulating Water Pumps | 0.670 | 1.030 | 0.471 | 1.030 |
| Concrete Pipe | 1.920 | 1.100 | 1.100 | 1.100 |
| Terminal Heat Sink Basins and Foundations | 10.880 | 2.940 | 5.308 | 11.600 |
| | 0.550 | 1,180 | 0.540 | 1.180 |
| Condensers, Installed | 4.780 | 4.950 | 4.780 | 4.950 |
| Electrical Work | 1.050 | 0.925 | 0.925 | 0.925 |
| Indirect Charges (+25%) | 5.100 | 3.209 | 3.459 | 5.38 |
| Total Capital Investment | <u>25.500</u> | <u>16,044</u> | <u>17.293</u> | <u>26,875</u> |

TABLE 6-4 Part 2

| Category | Dry Tower (WASH-1360) | Wet Tower (WASH-1360) | Wet/Dry Towers High Back Pressure Design Low Back Pressure Design | |
|---|--------------------------|--------------------------|---|--------------|
| Operating Penalty Costs | | | | |
| Capacity Penalty Due to highest turbine Back Pressure | 17.78 | 3.53 | 19.8 | 8.78 |
| Capitalized | | | | |
| Energy Replacement Penalty Due to turbine back pressure variations | 28.32 | 2.46 | 29.1 | 2.21 |
| Capitalized Annual Cost for Operating Cooling Water Pumps | 1.94 | 3.83 | 1.75 | 3.83 |
| Capitalized Cost for Operating Terminal Heat Sink | 6.80 | 2.04 | 2.42 | 5.29 |
| Capitalized Annual Cost for Water Makeup and Treatment | = | 0.18 | 0.07 | 0.07 |
| Capitalized Annual Maintenance Cost | 2.61 | 1.21 | 1.23 | 1.92 |
| Total Penalty Costs | <u>57.45</u> | <u>13.25</u> | <u>54.37</u> | <u>22.10</u> |
| Total Evaluated Cost | 82.95 | 29.29 | 71.66 | 48.70 |

6.4 Results

Referring to Table 6-4, it is now possible to get an estimate of the relative cost of the wet-dry alternative. The high back pressure design has an initial capital cost competitive with the mechanical draft wet tower of the WASH-1360 report. The price for this initial saving is paid, however, in the higher operating penalty costs which approach those of the mechanical draft dry tower. The low back pressure design wet-dry system has a higher initial cost due to the higher number of tower modules, but incurs penalty losses more on the order of the wet tower system, far below those of a dry tower system. On an overall basis, the wet-dry system has higher costs than a wet system, but again consumes only 40% as much water over the course of an operating year.

Using the results of Sec. 6.3 and Ref. [16] it is possible to roughly estimate the costs of the performance models described in Chapter 4. Although these models are not of exactly the same configuration, their behavior will be similar enough under varying operating conditions to permit sizing of each component for a combination tower system.

If the assumption is made of no economies of scale in tower construction (a good assumption for large multi-module systems), the capital costs of each type of cooling system may be compared as follows:

| System (See Chap. 4) | Cost/kwe (generated) Relative | Relative Water Consumption |
|-----------------------------|----------------------------------|-------------------------------|
| Wet System | 29.3 | 1.0 (basis) |
| WET + DRY @ 40° | 53.2 | 0.18 |
| WET + DRY @ 60° | 60.0 | 0.06 |
| DRY | 83.0 | 0.00 |
| WET/DRY + DRY @ 40° | 63.9 | 0.08 |
| WET/DRY + DRY @ 60° | 68.3 | 0.03 |
| WET/DRY (low back pressure) | 48.7 | 0.38 |

Comparable designs for reduced water consumption, WET + DRY @ 60° and WET/DRY @ 40°, show the WET/DRY combination system costing about 7% more than a WET and DRY combination system.

These figures are open to some criticism, which will appear here first. Material and fabrication costs have been rather arbitrarily assigned and may be much different in reality. For example, present day packing costs for a wet-dry tower have been estimated at about \$0.20 per ft² of air-side surface area (\$2.16/m²). A check with dry cooling tower vendors has produced an estimate of \$0.30 to \$0.40 per ft² (\$3.24 to \$4.32/m²) of air-side surface area. These figures do not demonstrate the major surface cost advantage that the wet-dry concept initially promised. Contact with an architect-engineering firm (United Engineers and Constructors, Inc., Philadelphia, PA) has suggested that doubling the packing plate raw material cost to account for fabrication and assembly may be too conservative. Using a suggested multiplier of 1.5 will give a reduction of 1.7×10^{-6} (6%) in the initial capital investment for a wet-dry system.

The treatment of lost capacity when figuring system penalty costs can also be argued. A fixed capital charge for the worst possible loading condition (high temperature, peak power load) may not accurately reflect the true cost of meeting this load. Pumped storage, long distance transmission or specific peaking facilities are possible solutions to the peak load problem that depend on each power company's resources. The availability and cost of this peaking capacity could significantly influence a particular company's choice of system. This is further discussed in Section 6.5.

It may also be possible to operate the wet-dry concept towers in a peaking mode by deluging the packing. This could provide the additional cooling capacity at a minimal increase in cost and cumulative water consumption.

With the above points in mind, these approximate costs are presented with the belief that further analysis and optimization can only reduce them.

6.5 Discussion of Lost Capacity Penalty

The penalty charged to each type of cooling system for the highest operating turbine back pressure will be discussed here. This penalty (designated P1 in section 6.2) compares the capacity of a base generating system operating at 1.5" Hg absolute ($5080 \text{ N/m}^2\text{a}$) turbine back pressure with the heat rate of an identical plant operating at the maximum yearly temperature, rejecting heat with the tower design being evaluated. The difference in the two generating capacities is called the maximum lost capacity and is charged to the tower at a rate representing the cost of replacing this capacity by gas turbine generators.

This charge as a percentage of the total penalty and evaluated costs varies with each type of tower system, wet tower (11% penalty, 12% total), dry tower (30%, 22%) wet-dry (40%, 18%).

Using the figures of Sect. 4.3 and Ref. [16], the following effects can be noted for a 50% reduction in the capacity charge rate:

Sensitivity of Tower Costs to Maximum Capacity

Penalty Rate (Base charge = \$150/kwe)

Reduced Charge \$75/kwe

| Tower | \$ Savings (millions) | % Penalty Cost Reduction | % Total Cost Reduction |
|---------|--------------------------|-----------------------------|---------------------------|
| DRY | 8.89 | 15% | 11% |
| WET | 1.77 | 8% | 6% |
| WET/DRY | 4.39 | 20% | 9% |

The cost of this replacement capacity is a notable factor in the case of the dry and wet-dry towers, and this price may easily vary from

system to system. Since this maximum temperature is encountered for only brief periods in the summer it is very likely that the system may have extra capacity elsewhere within its grid that is provided for scheduled maintenance and for peak periods such as this. The possibilities of pumped storage and inter-company sharing of power also come to mind. Each of these alternatives would differ in cost from gas turbine capacity and the penalty assessed to an individual plants' cooling tower system would vary with the available alternatives.

The applicability of the peak ambient temperature to this analysis can also be questioned. For Middletown (Boston, MA), the hypothetical site of this generating plant, the maximum temperature over the year was taken as 99 F (37C). Using the climatic data of Appendix G, it can be seen that the dry bulb temperature exceeds 90F (32C) for only 0.5% of the years. If the lost capacity is calculated for this temperature range:

Loss of Capacity for 1000 MWe Fossil Fuel Plants

| Tower Type | Dry Bulb Temperatures | | % Change in Lost Capacity | % Change in Total Cost |
|------------|-----------------------|--------|---------------------------|------------------------|
| | 99F | 89F | | |
| DRY | 118,560 | 97,410 | 22% | 5% |
| WET | 23,500 | 17,870 | 31% | 4% |
| WET/DRY | 58,500 | 36,500 | 40% | 7% |

The last column indicates that the wet-dry tower is being more heavily penalized for the last few degrees in maximum ambient temperature. Another way of looking at this is to say that 7% of the total wet-dry cooling system cost is due to conditions found only 0.5% of the time.

This severe high temperature penalty could be avoided by providing a companion wet tower to the wet-dry system for occasional use as the weather demands. Due to the short duration of this peak period, a very attractive proposal is to provide a few of the wet-dry cells with moveable baffles that would defeat the water channeling plates and force the cells to run wet for the necessary length of time. This method would not significantly increase the yearly water consumption total although the instantaneous water consumption would be higher when running in the deluge mode.

The maximum lost capacity penalty charged to each cooling tower type thus seems to be biased toward the wet cooling tower system which experiences very little loss of capacity at high ambient temperatures.

These factors should be taken under consideration when sizing and pricing a new cooling tower system.

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APPENDIX A: RESULTS OF THERMOCOUPLE CALIBRATION TESTS

The following table reports the output differences in microvolts between each thermocouple and a standard when both are placed in the same constant temperature bath. The first test was made using a standard couple which was connected through the switches, but not the plugs and other hardware used with the other thermocouples. The second run, made four days later, used thermocouple #28 as a reference, which was four microvolts higher than the previous standard at the time of the test. The position numbers refer to Figure A-1. For more details of thermocouple installation see Figure 2-2. One microvolt corresponds to about .05°F.

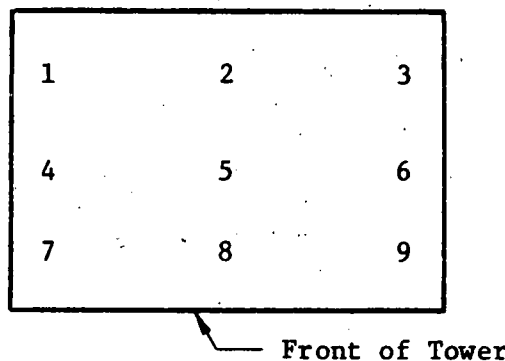


Figure A-1. Tower Cross Section showing the nine positions for Thermocouple Installation.

TABLE A-1: THERMOCOUPLE CALIBRATION RESULTS

| T.C. # | Location | Run #1 | Run #2 | Run #3 |
|--------|----------|--|--|--|
| | | 12/11/75 Temp. \approx 176°F External Std. | 12/15/75 Temp. \approx 176°F Standard: T.C. #28 | 12/17/75 Temp. \approx 104°F External Std. |

Distribution
Pipes

| | | | | |
|---|---|----|----|----|
| 1 | 1 | +3 | -2 | 0 |
| 2 | 2 | +4 | -1 | 0 |
| 3 | 3 | +3 | -5 | +1 |
| 4 | 4 | +4 | -3 | 0 |
| 5 | 5 | +4 | -3 | 0 |
| 6 | 6 | 0 | -2 | 0 |
| 7 | 7 | 0 | -1 | +1 |
| 8 | 8 | -1 | -2 | +1 |
| 9 | 9 | 0 | -1 | 0 |

Air at Top
of Plates

| | | | | |
|----|---|----|----|----|
| 10 | 1 | +3 | 0 | +1 |
| 11 | 2 | +5 | 0 | +4 |
| 12 | 3 | +3 | 0 | +3 |
| 13 | 4 | +2 | -3 | +2 |
| 14 | 5 | +1 | -3 | +1 |
| 15 | 6 | +2 | -2 | 0 |
| 16 | 7 | +2 | -3 | 0 |
| 17 | 8 | +1 | -3 | +1 |
| 18 | 9 | +1 | -3 | +1 |

Water at Top
of Plates

| | | | | |
|----|---|----|---|----|
| 19 | 1 | +3 | 0 | 0 |
| 20 | 2 | +4 | 0 | +3 |

Table A-1 continued

| T.C. # | Location | Run #1 | Run #2 | Run #3 |
|------------------------------|----------|--|--|--|
| | | 12/11/75 Temp. = 176°F External Std. | 12/15/75 Temp. = 176°F Standard: T.C. #28 | 12/17/75 Temp. = 104°F External Std. |
| Water at Top of plates | | | | |
| 21 | 3 | +3 | +1 | +4 |
| 22 | 4 | +1 | -4 | +2 |
| 23 | 5 | +2 | -1 | +1 |
| 24 | 6 | +1 | -2 | +1 |
| 25 | 7 | +3 | -1 | +1 |
| 26 | 8 | +1 | -2 | +1 |
| 27 | 9 | +1 | -3 | +1 |
| Air at Bottom of Plates | | | | |
| 28 | 1 | +3 | - | +1 |
| 29 | 2 | +1 | 0 | -1 |
| 30 | 3 | +1 | +1 | +1 |
| 31 | 4 | +3 | -3 | -1 |
| 32 | 5 | 0 | -1 | -1 |
| 33 | 6 | +1 | -4 | 0 |
| 34 | 7 | +2 | +1 | +2 |
| 35 | 8 | +4 | -3 | 0 |
| 36 | 9 | +3 | -1 | -2 |
| Water at Bottom of Plates | | | | |
| 37 | 1 | 0 | -1 | 0 |
| 38 | 2 | +2 | -2 | 0 |
| 39 | 3 | +1 | 0 | +1 |
| 40 | 4 | +4 | -1 | -2 |
| 41 | 5 | +1 | +1 | -2 |
| 42 | 6 | 0 | -3 | -1 |

Table A-1 continued

| T.C. # | Location | Run #1 | Run #2 | Run #3 |
|------------------------------|----------|--|---|--|
| | | 12/11/75 Temp. \approx 176°F External Std. | 12/15/75 Temp. \approx 176°F Standard:T.C. #28 | 12/17/75 Temp. \approx 104°F External Std. |
| Water at Bottom of Plates | | | | |
| 43 | 7 | +4 | -2 | +2 |
| 44 | 8 | +4 | -1 | +1 |
| 45 | 9 | 0 | -3 | 0 |
| Collection Channels | | | | |
| 46 | 3 | -2 | -2 | -1 |
| 47 | 6 | +1 | -1 | 0 |
| 48 | 9 | +2 | 0 | 0 |
| Top Rake | | | | |
| 49 | 1,4,7 | -2 | -1 | +1 |
| 50 | 2,5,8 | -1 | -1 | +1 |
| 51 | 3,6,9 | -1 | -1 | +1 |
| Bottom Rake | | | | |
| 52 | 1,4,7 | -2 | -1 | +1 |
| 53 | 2,5,8 | -1 | no data | no data |
| 54 | 3,6,9 | 0 | -2 | +1 |

APPENDIX B: SAMPLE CALCULATIONS

B.1 Error Analysis for the Heat Balance

From Chapter 2 the discrepancy in the heat balance is given by:

$$\begin{aligned} \delta \dot{Q} = & \dot{m}_{\text{air}} (h_{\text{air}_{\text{in}}} + w_{\text{in}} h_{\text{vapor}_{\text{in}}} - h_{\text{air}_{\text{out}}} - w_{\text{out}} h_{\text{vapor}_{\text{out}}}) \\ & + \dot{m}_{\text{water}} c_{p_{\text{water}}} (T_{\text{water}_{\text{in}}} - T_o) - \\ & c_{p_{\text{water}}} (\dot{m}_{\text{water}} - \dot{m}_{\text{air}} (w_{\text{out}} - w_{\text{in}})) (T_{\text{water}_{\text{out}}} - T_o) \quad (\text{B-1}) \end{aligned}$$

The uncertainty in $\delta \dot{Q}$ is given by:

$$u_{\delta \dot{Q}} = \left[\left(\frac{\partial \delta \dot{Q}}{\partial x_1} u_1 \right)^2 + \left(\frac{\partial \delta \dot{Q}}{\partial x_2} u_2 \right)^2 + \dots + \left(\frac{\partial \delta \dot{Q}}{\partial x_n} u_n \right)^2 \right]^{1/2} \quad (\text{B-2})$$

where x_1, x_2, \dots, x_n are the following ten variables:

$\dot{m}_{\text{air}}, h_{\text{vapor}_{\text{in}}}, h_{\text{vapor}_{\text{out}}}, h_{\text{air}_{\text{in}}}, h_{\text{air}_{\text{out}}}, w_{\text{in}}, w_{\text{out}}, \dot{m}_{\text{water}}, T_{\text{water}_{\text{in}}},$

$T_{\text{water}_{\text{out}}}$. u_1, u_2, \dots, u_n are the respective uncertainties of these variables. The calculation will be shown for a typical set of conditions:

$$\dot{m}_{\text{air}} = 150 \text{ lb}_m/\text{min}$$

$$\dot{m}_{\text{water}} = 170 \text{ lb}_m/\text{min}$$

$$\text{Air Temperature in} = 90^\circ\text{F}$$

$$\text{Dew Point Temperature in} = 50^\circ$$

$$\text{Air Temperature Out} = 100^{\circ}\text{F}$$

$$\text{Dew Point Temperature Out} = 55^{\circ}\text{F}$$

$$T_{\text{water in}} = 120^{\circ}\text{F}$$

$$T_{\text{water out}} = 116.4^{\circ}\text{F}$$

The following values may be obtained from air property tables using the air temperatures and the dew point temperatures.

$$h_{\text{air in}} = 13.938 \text{ BTU/lb}$$

$$h_{\text{air out}} = 16.341 \text{ BTU/lb}$$

$$h_{\text{vapor in}} = 1100.5 \text{ BTU/lb}$$

$$h_{\text{vapor out}} = 1104.8 \text{ BTU/lb}$$

$$w_{\text{in}} = .007655 \text{ lbm/lbm}$$

$$w_{\text{out}} = .009225 \text{ lbm/lbm}$$

The following uncertainties are determined directly by the instrument limitations:

$$u_{\text{m air}} = 8 \text{ lbm/min}$$

$$u_{\text{m water}} = 1.7 \text{ lbm/min}$$

$$u_{\text{Air Temp. in}} = .1^{\circ}\text{F}$$

$$u_{\text{Dew Point in}} = .3^{\circ}\text{F}$$

$$u_{\text{Air Temp. out}} = .1^{\circ}\text{F}$$

$$u_{\text{Dew Point out}} = .3^{\circ}\text{F}$$

$$u_{T_{\text{water in}}} = .1^{\circ}\text{F}$$

$$u_{T_{\text{water out}}} = .1^{\circ}\text{F}$$

The uncertainties in the enthalpies and absolute humidities are determined by linear interpolation using the air property tables:

$$u_{h_{\text{air in}}} = .024 \text{ BTU/Lbm}$$

$$u_{h_{\text{air out}}} = .024 \text{ BTU/Lbm}$$

$$u_{h_{\text{vapor in}}} = .045 \text{ BTU/Lbm}$$

$$u_{h_{\text{vapor out}}} = .045 \text{ BTU/Lbm}$$

$$u_{w_{\text{in}}} = .000087 \text{ Lbm/Lbm}$$

$$u_{w_{\text{out}}} = .000102 \text{ Lbm/Lbm}$$

The partial differential are given in Table 11.

Substituting the appropriate values into equation B-2 results in an uncertainty:

$$u_{\dot{Q}} = 45.87 \text{ BTU/min}$$

For the operating conditions given, 625.6 BTU/min is calculated using equation 2-4. The uncertainty therefore results in a 7.3% error.

TABLE B-1 PARTIAL DIFFERENTIALS FOR ERROR ANALYSIS

$$\frac{\partial \dot{Q}}{\partial \dot{m}_{\text{air}}} = (h_{\text{air, in}} + w_{\text{in}} h_{\text{vapor}} - h_{\text{air, out}} - w_{\text{out}} h_{\text{vapor, out}}) + (w_{\text{out}} - w_{\text{in}}) (T_{\text{water, out}} - T_o) c_{p, \text{water}} = -4.038$$

$$\frac{\partial \dot{Q}}{\partial h_{\text{vapor, in}}} = w_{\text{in}} \dot{m}_{\text{air}} = 1.148$$

$$\frac{\partial \dot{Q}}{\partial h_{\text{vapor, out}}} = -w_{\text{out}} \dot{m}_{\text{air}} = -1.384$$

$$\frac{\partial \dot{Q}}{\partial h_{\text{air, in}}} = \dot{m}_{\text{air}} = 150$$

$$\frac{\partial \dot{Q}}{\partial h_{\text{air, out}}} = -\dot{m}_{\text{air}} = -150$$

$$\frac{\partial \dot{Q}}{\partial w_{\text{in}}} = \dot{m}_{\text{air}} h_{\text{vapor, in}} - \dot{m}_{\text{air}} (T_{\text{water, out}} - T_o) c_{p, \text{water}} = 152415$$

$$\frac{\partial \dot{Q}}{\partial w_{\text{out}}} = \dot{m}_{\text{air}} h_{\text{vapor, out}} + \dot{m}_{\text{air}} (T_{\text{water, out}} - T_o) c_{p, \text{water}} = -153060.$$

$$\frac{\partial \dot{Q}}{\partial \dot{m}_{\text{water}}} = c_p (T_{\text{water, in}} - T_{\text{water, out}}) = 3.6$$

$$\frac{\partial \dot{Q}}{\partial T_{\text{water}}} = \dot{m}_{\text{water}} c_{p, \text{water}} = 170$$

$$\frac{\partial \dot{Q}}{\partial T_{\text{water, out}}} = -c_p (\dot{m}_{\text{water}} - \dot{m}_{\text{air}} (w_{\text{out}} - w_{\text{in}})) = -169.76$$

B.2 Energy balance for Experimental Data

The heat transfer to the air is given by:

$$\dot{Q}_{\text{air}} = \dot{m}_{\text{air}} (h_{\text{air}_{\text{out}}} + w_{\text{out}} h_{\text{vapor}_{\text{out}}} - h_{\text{air}_{\text{in}}} - w_{\text{in}} h_{\text{vapor}_{\text{in}}}) \quad (\text{B-3})$$

The heat transfer from the water is given by:

$$\begin{aligned} \dot{Q}_{\text{water}} = & \dot{m}_{\text{water}} c_{p_{\text{water}}} (T_{\text{water}_{\text{in}}} - T_o) - \\ & c_{p_{\text{water}}} (\dot{m}_{\text{water}} - \dot{m}_{\text{air}} (w_{\text{out}} - w_{\text{in}})) (T_{\text{water}_{\text{out}}} - T_o) \end{aligned} \quad (\text{B-4})$$

As an example the data from the first experimental run will be used.

$$\dot{m}_{\text{air}} = 143.4 \text{ lbm/min}$$

$$\text{Air Temperature in} = 88.4^{\circ}\text{F}$$

$$\text{Air Temperature out} = 97.9^{\circ}\text{F}$$

$$\text{Dew Point Temperature in} = 32.8^{\circ}\text{F}$$

$$\text{Dew Point Temperature out} = 41.5^{\circ}\text{F}$$

$$w_{\text{in}} = .003912 \text{ lbm/lbm}$$

$$w_{\text{out}} = .005528 \text{ lbm/lbm}$$

$$h_{\text{air}_{\text{in}}} = 13.5526 \text{ BTU/lbm}$$

$$h_{\text{air}_{\text{out}}} = 15.8360 \text{ BTU/lbm}$$

$$h_{\text{vapor}_{\text{in}}} = 1099.76 \text{ BTU/lbm}$$

$$h_{\text{vapor}_{\text{out}}} = 1103.86 \text{ BTU/lbm}$$

Substituting these values into equation B-3 and B-4.

$$\dot{Q}_{\text{air}} = 585.5 \text{ BTU/min}$$

$$\dot{Q}_{\text{water}} = 670.62$$

$$\dot{Q}_{\text{average}} = 628.1$$

$$\text{percent error} = \frac{\dot{Q}_{\text{water}} - \dot{Q}_{\text{air}}}{\dot{Q}_{\text{average}}} (100) = 13.5\%$$

APPENDIX C

RAW DATA

Table C-1 contains the raw inputs to the model tower and the computer analogy. The measured response of the model is also compared with the computer prediction. The first 5 runs were done with the unpainted packing plates and the last 4 (after 5/5) used the painted plates (see Section 2.4 and App.E).

Table C-1

| Date of Run | 3/19 | | 3/22 | | 3/23 AM | |
|--|--------|----------|--------|----------|---------|----------|
| | Data | Computer | Data | Computer | Data | Computer |
| Air Flow Rate (lbm/min) | | 136 | | 110 | | 110 |
| Inlet Water Flow Rate (lbm/min) | | 152 | | 152 | | 192 |
| Inlet Air Temperature (F) | | 87.8 | | 84.2 | | 86.6 |
| Inlet Air Humidity (lbm/lbm) | | .005065 | | .002212 | | .002373 |
| Inlet Water Temperature (F) | | 131.5 | | 120.9 | | 124.7 |
| Outlet Air Temperature (F) | 99.9 | 101.7 | 94.5 | 95.9 | 97.4 | 98.9 |
| Outlet Air Humidity (lbm/lbm) | .00870 | .00920 | .00521 | .00541 | .00619 | .00598 |
| Outlet Water Temperature (F) | 125.5 | 126.0 | 116.3 | 116.5 | 120.9 | 120.9 |
| Air Temperature Change (F) | 12.1 | 13.9 | 10.3 | 11.7 | 10.8 | 12.25 |
| Water Temperature Change (F) | 6.0 | 5.5 | 4.6 | 4.4 | 3.8 | 3.8 |
| Air Humidity Change (lbm/lbm) | .00364 | .00414 | .00300 | .00320 | .00382 | .00361 |
| Air Side of Energy Balance (Btu/min) | 738 | - | 638 | - | 750 | - |
| Water Side of Energy Balance (Btu/min) | 938 | - | 727 | - | 767 | - |
| Average Heat Transfer Rate (Btu/min) | 839 | 875 | 677 | 700 | 759 | 766 |
| Percent Error in Energy Balance | 24 | - | 13 | - | 2 | - |
| Evaporative Heat Transfer (Btu/min) | 405 | 460 | 336 | 358 | 427 | 403 |
| Percent Evaporative Heat Transfer | 48 | 53 | 50 | 51 | 56 | 53 |

Table C-1 con't

| Date of Run | 3/23 PM | | 3/25 | | 5/6 | |
|--|---------|----------|--------|----------|--------|----------|
| | Data | Computer | Data | Computer | Data | Computer |
| Air Flow Rate (lbm/min) | | 110 | | 109 | | 105 |
| Inlet Water Flow Rate (lbm/min) | | 190 | | 126 | | 170 |
| Inlet Air Temperature (F) | | 91.7 | | 89.8 | | 90.8 |
| Inlet Air Humidity (lbm/lbm) | | .002613 | | .005400 | | .009363 |
| Inlet Water Temperature (F) | | 135.0 | | 134.3 | | 133.1 |
| Outlet Air Temperature (F) | 103.0 | 105.2 | 102.6 | 103.7 | 101.1 | 104.3 |
| Outlet Air Humidity (lbm/lbm) | .00703 | .00736 | .00961 | .00978 | .01200 | .01200 |
| Outlet Water Temperature (F) | 129.9 | 130.3 | 127.4 | 127.5 | 129.9 | 129.4 |
| Air Temperature Change (F) | 11.3 | 14.1 | 12.8 | 13.9 | 10.2 | 13.5 |
| Water Temperature Change (F) | 5.1 | 4.8 | 6.9 | 6.8 | 3.2 | 3.7 |
| Air Humidity Change (lbm/lbm) | .00441 | .00475 | .00421 | .00438 | .00263 | .00263 |
| Air Side of Energy Balance (Btu/min) | 853 | - | 844 | - | 518 | - |
| Water Side of Energy Balance (Btu/min) | 1017 | - | 909 | - | 571 | - |
| Average Heat Transfer Rate (Btu/min) | 935 | 955 | 877 | 899 | 544 | 654 |
| Percent Error in Energy Balance | 18 | - | 7 | - | 10 | - |
| Evaporative Heat Transfer (Btu/min) | 490 | 527 | 463 | 482 | 279 | 278 |
| Percent Evaporative Heat Transfer | 52 | 55 | 55 | 54 | 51 | 43 |

Table C-1 con't

| Date of Run | 5/11 | | 5/25 | | 6/24 | |
|--|--------|----------|--------|----------|--------|----------|
| | Data | Computer | Data | Computer | Data | Computer |
| Air Flow Rate (lbm/min) | | 111 | | 111 | | 91 |
| Inlet Water Flow Rate (lbm/min) | | 171 | | 95 | | 164 |
| Inlet Air Temperature (F) | | 89.2 | | 85.5 | | 95.9 |
| Inlet Air Humidity (lbm/lbm) | | .007512 | | .006241 | | .01946 |
| Inlet Water Temperature (F) | | 142.3 | | 138.3 | | 124.3 |
| Outlet Air Temperature (F) | 101.3 | 106.0 | 96.9 | 101.6 | 104.8 | 105.2 |
| Outlet Air Humidity (lbm/lbm) | .01146 | .01085 | .00909 | .00899 | .02156 | .02131 |
| Outlet Water Temperature (F) | 138.5 | 137.5 | 130.5 | 130.5 | 122.3 | 122.0 |
| Air Temperature Change (F) | 12.1 | 16.8 | 11.4 | 16.3 | 8.9 | 9.3 |
| Water Temperature Change (F) | 3.8 | 4.8 | 7.8 | 7.8 | 2.0 | 2.3 |
| Air Humidity Change (lbm/lbm) | .00395 | .00334 | .00285 | .00275 | .00210 | .00185 |
| Air Side of Energy Balance (Btu/min) | 812 | - | 653 | - | 413 | - |
| Water Side of Energy Balance (Btu/min) | 697 | - | 749 | - | 346 | - |
| Average Heat Transfer Rate (Btu/min) | 755 | 866 | 701 | 771 | 379 | 399 |
| Percent Error in Energy Balance | 15 | - | 14 | - | 17 | - |
| Evaporative Heat Transfer (Btu/min) | 440 | 372 | 316 | 305 | 195 | 171 |
| Percent Evaporative Heat Transfer | 58 | 43 | 45 | 40 | 51 | 43 |

APPENDIX D

DESIGN FACTORS OF COMPARISON TOWERS

To provide some basis for the comparison of the advanced wet/dry packing section to the types of heat transfer surfaces now used by power generating companies, three types of towers have been modeled which meet the same standards of heat rejection and water temperature drop at a design inlet air temperature.

These three towers are: 1) Completely non-evaporative (DRY) where the water to be cooled is circulated through a closed system of finned tubes and the heat transferred by forced convection. 2) Evaporative (WET) where the hot water is distributed over a film-type packing to maximize the air-to-water surface area. Approximately 85% of the heat transfer in this tower is due to evaporation with the remaining heat transfer coming from convection at the water surface. 3) Wet/dry (WET/DRY) with the V-trough type packing now being tested at M.I.T. This design seeks to minimize the air-water contact area thus cutting evaporation while still allowing convective heat transfer by the fin effect of the unwetted plate surface.

The inlet conditions for each of the models was then varied from 10°F (-12°C) to 120°F (49°C) and their performance calculated by means of the computer program. This program is essentially the same as listed in [1] with a few additions determined by experimentation and design considerations.

No effort has been made to optimize any one model, rather the goal was to give each an equal advantage (or handicap) in order to make

a fair comparison. Thus the results can be extended, at least qualitatively, to give an indication of more efficient designs.

The following discussion will review the design parameters and summarize the final tower configurations.

Inlet Water Temperature: 130°F (54°C), corresponding to a turbine back pressure of 3.5 in Hg absolute (8.9 cm Hg).

Water Temperature Drop: 20°F (11°C) at the design point.

Design Point: inlet air at 90°F (32°C) dry-bulb and 40% R.H, giving an ITD of 40°F (22°C) and an approach temperature of 20°F (11°C) to the dry-bulb and 39°F (22°C) to the wet-bulb.

Packing Configuration: DRY tower, no exact fin arrangement was assumed in this case as the values for the surface heat transfer coefficient and the fin efficiency were arbitrarily fixed in this analysis (see Table D-1). For this reason, while the total surface area of the DRY model is inaccurate for a counterflow design, the surface area per plan area value is subjective and therefore not included in the summary table. WET/DRY tower, this was chosen to be as close to the heat transfer model tested at M.I.T. Each packing plate is made of galvanized sheet steel 0.0233 in (.059 cm) thick. The V-troughs are one inch (2.5 cm) on a side and are bent to an angle of 60° at the bottom of the V. The wet-to-dry surface area ratio was set at 5%, close to the experimentally measured value. The plates were spaced 1.5 in (3.8 cm) center-to-center and no evaporation was assumed in

the distribution and collection processes. WET tower, this was modeled as a film-type packing with all heat transfer done at the water surface. The packing plates were considered to be flat and evenly coated with water. Plate spacing was increased to 4 in. (10 cm) and the plate angle to the vertical was changed from 10° to 45° .

Heat and Mass Transfer Coefficients: DRY tower, the surface convective heat transfer coefficient and fin efficiency were set at $8 \text{ BTU/hr-ft}^2\text{ }^\circ\text{F}$ ($39 \text{ kcal/h-m}^2\text{ }^\circ\text{C}$) and 0.8, respectively. These figures did not change with operating conditions. WET/DRY tower, the dry surface convective heat transfer coefficient was also set at $8 \text{ BTU/hr-ft}^2\text{ }^\circ\text{F}$ ($39 \text{ kcal/h-m}^2\text{ }^\circ\text{F}$) the reasoning begin that augmentation of the surface could produce a higher convective coefficient than was measured on the simple experimental packing plates. The water surface convective heat transfer coefficient was calculated from the Dittus-Boelter (see Section 2.1) equation and modified on the basis of experimental evidence [3-2]. The average value was approximately $5.3 \text{ BTU/hr-ft}^2\text{ }^\circ\text{F}$ ($26 \text{ kcal/h-m}^2\text{ }^\circ\text{C}$). The mass transfer coefficient was calculated by the Chilton-Colburn analogy and had an average value of 370 ft/hr (113m/h) (Section 2.1). WET tower, the coefficients for this model were calculated as above and had average values of $4.4 \text{ BTU/hr-ft}^2\text{ }^\circ\text{F}$ ($21 \text{ kcal/h-m}^2\text{ }^\circ\text{C}$) and 300 ft/hr (90 m/h) for the respective heat and mass transfer coefficients.

Water Flow per Unit Area - DRY tower, not relevant due to the fixed values of heat transfer surface area per unit plan area. WET/DRY tower, approximately 4 gpm/plan ft² (2.7ℓ/s/plan m²). This value was within the range of the model tower tests. WET tower, approximately 6 gpm/plan ft² (4.1ℓ/s/plan m²).

Air Flow Ratios - DRY tower, to give a temperature effectiveness of 0.7 where effectiveness is defined as:

$$\epsilon = \frac{T_{C2} - T_{C1}}{T_{h1} - T_{c1}}$$

This produced a mass flow ratio of $\dot{M}_{air}/\dot{M}_{water} = 3.0$. WET/DRY tower, to give an air temperature rise approximately equal to that of the DRY model, $\dot{M}_{air}/\dot{M}_{water} = 1.5$. This was fairly close to the mass flow ratios used with the experimental model tower. WET tower, the mass flow ratio was set here at $\dot{M}_{air}/\dot{M}_{water} = 1$, a widely used value.

The following table summarized the three computer models used in the performance comparisons. Wherever possible parameters have been non-dimensionalized or related to heat load so that the results could be extended for any desired heat rejection rate.

Table D-1

Summary of Design Parameters of Tower Models

| Tower type | DRY | WET/DRY | WET |
|---|------|---------|------|
| Total Surface Area | | | |
| ft ² /1000 Btu/hr | 9.8 | 7.1 | 0.4 |
| m ² /1000 kcal/h | 3.6 | 2.6 | 0.2 |
| Mass Flow Rate of Air @ Design Point* | | | |
| lb _m /1000 Btu | 150 | 75 | 47 |
| kg/1000 kcal | 270 | 135 | 85 |
| Mass Flow Rate of Water @ Design Point* | | | |
| lb _m /1000 Btu | 6.0 | 5.7 | 5.6 |
| kg/1000 kcal | 1.5 | 1.4 | 1.4 |
| Specific Heat Ratio | | | |
| $\dot{M}_{air} C_{p,air} / \dot{M}_{water} C_{p,water}$ | 0.72 | 0.38 | 0.24 |
| Average Fin Efficiency | 0.8 | 0.7 | - |
| Average Dry Plate Surface Heat Transfer Coefficient | | | |
| Btu/hr-ft ² -F | 8.0 | 8.0 | - |
| kcal/h-m ² -C | 39. | 39. | - |
| Average Wet Surface Heat Transfer Coefficient | | | |
| Btu/hr-ft ² -F | - | 5.3 | 4.4 |
| kcal/h-m ² -C | - | 26. | 21. |
| Average Water Surface Mass Transfer Coefficient | | | |
| ft/hr | - | 370 | 300 |
| m/h | - | 113 | 91 |

*Design Point, all Towers

Inlet Air 90 F (32 C) Dry Bulb, 40% R.H.

Inlet Water 130 F (54 C)

 ΔT_{water} 20 F (11 C)

APPENDIX E

CORROSION AND PAINTING OF PACKING PLATES

After several data runs had been completed with the model tower a white, crusty deposit was noticed in the bottom of the V-troughs. Visual observations showed that when water was present in the bottom of the troughs it tended to climb up the deposit layer in the same way water would soak up into a blotter.

The "blotter effect" increased the wet-to-dry surface area ratio and raised the portion of heat transferred by evaporation. Attempts to scrub away the scaling were only temporarily successful as the deposits returned very quickly.

Close examination of the packing plates revealed that the protective galvanizing had been worn away in some places and the reaction of the steel plate with the city water was contributing to the scaling.

To prevent this corrosion and to help keep the wet-to-dry surface area ratio low, one packing plate was coated with an acrylic spray paint as a test. Several combinations of primer and topcoat were compared and the one which seemed most resistant to deposit buildup was used. The plates were cleaned by scrubbing down the water side with citric acid and steel wool, then "aged" by rubbing down with acetic acid. The primer was standard for galvanized material, two coats of zinc chromate primer. This was followed by two spray coats of Crylon^R flat back.

The surface was fairly smooth to the touch and was not wetted by the water in the troughs. This tendency appeared to decrease after several hours of running but little or no deposit buildup was observed.

Micrometer measurements showed that the thickness of the paint layer was approximately .001 in (2.5×10^{-3} mm). Assuming the paint to have a thermal conductivity equal to that of hard rubber, 6.09 BTU/hr-ft²°F (0.13 kcal/h-m°C), calculations showed a change in the surface convective heat transfer coefficient from 3.50 to 3.49 BTU/hr-ft²°F (17.09 to 17.04 kcal/h-m²°C), an insignificant change.

Only the side of the plates wetted by the water streams was painted as the dry side showed no corrosion or wear.

After approximately 48 running hours since painting very little deposition can be seen. What there is wipes off easily and does not seem to affect the wet-to-dry surface area ratio.

APPENDIX F

FLOW VISUALIZATION

The flow visualization test apparatus used in the initial design of the V-trough packing plates was used again to provide qualitative information on the relationship of vertical angle to the water channeling characteristics.

The angle presently used in the model tower is 10° from the vertical. After the plate surface had been cleaned and painted the water flowing down the plates was observed to wander at the top and in some cases did not settle in the bottom of the trough until well down the plate. These tendencies diminished somewhat as total running time increased, but it was felt that observations of the channeling properties of a painted packing plate should be performed.

Using a small test plate whose surface had been treated similarly to those in the model tower, qualitative observations were taken for plate angles from 10° to 45° from the vertical.

At 45° water spread over the top portion of the plate channeled itself immediately into the bottom of the troughs and stayed channeled down the plate with no wandering. The water channel was approximately 0.125 in wide (0.32 cm) for the apparatus flowrate.

At 10° the water channeled itself poorly by comparison, in some cases jumping from one trough to another and in most troughs requiring several inches before becoming channeled in the trough bottoms. Once channeled the stream tended to stay that way, although several troughs

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At 10° the water channeled itself poorly by comparison, in some cases jumping from one trough to another and in most troughs requiring several inches before becoming channeled in the trough bottoms. Once channeled the stream tended to stay that way, although several troughs

did exhibit a sort of meandering flow up and down the sides of the V-trough. The water surface, once channeled, was about .063 inches (0.159 cm), or about half that at 45°.

The compromise angle was felt to be about 30° from the vertical. At this angle the water was channeled much better than at 10° or 20° and the water surface was narrower than at either 45° or 40°.

The resulting recommendation is based on simple, qualitative observations and is meant to provide a starting point for further work. The flow visualization apparatus will have to be modified to allow more realistic flow rates and distribution if quantitative data are desired.

APPENDIX G

CLIMATIC DATA

Climatic Data from four representative cities were summarized from Reference [14]. The summaries show the number of hours each month that the ambient dry-bulb temperature was within each of a range of temperature spans. These spans are:

| | |
|-------------|------------|
| 111 - 110 F | 48 - 43 C |
| 109 - 100 F | 43 - 38 C |
| 99 - 90 F | 37 - 32 C |
| 89 - 80 F | 32 - 27 C |
| 79 - 70 F | 26 - 21 C |
| 69 - 60 F | 21 - 16 C |
| 59 - 50 F | 15 - 10 C |
| 49 - 40 F | 9 - 4 C |
| 39 - 30 F | 4 - 1 C |
| 29 - 20 F | -2 - -7 C |
| 19 - 10 F | -7 --12 C |
| 9 - 0 F | -13 --18 C |
| -1 - -10 F | -18 --23 C |
| -11 - -20 F | -24 --29 C |

Also included for each month is the total number of hours in the month and the average relative humidity.

Table G-1

Albuquerque, N.M.

| | | | | | |
|-------|----------|---------|---------|----------|--------|
| Jan. | 7440 hrs | 50% RH | Feb. | 6792 hrs | 30% RH |
| 69/60 | 45 | | 69/60 | 298 | |
| 59/50 | 722 | | 59/50 | 1013 | |
| 49/40 | 2475 | | 49/40 | 2089 | |
| 39/30 | 2938 | | 39/30 | 2206 | |
| 29/20 | 1364 | | 29/20 | 924 | |
| 19/10 | 178 | | 19/10 | 230 | |
| 9/0 | 18 | | 9/0 | 22 | |
| Mar. | 7440 hrs | 30% RH | Apr. | 6792 hrs | 30% RH |
| 79/70 | 120 | | 89/80 | 99 | |
| 69/60 | 097 | | 79/70 | 918 | |
| 59/50 | 1812 | | 69/60 | 1758 | |
| 49/40 | 2395 | | 59/50 | 2201 | |
| 39/30 | 1709 | | 49/40 | 1716 | |
| 29/20 | 484 | | 39/30 | 498 | |
| 19/10 | 13 | | 29/20 | 10 | |
| May | 7440 hrs | 30 % RH | June | 7200 hrs | 15% RH |
| 99/90 | 75 | | 109/100 | 6 | |
| 89/80 | 849 | | 99/90 | 968 | |
| 79/70 | 1821 | | 89/80 | 1933 | |
| 69/60 | 2335 | | 79/70 | 2295 | |
| 59/50 | 1752 | | 69/60 | 1660 | |
| 49/40 | 561 | | 59/50 | 328 | |
| 39/30 | 47 | | 49/40 | 10 | |

Table G-1 (Continued)

July 7440 hrs 30 % RH

109/100 27

99/90 1089

89/80 2135

79/70 2692

69/60 1483

59/50 14

Aug 7440 hrs 40% RH

99/90 622

89/80 2027

79/70 2792

69/60 2971

59/50 29

Sept 7200 hrs 30 % RH

99/90 188

89/80 1527

79/70 2223

69/60 2409

59/50 835

49/40 28

Oct 7440 hrs 25% RH

89/80 165

79/70 1132

69/60 2063

59/50 2428

49/40 1469

39/30 183

Nov 7200 hrs 30 % RH

79/70 24

69/60 505

59/50 1586

49/40 2533

39/30 1848

29/20 630

19/10 74

Dec 7440 hrs 50% RH

79/70 2

69/60 65

59/50 658

49/40 1874

39/30 2974

29/20 1789

19/10 273

9/0 7

Table G-2

Los Angeles, Calif.

| Jan | 7440 hrs | 75% RH | Feb | 2792 hrs | 70% RH |
|-------|----------|--------|-------|----------|--------|
| 89/80 | 22 | | 89/80 | 36 | |
| 79/70 | 196 | | 79/70 | 218 | |
| 69/60 | 1234 | | 69/60 | 1510 | |
| 59/50 | 4212 | | 59/50 | 3987 | |
| 49/40 | 172 | | 49/40 | 1132 | |
| 39/30 | 49 | | 39/30 | 9 | |
| Mar | 7440 hrs | 75% RH | Apr | 7200 hrs | 80% RH |
| 89/80 | 40 | | 89/80 | 58 | |
| 79/70 | 300 | | 79/70 | 368 | |
| 69/60 | 1953 | | 69/60 | 2570 | |
| 59/50 | 4371 | | 59/50 | 4025 | |
| 49/40 | 765 | | 49/40 | 189 | |
| 39/30 | 11 | | | | |
| May | 7440 hrs | 70% RH | June | 7200 hrs | 75% RH |
| 99/90 | 6 | | 99/90 | 13 | |
| 89/80 | 49 | | 89/80 | 35 | |
| 79/70 | 540 | | 79/70 | 1286 | |
| 69/60 | 4090 | | 69/60 | 5094 | |
| 59/50 | 269 | | 59/50 | 772 | |
| 49/40 | 8 | | | | |

Table G-2 (Continued)

| July | 7440 hrs | 75% |
|------|----------|-----|
|------|----------|-----|

| | | |
|-------|---|--|
| 99/90 | 7 | |
|-------|---|--|

| | | |
|-------|-----|--|
| 89/80 | 287 | |
|-------|-----|--|

| | | |
|-------|------|--|
| 79/70 | 2617 | |
|-------|------|--|

| | | |
|-------|------|--|
| 69/60 | 4283 | |
|-------|------|--|

| | | |
|-------|-----|--|
| 59/50 | 246 | |
|-------|-----|--|

| Aug | 7440 hrs | 80% RH |
|-----|----------|--------|
|-----|----------|--------|

| | | |
|-------|---|--|
| 99/90 | 6 | |
|-------|---|--|

| | | |
|-------|-----|--|
| 89/80 | 236 | |
|-------|-----|--|

| | | |
|-------|------|--|
| 79/70 | 2695 | |
|-------|------|--|

| | | |
|-------|------|--|
| 69/60 | 4469 | |
|-------|------|--|

| | | |
|-------|----|--|
| 59/50 | 34 | |
|-------|----|--|

| Sept | 7200 hrs | 75% RH |
|------|----------|--------|
|------|----------|--------|

| | | |
|---------|---|--|
| 109/100 | 6 | |
|---------|---|--|

| | | |
|-------|----|--|
| 99/90 | 36 | |
|-------|----|--|

| | | |
|-------|-----|--|
| 89/80 | 302 | |
|-------|-----|--|

| | | |
|-------|------|--|
| 79/70 | 2252 | |
|-------|------|--|

| | | |
|-------|------|--|
| 69/60 | 4276 | |
|-------|------|--|

| | | |
|-------|-----|--|
| 59/50 | 338 | |
|-------|-----|--|

| Oct | 7440 hrs | 80% RH |
|-----|----------|--------|
|-----|----------|--------|

| | | |
|---------|---|--|
| 109/100 | 4 | |
|---------|---|--|

| | | |
|-------|----|--|
| 99/90 | 39 | |
|-------|----|--|

| | | |
|-------|-----|--|
| 89/80 | 170 | |
|-------|-----|--|

| | | |
|-------|------|--|
| 79/70 | 1119 | |
|-------|------|--|

| | | |
|-------|------|--|
| 69/60 | 4369 | |
|-------|------|--|

| | | |
|-------|------|--|
| 59/50 | 1734 | |
|-------|------|--|

| | | |
|-------|---|--|
| 49/40 | 5 | |
|-------|---|--|

| Nov | 7200 hrs | 75% RH |
|-----|----------|--------|
|-----|----------|--------|

| | | |
|-------|---|--|
| 99/90 | 9 | |
|-------|---|--|

| | | |
|-------|-----|--|
| 89/80 | 142 | |
|-------|-----|--|

| | | |
|-------|-----|--|
| 79/70 | 622 | |
|-------|-----|--|

| | | |
|-------|------|--|
| 69/60 | 2774 | |
|-------|------|--|

| | | |
|-------|------|--|
| 59/50 | 3267 | |
|-------|------|--|

| | | |
|-------|-----|--|
| 49/40 | 386 | |
|-------|-----|--|

| Dec | 7440 hrs |
|-----|----------|
|-----|----------|

| | |
|-------|---|
| 99/90 | 2 |
|-------|---|

| | |
|-------|----|
| 89/80 | 70 |
|-------|----|

| | |
|-------|-----|
| 79/70 | 351 |
|-------|-----|

| | |
|-------|------|
| 69/60 | 1848 |
|-------|------|

| | |
|-------|------|
| 59/50 | 3992 |
|-------|------|

| | |
|-------|------|
| 49/40 | 1143 |
|-------|------|

| | |
|-------|----|
| 39/30 | 34 |
|-------|----|

TABLE G-3

Boston, Mass.

| Jan. | 7440 hrs | 50% RH | Feb | 6792 hrs | 50% RH |
|--------|----------|--------|---------|----------|--------|
| 69/60 | 6 | | 69/60 | 20 | |
| 59/50 | 253 | | 59/50 | 261 | |
| 49/40 | 825 | | 49/40 | 1144 | |
| 39/30 | 3055 | | 39/30 | 3092 | |
| 29/20 | 2180 | | 29/20 | 1561 | |
| 19/10 | 906 | | 19/10 | 554 | |
| 9/0 | 168 | | 9/0 | 119 | |
| -1/-10 | 47 | | -1/-10 | 41 | |
| Mar | 7440 hrs | 50% RH | Apr | 7200 hrs | 50% RH |
| 70/70 | 1 | | 89/80 | 26 | |
| 69/60 | 73 | | 79/70 | 133 | |
| 59/50 | 391 | | 69/60 | 633 | |
| 49/40 | 2168 | | 59/50 | 1990 | |
| 39/30 | 3521 | | 49/40 | 2579 | |
| 29/20 | 1115 | | 39/30 | 808 | |
| 19/10 | 171 | | 29/20 | 29 | |
| | | | 19/10 | 2 | |
| May | 7440 hrs | 50 RH | June | 7200 hrs | 60% RH |
| 99/90 | 11 | | 109/100 | 1 | |
| 89/80 | 192 | | 99/90 | 107 | |
| 79/70 | 729 | | 89/80 | 737 | |
| 69/60 | 1925 | | 79/70 | 1909 | |
| 59/50 | 3270 | | 69/60 | 2895 | |
| 49/40 | 1312 | | 59/50 | 1500 | |
| 39/30 | 12 | | 49/40 | 51 | |

Table G-3 Continued

| July | 7440 hrs | 60% RH |
|-------|----------|--------|
| 99/90 | 209 | |
| 89/80 | 1367 | |
| 79/70 | 3363 | |
| 69/60 | 2410 | |
| 59/50 | 91 | |

| Aug | 7440 hrs | 65% RH |
|---------|----------|--------|
| 109/100 | 1 | |
| 99/90 | 142 | |
| 89/80 | 997 | |
| 79/70 | 2931 | |
| 69/60 | 3149 | |
| 59/50 | 220 | |

| Sept | 7200 hrs | 70% RH |
|---------|----------|--------|
| 109/100 | 1 | |
| 99/90 | 26 | |
| 89/80 | 357 | |
| 79/70 | 1703 | |
| 69/60 | 2949 | |
| 59/50 | 1947 | |
| 49/40 | 216 | |
| 39/30 | 2 | |

| Oct | 7440 hrs | 60% RH |
|-------|----------|--------|
| 89/80 | 44 | |
| 79/70 | 289 | |
| 69/60 | 1658 | |
| 59/50 | 3344 | |
| 49/40 | 1848 | |
| 39/30 | 263 | |

| Nov | 7200 hrs | 50% RH |
|-------|----------|--------|
| 79/70 | 32 | |
| 69/60 | 479 | |
| 59/50 | 1740 | |
| 49/40 | 2912 | |
| 39/30 | 1726 | |
| 29/20 | 286 | |
| 19/10 | 25 | |

| Dec | 7440 hrs | 50% RH |
|-------|----------|--------|
| 69/60 | 44 | |
| 59/50 | 461 | |
| 49/40 | 1799 | |
| 39/30 | 2745 | |
| 29/20 | 1676 | |
| 19/10 | 596 | |
| 9/10 | 111 | |
| 0 | 8 | |

TABLE G- 4

New York, N.Y.

Jan 7440 hrs 50% RH

59/50 95

49/40 1511

39/30 3491

29/20 1835

19/10 485

9/0 23

Feb 6792 hrs 50% RH

69/60 13

59/50 206

49/40 1871

39/30 3212

29/20 1096

19/10 337

9/0 57

Mar 7440 hrs 50% RH

79/70 3

69/60 74

59/50 560

49/40 2345

39/30 3097

29/20 740

19/10 21

Apr 7200 hrs 50% RH

89/80 24

79/70 164

69/60 757

59/50 2713

49/40 3064

39/30 463

29/20 16

May 7440 hrs 50% RH

99/90 3

89/80 78

79/70 772

69/60 2669

59/50 3244

49/40 671

39/30 3

June 7200 hrs 50% RH

99/90 88

89/80 614

79/70 2464

69/60 3358

59/50 676

Table G-4 (continued)

| July | 7440 hrs | 60% RH | Aug | 7440 hrs | 70% RH |
|---------|----------|--------|-------|----------|--------|
| 109/100 | 3 | | 99/90 | 85 | |
| 99/90 | 132 | | 89/80 | 1223 | |
| 89/80 | 1405 | | 79/70 | 4296 | |
| 79/70 | 4524 | | 69/60 | 1881 | |
| 69/60 | 1354 | | 59/50 | 55 | |
| 59/50 | 22 | | | | |
| Sept | 7200 hrs | 70% RH | Oct | 7440 hrs | 50% RH |
| 99/90 | 19 | | 89/80 | 33 | |
| 89/80 | 317 | | 79/70 | 547 | |
| 79/70 | 2532 | | 69/60 | 2491 | |
| 69/60 | 3297 | | 59/50 | 3043 | |
| 59/50 | 969 | | 49/40 | 1229 | |
| 49/40 | 66 | | 39/30 | 97 | |
| Nov | 7200 hrs | 50% RH | Dec | | |
| 79/70 | 5 | | 69/60 | 14 | |
| 69/60 | 402 | | 59/50 | 659 | |
| 59/50 | 2428 | | 49/40 | 2143 | |
| 49/40 | 2840 | | 39/30 | 2999 | |
| 39/30 | 1350 | | 29/20 | 1330 | |
| 29/20 | 167 | | 19/10 | 376 | |
| 19/10 | 8 | | 9/0 | 19 | |

APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

REAL K,KP,L,MA,MUA,MUL,MV,NG,NPL,NU,ML,MLIN,MIOUT
DIMENSION RT(200),RML(200)
2  NAMELIST(MA,MLIN,P,TLIN,TIN,WIN,B,L,NPL,SP,THETA,TP,KP,
  *RATIO,NG,PACKW)
  J=0
7  READ(8&2)
  J=J+1
  IF(MA.LT.0.)GO TO 80
  WRITE(6,100)J
100 FORMAT(I3)
  TO=TLIN
  TO=32.
  MA=MA*60.
  MLIN=MLIN*60.
  P=P*70.727
  TLIN=TLIN+459.67
  TIN=TIN+459.67
  F=B/12.
  TO=TO+459.67
  L=L/12.
  SP=SP/12.
  THETA=THETA*.01745
  PACKW=PACKW/12.
  TP=TP/12.
  DZ=L/25.
  CA=.241
  CL=1.
  PR=.72
  RV=85.83333
  RA=53.35
  IF(RATIO.EQ.0.)BL=0.
  IF(RATIO.GT.0.)BL=B/(.5/RATIO+1.)

```

APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

IF(RATIO.EQ.1.)BL=2.*B
ED=B-2.*BL
D=ED/NG
HFGO=-.5942*TO+1370.16
DH=2.*PACKW*SP/(SP+B)
MLOUT=.98*MLIN
IF(RATIO.EQ.0.) MLOUT=MLIN
TLOUT=TIIN-30.
I=2
RT(1)=0.
RT(2)=0.
RML(1)=0.
RML(2)=0.
1 I=I+1
IF(I.GT.100)GO TO 40
ML=MLOUT
TL=TLOUT
T=TIIN
W=WIN
Z=0.
QCL=0.
QEL=0.
QDP=0.
C=0.
NV=W*MA
5 ROA=P/((W*RV+RA)*T)
IF(Z.GT.(L-.001))GOTO555
ROV=W*ROA
IF(TL-500.)3,3,4
3 ROL=62.4
GO TO 6
4 ROL=62.4-.00024792*(TL-500.)*1.8
6 PA=ROA*RA*T

```

APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

PV=P-PA
HFG=-.5942*TL+1370.16
PVSAT=((.0006369*TL+2.0883)*1.E24*TL**-5.387*EXP(-12386./TL))*144
*
ROVSAT=PVSAT/(RV*TL)
ROMIX=(ROA+ROV*W)/(1.+W)
K=.0008946*SQRT(T)/(1.+205.2/T)
MUA=7.420E-7*SQRT(T)/(1.+205.2/T)
MUL=(EXP(-(TL-492.)/49.5)+.2)*1.E-3
IF(TL.GT.613.5)MUL=MUL-.5936E-6*(TL-613.5)
VA=(MA+MV)/(ROMIX*PACKW*SP*NPL)/3600.
IF(RATIO.EQ.1.)VL=ROL*32.2*COS(THETA)*(3.*ML*MUL/ROL**2/BL/32.2/3
*600./COS(THETA)/NPL)**(2./3.)/2./MUL
IF(RATIO.GT.0..AND.RATIO.LT.1.)VL=ML/NPL/NG/ROL/.433/(BL/NG)**2/3
*600.
IF(RATIO.EQ.0.)VL=0.
V=VA+VL
CV=(19.86-597./SQRT(T)+7500./T)/18.
CMIX=(CA+CV*W)/(1.+W)
RE=V*DH*ROMIX/MUA
H=.0230*K/DH*PR**.6*RE**.8
E=H*1.5
DV=.000146*((T+TL)/2. )**2.5/(((T+TL)/2. )+441.)*14.696*144./P
HD=H/ROMIX/CMIX*(ROMIX*CMIX*DV/K)**(2./3.)
REDP=VA*DH*ROMIX/MUA
HDP=.0230*K/DH*PR**.6*REDP**.8
HDP=8.
IF(RATIO.GT.0..AND.RATIO.LT.1.)NU=TANH(SQRT(2.*DZ*(D/2.))**2*
*HDP/KP/DZ/TP))/SQRT(2.*DZ*(D/2.))**2*HDP/KP/DZ/TP)
IF(RATIO.EQ.0.0)NU=0.8
IF(RATIO.EQ.1.)NU=0.
IF(RATIO.EQ.0.0)NU=0.8
DML=+HD*(ROVSAT-ROV)*BL*DZ*NPL

```

APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

DQCL=H*(TL-T)*BL*DZ*NPL
DQEL=+(HFG+CL*(TL-TO))*DML
DQDP=(NU*HDP*(TL-T)*BD*2.*DZ+HDP*(TL-T)*BL*2.*DZ)*NPL
IF(RATIO.EQ.1.)DQDP=0.
DQ=DQCL+DQEL+DQDP
DTL=+((DQ+(ML*CL*(TL-TO)))/((ML+DML)*CL))-TL+TO
DMV=+DML
DW=(MV+DMV)/MA-W
DT=(-DQ-MA*CA*(T-TO)-W*MA*(CV*(T-TO)+HFGO)+(W+DW)*MA*HFGO)/(-MA*C
*A-(W+DW)*MA*CV)-T+TO
ML=ML+DML
QCL=QCL+DQCL
CEL=CEL+DQEL
QDP=QDP+DQDP
Q=Q+DQ
TL=TL+DTL
MV=MV+DMV
W=W+DW
T=T+DT
Z=Z+DZ
GOTO5
555 DELTL=TLOUT-TL
DELT=T-TIN
FML=(ML-MLOUT)/ML*100.
RT(I)=TL
FML(I)=ML
TOLT=TLIN-TL
IF(ABS(TOLT).LE..001)GO TO 30
IF(AINT(RT(I)*1.E3).EQ.AINT(RT(I-2)*1.E3).AND.ABS(TOLT).LE..002)
*GO TO 30
TLOUT=TLOUT+.50*TOLT
GO TO 1
30 TOLM=MLIN-ML

```


APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

IF(ABS(TOLM).LE..01)GO TO 40
IF(AINT(RML(I)*10.).EQ.AINT(RML(I-2)*10.).AND.ABS(TOLM).LE..02)
*GO TO 40
MLOUT=MLOUT+.5*TOLM
GO TO 1
40 L=L*12.
E=E*12.
PACKW=PACKW*12.
SP=SP*12.
THETA=THETA/.01745
TP=TP*12.
TO=TO-459.67
MA=MA/60.
P=P/70.727
TIN=TIN-459.67
T=T-459.67
MLIN=MLIN/60.
TLIN=TLIN-459.67
MLOUT=MLOUT/60.
ML=ML/60.
TLOUT=TLOUT-459.67
TL=TL-459.67
WRITE(5,45) J
Q=(TLIN-TLOUT)*MLOUT+MA*(W-WIN)*(TLIN-32.)
QEL=(MLIN-MLOUT)*(-.5942*(TLIN+460.))+1370.16-
* (.241)*(TLIN-T))
POEL=QEL/Q*100.
PUT HDP,H,HD,NU ,VL
WRITE(5,50)RATIO,L,B,PACKW,NPL,NG,SP,THETA
WRITE(5,60)TP,KP,TC,MA,P,TIN,T,WIN,W,ML,MLOUT
WRITE(5,70)TL,TLOUT,DELTL,PML,DELT,Q,QEL,POEL
45 FORMAT(/1H1,40X,I3/)
50 FORMAT(41X,'TOWER GEOMETRY'//43X,'WET-DRY SURFACE ',

```

APPENDIX H. COUNTERFLOW COMPUTER LISTING

```

* 'RATIO=' ,F5.3//43X, 'PACKING HEIGHT=' ,F6.2, ' IN.'//43X, 'TOTAL ' ,
* 'HEAT TRANSFER SURFACE WIDTH=' ,F7.2, ' IN.'//43X, 'PACKING ' ,
* 'WIDTH=' ,F7.2, ' IN.'//43X, 'NUMBER OF PLATES=' ,F5.0//43X, 'NUMBER' ,
* ' OF CHANNELS PER PLATE=' ,F5.0//43X, 'PLATE SPACING=' ,F5.2
* , ' IN.'//43X, 'PLATE ANGLE FROM VERTICAL=' ,F4.1, ' DEGREES'//)
60 FORMAT(43X, 'PLATE THICKNESS=' ,F5.3, ' IN.'//43X, 'PLATE CONDUCTIVITY
*=' ,F7.3, ' BTU/HR FT F'//41X, 'REFERENCE TEMPERATURE=' ,F7.2,
* ' F'////64X, 'INLET' ,17X, 'OUTLET'//41X, 'AIR FLOW RATE' ,6X, F8.2
* , ' LBM/MIN'//41X, 'AIR PRESSURE' ,9X, F5.2, ' IN. HG'//41X,
* 'AIR TEMPERATURE' ,5X, 2(F7.2, ' F' ,13X)//41X, 'HUMIDITY' ,13X, 2(F8.6,
* ' LB%/LBM' ,7X)//41X, 'WATER FLOW RATE' ,4X, 2(F9.3, ' LBM/MIN' ,6X)//)
70 FORMAT(41X, 'WATER TEMPERATURE' ,3X, 2(F7.2, ' F' ,14X)////41X, 'WATER
* ' TEMPERATURE CHANGE=' ,F7.2, ' F'//41X, 'PERCENT WATER LOSS=' ,
* F6.3, ' %'//41X, 'AIR TEMPERATURE CHANGE=' ,
* F7.2, ' F'//41X, 'TOTAL HEAT TRANSFER=' ,F11.2, ' BTU/MIN'//
* 41X, 'EVAPORATIVE HEAT TRANSFER=' ,F11.2, ' BTU/MIN'//41X,
* 'PERCENT EVAPORATIVE HEAT TRANSFER=' ,F4.1, ' %')
GO TO 7
80 WRITE(5,85)
85 FORMAT(1H1)
CONTINUE
STOP
END

```

TABLE H-1 LIST OF COMPUTER PROGRAM VARIABLES

| | | |
|-------------------|--|----------------------|
| B | Total Surface Area of a Plate Per Foot of Height | ft. |
| BD | Total Width of Dry Portion of a Plate | ft. |
| BL | Total Width of Wet Portion of a Plate | ft. |
| CA | Specific Heat at Constant Pressure for Air | BTU/lbm °R |
| CL | Specific Heat at Constant Pressure for Liquid Water | BTU/lbm °R |
| CMIX | Specific Heat at Constant Pressure for Air and Water Mixture | BTU/lbm °R |
| CV | Specific Heat at Constant Pressure for Water Vapor | BTU/lbm °R |
| D | Distance Between Water Channels in the Packing | ft. |
| DELT | Air Temperature Change | °F |
| DELT _L | Water Temperature Change | °F |
| DH | Hydraulic Diameter of the Space Between Packing Plates | ft. |
| DML | Change in Water Flow Rate | lbm/hr. |
| DMV | Change in Vapor Flow Rate | lbm/hr. |
| DQ | Change in Heat Transfer Rate | BTU/hr. |
| DQCL | Change in Convective Heat Transfer from Water Surface | BTU/hr. |
| DQDP | Change in Convective Heat Transfer from Dry Surface | BTU/hr. |
| DQEL | Change in Evaporative Heat Transfer | BTU/hr. |
| DT | Change in Air-Vapor Temperature | °R |
| DTL | Change in Water Temperature | °R |
| DV | Diffusion Coefficient for Water Vapor in Air | ft ² /hr. |
| DW | Change in Absolute Humidity | lbm/lbm |

| | | |
|-------|---|---------------------------|
| DZ | Change in Distance Through the Packing | ft. |
| H | Convective Heat Transfer Coefficient from Liquid Water to Air | BTU/hr.ft ² °R |
| HD | Mass Transfer Coefficient for Water in Air | ft/hr. |
| HDP | Convective Heat Transfer Coefficient from Dry Surface to Air | BTU/hr.ft ² °R |
| HFG | Heat of Vaporization for Water | BTU/lbm |
| HFGO | Heat of Vaporization for Water at T ₀ | BTU/lbm |
| I | Counter for Number of Iterations | |
| J | Counter for Number of Load Cases | |
| K | Thermal Conductivity of Air (used for Air-Vapor Mixture) | BTU/hr.ft°R |
| KP | Thermal Conductivity of the Plates | BTU/hr.ft°R |
| L | Total Height of Packing | ft. |
| MA | Mass Flow Rate of Air | lbm/hr. |
| ML | Mass Flow Rate of Water | lbm/hr. |
| MLIN | Mass Flow Rate of Water at Inlet | lbm/hr. |
| MLOUT | Mass Flow Rate of Water at Outlet | lbm/hr. |
| MUA | Absolute Viscosity of Air (Used for Air-Vapor Mixture) | lbm/ft.sec. |
| MV | Mass Flow Rate of Vapor | lbm/hr. |
| NG | Number of Channels on Each Packing Plate | |
| NPL | Number of Plates in the Packing | |
| NU | Fin Efficiency Between the Channels | |
| P | Total Pressure of the Air-Vapor Mixture | lbf/ft ² |
| PA | Partial Pressure of the Air | lbf/ft ² |

| | | |
|--------|---|---------------------|
| PACKW | Actual Width of Packing | ft. |
| PQEL | Percent Evaporative Heat Transfer | |
| PR | Prandtl Number for Air (used for Air-Water Mixture) | |
| PML | Percent Water Loss | |
| PV | Partial Pressure of Water Vapor | lb/ft ² |
| PVSAT | Partial Pressure of Water Vapor at Saturation | lb/ft ² |
| Q | Total Heat Transfer Rate | BTU/hr. |
| QCL | Convective Heat Transfer from Water Surface | BTU/hr. |
| QDP | Convective Heat Transfer from Dry Surface | BTU/hr. |
| QEL | Evaporative Heat Transfer | BTU/hr. |
| RA | Gas Constant for Air | ft. lb/1bm°R |
| RATIO | Fraction of Packing Surface Which is Wet | |
| RE | Reynold's Number for Air Over Water with Counter Flow | |
| REDP | Reynold's Number for Air Over Dry Surface | |
| RML | Array which Records the Values of ML After Each Iteration | |
| ROA | Density of Air | lbm/ft ³ |
| ROL | Density of Water | lbm/ft ³ |
| ROMIX | Density of Air-Vapor Mixture | lbm/ft ³ |
| ROV | Density of Water Vapor | lbm/ft ³ |
| ROVSAT | Density of Water Vapor at Saturation | lbm/ft ³ |
| RT | Array which Records the Values of TL After Each Iteration | |
| RV | Gas Constant for Water Vapor | ft./lb/1bm°R |
| SP | Spacing Between Packing Plates | ft. |
| T | Temperature of Air-Vapor Mixture | °R |

| | | |
|-------|---|---------|
| THETA | Angle Between the Plates and Vertical | radians |
| TIN | Temperature of Air at Inlet | °R |
| TL | Temperature of Water | °R |
| TLIN | Temperature of Water at Inlet | °R |
| TLOUT | Temperature of Water at Outlet | °R |
| TO | Reference Temperature | °R |
| TOLM | Parameter Used to Check Convergence of TL | |
| TOLT | Parameter Used to Check Convergence of ML | |
| TP | Thickness of the Plates | ft. |
| V | Velocity of Air Relative to Water | ft/sec. |
| VA | Air Velocity | ft/sec. |
| VL | Water Velocity | ft/sec. |
| W | Absolute Humidity of the Air | lbm/lbm |
| WIN | Absolute Humidity of the Air at Inlet | lbm/lbm |
| Z | Distance from Air Inlet | ft. |

TABLE H-2: LIST OF PROGRAM INPUT PARAMETERS

| | | |
|-------|--|-------------------------|
| B | Total Surface Area of Plate Per Foot of Height | in. |
| KP | Thermal Conductivity of Plates | BTU/hr.ft ^{°R} |
| L | Total Height of Packing | in. |
| MA | Mass Flow Rate of Air | lbm/min. |
| MLIN | Mass Flow Rate of Water at Inlet | lbm/min. |
| NG | Number of Channels on Each Packing Plate | |
| NPL | Number of Plates in the Packing | |
| P | Total Pressure of the Air-Vapor Mixture | in.hg |
| PACKW | Actual Width of Packing | in. |
| RATIO | Fraction of Packing Surface Which is Wet | |
| SP | Spacing Between Packing Plates | in. |
| THETA | Angle Between the Plates and Vertical | degrees |
| TIN | Temperature of Air at Inlet | °F |
| TLIN | Temperature Water at Inlet | °F |
| TP | Thickness of the Plates | in. |
| WIN | Absolute Humidity of Air at Inlet | lbm/lbm |

APPENDIX I. CROSSFLOW COMPUTER LISTING

```

INTEGER AIRHUM,AIRTEM,AIRFLO
REAL OUTAIR(50),OUTHUM(50),OUTLIQ(50),OUTFLO(50),NU
REAL K,KP,L,NG,NPL,MLIN,MLOUT,DATA(41,41,5),MA,NOMIX
LOGICAL LMIX,SCAN
DATA YESMIX/' ' //,NOMIX/'UN'/
2  NAMELIST(MA,MLIN,P,TLIN,TIN,WIN,B,L,NPL,SP,THETA,
C TP,KP,RATIO,NG,PACKW,LMIX,SCAN)
   JJJ=0
7  LMIX=.FALSE.
   SCAN=.FALSE.
   JJJ=JJJ+1
   READ(8,2)
   IF(MA.LT.0.) GOTO444
   IXX=25.
   IXX=20
   XX=IXX
   LIQTEM=1
   AIRTEM=2
   LIQFLO=3
   AIRFLO=4
   AIRHUM=5
   REABLK=L*NPL*SP/144./XX
   TO=TLIN+459.67
   DO 11 I=1,IXX
   DATA(1,I,LIQTEM)=TLIN+459.67
   DATA(I,1,AIRTEM)=TIN+459.67
   DATA(1,I,LIQFLO)=MLIN*60./XX
   DATA(I,1,AIRFLO)=MA*60./XX
   DATA(I,1,AIRHUM)=WIN
11  CONTINUE
   IF(RATIO.EQ.0.)ABLIQ=0.
   IF(RATIO.GT.0.) ABLIQ=B/(.5/RATIO+1.)*L*NPL/XX**2.

```


APPENDIX I. CROSSFLOW COMPUTER LISTING

```

C /144.
IF(RATIO.EQ.1.) ABLIQ=2.*B*L*NPL/144./XX**2.
ABDRY=L*NPL*D/XX**2./144. -2.*ABLIQ
DH=2.*AFABLK/L/NPL*12.*XX*PACKW/B
D=ABDRY/L*XX/EG*XX*12./NPL
DO 33 J=1,IXX
DO 22 I=1,IXX
II=I+1
JJ=J+1
CALL BLOCK(DATA(I,J,AIRTEM),DATA(I,J,LIQTEM),
C DATA(I,J,AIRHUM),DATA(I,J,LIQFLO),DATA(I,JJ,AIRTEM),
C DATA(II,J,LIQTEM),DATA(I,JJ,AIRHUM),DATA
C (II,J,LIQFLO),IXX,AFABLK,DH,ABLIQ,ABDRY,P,DATA
C (1,1,4),D,TC,KP,TP,RATIO,NU,HDP,H,HD)
22 CONTINUE
IF(LMIX)CALL MIX(DATA,IXX,J+1,DATA(1,1,4),XWAV,XTAV)
IF(.NOT.LMIX)GOTO33
DO 34 IRCW=1,IXX
DATA(IRCW,J+1,AIRHUM)=XWAV
34 DATA(IRCW,J+1,AIRTEM)=XTAV
33 CONTINUE
DO 18 I=1,IXX
CUTAIR(I)=DATA(I,IXX+1,AIRTEM)-459.67
GUTHUM(I)=DATA(I,IXX+1,AIRHUM)
18 CONTINUE
DO 19 I=1,IXX
OUTLIQ(I)=DATA(IXX+1,I,LIQTEM)-459.67
OUTFLO(I)=DATA(IXX+1,I,LIQFLO)/60.
19 CONTINUE
24 FORMAT(/1X,I5,F10.4,F10.4)
FL=0.
ATL=0.
DO 31 I=1,IXX

```

APPENDIX I. CROSSFLOW COMPUTER LISTING

```

      ATL=ATL+DATA(IXX+1,I,LIQTEM)
      FL=FL+DATA(IXX+1,I,LIQFLO)
31  CONTINUE
      MLCUT=FL/60.
      TLOUT=ATL/XX-459.67
      J=IXX+1
      CALL MIX(DATA,IXX,J,DATA(1,1,4),WOUT,TAV)
      TOUT=TAV-459.67
      QTOT=(TLIN-TLOUT)*MLCUT+MA*(WOUT-WIN)*(TLIN-32.)
      QEL=(MLIN-MLCUT)*(-.5942*(TLIN+460.)+1370.16
C    -(.241)*(TLIN-TOUT))
      QRAT=QEL/QTOT*100.
      DELT=TOUT-TIN
      DELTL=TLOUT-TLIN
      FML=(MLIN-MLCUT)/MLIN*100.
      TO=TO-459.67
      OUTMIX=NOMIX
      IF(LMIX)OUTMIX=YESMIX
      WRITE(5,45)JJJ,OUTMIX
      PUT JJJ,H,HDP,HD,NU
45  FORMAT(1H1      ,40X,I3,4X,'CROSSFLOW CONFIGURATION, AIRFLOW ',
C    A2,'MIXED'//)
      WRITE(5,50)RATIO,L,B,PACKW,NPL,NG,SP,THETA
      WRITE(5,60)TP,KP,TO,MA,P,TIN,TOUT,WIN,WOUT,MLIN,MLCUT
      WRITE(5,70)TLIN,TLOUT,DELT,FML,DELT,QTOT,QEL,QRAT
50  FORMAT(41X,'TOWER GEOMETRY'//43X,'WET-DRY SURFACE ',
* 'RATIO=',F5.3//43X,'PACKING HEIGHT=',F6.2,' IN.'//43X,'TOTAL ',
* 'HEAT TRANSFER SURFACE WIDTH=',F7.2,' IN.'//43X,'PACKING ',
* 'WIDTH=',F7.2,' IN.'//43X,'NUMBER OF PLATES=',F7.0//43X,'NUMBER',
* ' OF CHANNELS PER PLATE=',F5.0//43X,'PLATE SPACING=',F5.2
* ', IN.'//43X,'PLATE ANGLE FROM VERTICAL=',F4.1,' DEGREES'//)
60  FORMAT(43X,'PLATE THICKNESS=',F5.3,' IN.'//43X,'PLATE CONDUCTIVITY
* ',F7.3,' BTU/HR FT F'//41X,'REFERENCE TEMPERATURE=',F7.2,

```

APPENDIX I. CROSSFLOW COMPUTER LISTING

```

* F'////64X,'INLET',17X,'OUTLET'//41X,'AIR FLOW RATE',6X,E9.3
*, 'LBM/MIN'//41X,'AIR PRESSURE',9X,F5.2,' IN. HG'//41X,
*'AIR TEMPERATURE',5X,2(F7.2,' F',13X)//41X,'HUMIDITY',13X,2(F8.6,
*'LBM/LBM',7X)//41X,'WATER FLOW RATE',4X,2(E9.3,' LBM/MIN',6X)/)
70 FORMAT(41X,'WATER TEMPERATURE',3X,2(F7.2,' F',14X)////41X,'WATER
*, 'TEMPERATURE CHANGE=',F7.2,' F'//41X,'PERCENT WATER LOSS=',
*F6.3,' %'//41X,'AIR TEMPERATURE CHANGE=',
*F7.2,' F'//41X,'TOTAL HEAT TRANSFER=',E11.3,' BTU/MIN'//
*41X,'EVAPORATIVE HEAT TRANSFER=',E11.3,' BTU/MIN'//41X,
*'PERCENT EVAPORATIVE HEAT TRANSFER=',F4.1,' %')
IF(.NOT.SCAN)GOTO7
WRITE(5,878)
DO 879 I=1,IXX
879 WRITE(5,877)I,OUTLIQ(I),OUTFLO(I),OUTAIR(I),OUTHUM(I),I
878 FORMAT(1H1,5X,'WATER OUTLET CONDITIONS-',6X,'AIR OUTLET CONDIT',
C 'IONS'/6X,'IN DIRECTION OF AIRFLOW',7X,'TOP TO BOTTOM'/7X,
C 'TEMP.',7X,'FLOW RATE',10X,'TEMP.',6X,'SPEC. HUMIDITY'/7X,
C 'DEG. F',5X,'LBM/MIN',12X,'DEG. F',5X,'LBM/LBM'/)
877 FORMAT(1X,I3,2X,F7.1,5X,F7.1,11X,F7.1,5X,F8.5,4X,I3)
GOTO7
444 STOP
END
PROGRAM *MAIN* HAS NO ERRORS

```

APPENDIX I. CROSSFLOW COMPUTER LISTING

```

SUBROUTINE BLOCK(TIN,TLIN,WIN,MLIN,TOUT,TLOUT,
C WOUT,MIOUT,IXX,AFABLK,DH,ABLIQ,ABDRY,PP,MA,D,
C TO,KP,TP,RATIO,NU,HDP,H,HD)
REAL K,KP,L,MA,MUA,MUL,MV,NU,MLIN,MIOUT
CA=.241
CL=1.
ER=.72
RV=85.8333
F=PP*70.727
RA=53.35
HFGO=-.5942*TO+1370.16
NV=WIN*MA
ROA=P/((WIN*RV+RA)*TIN)
ROV=WIN*ROA
IF(TLIN-500.)3,3,4
3  ROL=62.4
   GOTO6
4  ROL=62.4-.00024792*(TLIN-500.)**1.8
6  PA=ROA*RA*TIN
   PV=P-PA
   HFG=-.5942*TLIN+1370.16
   PVSAT=((.0006369*TLIN+2.0883)*1.E24*TLIN**-
C 5.387*FXP(-12386./TLIN))*144.
   ROVSAT=PVSAT/RV/TLIN
   ROMIX=(ROA+ROV*WIN)/(1.+WIN)
   K=.0008946*SQRT(TIN)/(1.+205.2/TIN)
   MUA=7.42E-7*SQRT(TIN)/(1.+205.2/TIN)
   MUL=(FXP(-(TLIN-492.)/49.5)+.2)*1.E-3
   IF(TLIN.GT.613.5)MUL=MUL-.5936E-6*(TLIN-613.5)
   VA=(MA+NV)/ROMIX/AFABLK/3600.
   CV=(19.86-597./SQRT(TIN)+7500./TIN)/18.
   CMIX=(CA+CV*WIN)/(1.+WIN)
   EE=VA*DH*ROMIX/MUA

```

APPENDIX I. CROSSFLOW COMPUTER LISTING

```

H=5.78
DV=.000146*((TIN+TLIN)/2.))**2.5/(((TIN+TLIN)/
C 2.))+441.))*14.696*144./P
HD=H/ROMIX/CHIX*(ROMIX*CHIX*DV/K)**(2./3.)
HDP=5.78

```

C
C
C

```

H AND HDP CALCULATED BY HAND FROM JE-CHIN CORRELATION

IF(RATIO.GE.0.)NU=TANH(SQRT(2.*(D/2.))**2.*HDP/
C KP/TP*12.))/SQRT(2.*(D/2.))**2.*HDP/KP/TP*12.)
IF(RATIO.EQ.1.)NU=0.
IF(RATIO.EQ.0.)NU=.8
DML=HD*(ROVSAT-ROV)*ABLIQ
DQCL=H*(TLIN-TIN)*ABLIQ
DQEL=(HFG+CL*(TLIN-TO))*DML
DQDP=(NU*HDP*(TLIN-TIN)*ABDRY*2.))+HDP*(TLIN-TIN)
C *ABLIQ*2.
IF(RATIO.EQ.1.)DQDP=0.
DQ=DQCL+DQEL+DQDP
DTL=+((DQ+((MLIN-DML)*CL*(TLIN-TO)))/(+(MLIN*CL)))-TLIN+TO
DMV=DML
DW=(MV+DMV)/MA-WIN
DT=(-DQ-MA*CA*(TIN-TO)-WIN*MA*(CV*(TIN-TO)+HFGO)
C +(WIN+DW)*MA*HFGO)/(-MA*CA-(WIN+DW)*MA*CV)
C -TIN+TO
MLOUT=MLIN-DML
TLOUT=TLIN-DTL
WOUT=WIN+DW
TOUT=TIN+DT
RETURN
END

```

PROGRAM BLOCK HAS NO ERRORS

APPENDIX I. CROSSFLOW COMPUTER LISTING

```
SUBROUTINE MIX(DATA, IXX, J, AIRFLO, WOUT, TOUT)
  INTEGER AIRHUM, AIRTEM
  REAL DATA(41,41,5)
  AIRTEM=2
  AIRHUM=5
  HAIR=0.
  WAIR=0.
  DO 1 I=1, IXX
    HAIR=HAIR+((.241)+(.45)*DATA(I,J,AIRHUM))*AIRFLO*
C    DATA(I,J,AIRTEM)
1    WAIR=WAIR+DATA(I,J,AIRHUM)
    XX=IXX
    WOUT=WAIR/XX
    TOUT=HAIR/XX/AIRFLO/((.241+.45*WOUT)
  RETURN
  END
```

PROGRAM MIX HAS NO ERRORS

TABLE I-1 LIST OF VARIABLES - CROSSFLOW PROGRAM UNITS

| VARIABLE | DESCRIPTION | UNITS |
|----------|--|---------------------------|
| ABDRY | Convective heat transfer surface area per grid section | ft ² |
| ABLIQ | Water free surface area per grid section | ft ² |
| AFABLK | Airflow area per grid section | ft ² |
| AIRFLO | Dummy index for DATA array | |
| AIRHUM | Dummy index for DATA array | |
| AIRTEM | Dummy index for DATA array | |
| ATL | Intermediate variable for averaging exit water temperature | F |
| B | Total plate width | in |
| BLOCK | Subroutine which calculates heat and mass transfer for a grid section | |
| D | Distance between water channels in the packing plates | ft |
| DATA | Array holding air and water inlet conditions for each grid section | |
| DELT | Average air temperature change across the tower | F |
| DELTL | Average water temperature change across the tower | F |
| DH | Hydraulic diameter of airflow channel | ft |
| FL | Sum of exit water flow rates | lb _m /hr |
| FML | Ratio of water evaporation rate to inlet flow rate | % |
| H | Convective heat transfer coefficient from free water surface | BTU/hr-ft ² ·F |
| HD | Mass transfer coefficient from free water surface | ft/hr |
| HDP | Convective heat transfer coefficient from dry surface of packing plate | BTU/hr-ft ² ·F |
| I,II | Dummy counting variables | |
| IROW | Dummy variable | |
| IXX | Number of rows and columns of grid | |
| J,JJJ | Dummy counting variables | |

| VARIABLE | DESCRIPTION | UNITS |
|----------|--|----------------------------------|
| KP | Thermal conductivity of packing plate | BTU/hr-ft°F |
| L | Height of packing plate | in |
| LIQFLO | Dummy index for DATA array | |
| LIQTEM | Dummy index for DATA array | |
| LMIX | Logical switch for unmixed or well-mixed flow conditions | |
| MA | Inlet air mass flow rate | lb _m /min |
| MIX | Subroutine for mixing air flows across tower | |
| MLIN | Water mass flow rate into tower | lb _m /min |
| MLOUT | Water mass flow rate out of tower | lb _m /min |
| NG | Number of water channels per plate | |
| NOMIX | Character string for printout title | |
| NPL | Number of packing plates in tower | |
| NU | Fin efficiency | |
| OUTAIR | Scan of air outlet temperatures | F |
| OUTFLO | Scan of water outlet flow rates | lb _m /min |
| OUTHUM | Scan of outlet air specific humidity | lb _m /lb _m |
| OUTLIQ | Scan of outlet water temperatures | F |
| OUTMIX | Character string for printout title | |
| P | Air pressure | in. Hg. |
| PACKW | Width of packing section | in. |
| QEL | Evaporative heat transfer rate | BTU/min |
| QRAT | Ratio of Evaporative to total heat transfer rate | % |
| QTOT | Total heat transfer rate | BTU/min |
| RATIO | Wet-to-dry surface area ratio | |
| SCAN | Logical switch for exit scan output | |
| SP | Packing plate spacing | in |
| TAV | Average column air temperatures from MIX | R |
| THETA | Packing plate angle from vertical | ° |
| TIN | Inlet air temperature | F |
| TLIN | Inlet water temperature | F |

| VARIABLE | DESCRIPTION | UNITS |
|----------|---|---------------------------|
| TLOUT | Average Outlet water temperature | F |
| TO | Reference Temperature | R |
| TOUT | Average outlet water temperature | F |
| TP | Packing plate thickness | in. |
| WIN | Inlet air specific humidity | lb_m/lb_m |
| WOUT | Outlet air specific humidity | lb_m/lb_m |
| XTAV | Average column air temperature from MIX | R |
| XWAV | Average column air specific humidity from MIX | lb_m/lb_m |
| XX | Number of rows and columns in grid | |
| YESMIX | Character string for title printout | |

SUBROUTINE BLOCK

| VARIABLE | DESCRIPTION | UNITS |
|----------|---|--|
| ABDRY | Convective heat transfer surface area per grid section | ft^2 |
| ABLIQ | Water free surface area per grid section | ft^2 |
| AFABLK | Air flow area in grid section | ft^2 |
| CA | Heat capacity of air | $\text{BTU}/\text{lb}_m^\circ\text{F}$ |
| CL | Heat capacity of liquid water | $\text{BTU}/\text{lb}_m^\circ\text{F}$ |
| CMIX | Heat capacity of air-water vapor mixture | $\text{BTU}/\text{lb}_m^\circ\text{F}$ |
| CV | Heat capacity of water vapor | $\text{BTU}/\text{lb}_m^\circ\text{F}$ |
| D | Distance between water channels in packing plate | ft |
| DH | Hydraulic diameter of air flow channel | ft |
| DML | Evaporation rate of water in grid section | lb_m/hr |
| DMV | Rate of change of water vapor mass flow across grid section (= DML) | lb_m/hr |
| DQ | Total heat transfer rate from grid | BTU/hr |
| DQCL | Heat transfer rate due to convection from water free surface | BTU/hr |
| DQDP | Heat transfer rate due to convection from dry plate surface | BTU/hr |
| DQEL | Heat transfer rate due to mass transfer from water free surface | BTU/hr |
| DT | Change in air temperature across grid section | F |
| DTL | Change in water temperature across grid section | F |
| DV | Diffusivity of water vapor in air | ft^2/hr |

| VARIABLE | DESCRIPTION | UNITS |
|----------|---|--|
| DW | Change in specific humidity across grid section | lb_m/lb_m |
| H | Convective heat transfer coefficient for water free surface | $\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$ |
| HD | Mass transfer coefficient for water free surface | ft/hr |
| HDP | Convective heat transfer coefficient for dry surface of packing plate | $\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$ |
| HFG | Latent heat of vaporization of liquid water | BTU/lb_m |
| HFGO | Latent heat of vaporization of liquid water at T0 | BTU/lb_m |
| IXX | Number of rows and columns of grid | |
| K | Thermal conductivity of air | $\text{BTU/hr-ft}^\circ\text{F}$ |
| KP | Thermal conductivity of packing plate | $\text{BTU/hr-ft}^\circ\text{F}$ |
| MA | Air mass flow rate through grid section | lbm/hr |
| MLIN | Water mass flow rate into grid section | lbm/hr |
| MLOUT | Water mass flow rate out of grid section | lbm/hr |
| MV | Water vapor mass flow rate into grid section | lbm/hr |
| NU | Fin efficiency | |
| P | Atmospheric pressure | in.Hg. |
| PA | Partial pressure of dry air | lb_f/ft^2 |
| PP | Atmospheric pressure | in. Hg. |
| PR | Prandtl number of air | |
| PV | Partial pressure of water vapor | lb_f/ft^2 |
| PVSAT | Partial pressure of water vapor at saturation conditions | lb_f/ft^2 |
| RA | Gas constant for air | $\text{ft-lb}_g/\text{lb}_m\text{-}^\circ\text{R}$ |
| RATIO | Ratio of wet-to-dry surface area | |
| RE | Reynolds number | |
| ROA | Density of air | lbm/ft^3 |
| ROL | Density of liquid air | lbm/ft^3 |
| ROMIX | Density of air-water vapor mixture | lbm/ft^3 |
| ROV | Density of water vapor | lbm/ft^3 |
| ROVSAT | Density of water vapor at saturation conditions | lbm/ft^3 |
| RV | Gas constant for water vapor | $\text{ft-lb}_f/\text{lb}_m\text{-}^\circ\text{R}$ |

| VARIABLE | DESCRIPTION | UNITS |
|----------|--|---------|
| TIN | Temperature of air entering grid section | R |
| TLIN | Temperature of water entering grid section | R |
| TLOUT | Temperautre of water exiting grid section | R |
| To | Reference temperature | R |
| TOUT | Temperature of air leaving grid section | R |
| TP | Thickness of packing plate | in |
| VA | Velocity of air through grid section | ft/sec |
| WIN | Specific humidity of air entering grid section | lbm/lbm |
| WOUT | Specific humidity of air leaving grid section | lbm/lbm |

Subroutine MIX

| VARIABLE | DESCRIPTION | UNITS |
|----------|--|---------|
| AIRFLO | Mass flow rate of air through grid section | lbm/hr |
| AIRHUM | Dummy index for DATA matrix | |
| AIRTEM | Dummy index for DATA matrix | |
| DATA | Array holding air and water inlet conditions for each grid section | |
| HAIR | Summing variable for air enthalpy averaging | BTU/lbm |
| I | Counting variable | |
| IXX | Number of rows and columns in grid | |
| J | Counting variable | |
| TOUT | Average air temperature of well-mixed flow | R |
| WAIR | Summing variable for specific humidity average | lbm/lbm |
| WOUT | Average specific humidity of well-mixed flow | lbm/lbm |
| XX | Number of rows and columns in grid | |

APPENDIX J

DETERMINATION OF THE DRY PLATE SURFACE HEAT TRANSFER COEFFICIENTS

The dry surface heat transfer coefficient for the model tower packing plates was experimentally determined by monitoring the temperature response of a single plate when subjected to a step change in air temperature.

Chilled air was used to cool the metal packing plates 15 to 20°F below ambient laboratory conditions. When the plate reached a uniform temperature the model tower exhaust fan was started, drawing the warmer laboratory air over the plates. Plate surface and local air temperatures were recorded at 10 second intervals until plate temperature matched air temperature. A set of typical data is shown in Table J-1.

Reference [15] contains an analysis for this type of experimental data. Given air flowrate intake temperature, time-temperature behavior of the plates and air stream, and tower physical properties, the average dry surface heat transfer coefficient may be calculated. The steps involved in this calculation are outlined below with the values in parentheses being a sample calculation with the data of Table J-1.

A more detailed explanation of the assumptions and dimensionless parameters can be found in Reference [15].

TABLE J-1

Temperature Response of Glavanized Packing Plate

| Time | Air Temperature°F | Time | Plate Temperature°F |
|------|-------------------|------|---------------------|
| 0 | 56.8 | 10 | 57.3 |
| 20 | 64.4 | 30 | 60.4 |
| 40 | 67.9 | 50 | 63.0 |
| 60 | 69.8 | 70 | 65.2 |
| 80 | 71.1 | 90 | 67.1 |
| 100 | 72.0 | 110 | 68.6 |
| 120 | 72.7 | 130 | 69.9 |
| 140 | 73.3 | 150 | 71.0 |
| 160 | 73.7 | 170 | 71.9 |
| 180 | 73.8 | 190 | 72.5 |
| 200 | 74.0 | 210 | 73.0 |
| 220 | 74.0 | 230 | 73.4 |
| 240 | 74.2 | 250 | 73.7 |
| 260 | 74.4 | 270 | 74.0 |

Plate Properties

Length 28 inches
 Width 43 inches
 Spacing 1.5 inches
 Thickness .0225 inches
 Density 487 lb_m/ft³
 Heat Capacity 0.113 BTU/lb_m-F
 Initial Temperature 56.8°F

Air Properties

Density 0.24 lbm/ft³
 Heat Capacity .08 BTU/lb_m-F
 Velocity 476 ft/min
 Temperature 74.4°F

Steps for Calculating h

1. Compute C_w from properties and air flow rate ($C_w = 104.8$)
2. Compute $\theta^*(t)$ ($\theta(t) = 3.4 t$)
3. Compute NTU from properties allowing h as an unknown (NTU = 0.148 h)
4. Compute $\theta^* t - 1/C_w^*$ ($\theta^*(t) - 1/C_w^* \approx \theta^*(t)/C_w^* = .0327 t$)
5. Select a value of $\theta^* - 1/C_w^*$ and compute the time at which that value occurs. (For $\theta^* - 1/C_w^* = 4$, $t = 122$ sec)
6. From experimental data find T_{plate} at time calculated in Step 5.
 ($T_{plate} = 69.5$).

7. Using this value of T_{plate} calculate ϵ_w^* ($\epsilon_w^* = 0.716$).
8. From Fig. 3-14 (Ref [15]) use ϵ_w^* from Step 7 and $\theta^* - 1/C_w^*$ from Step 4 to find NTU. (NTU = 0.48).
9. Use NTU values from Steps 4 and 8 to solve for h . ($h = 3.3 \text{ BTU/hr-ft}^2\text{°F}$).