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"WET COOLING TOWERS: 'RULE-OF-THUMB' DESIGN AND SIMULATION"

Stephen A. Leeper

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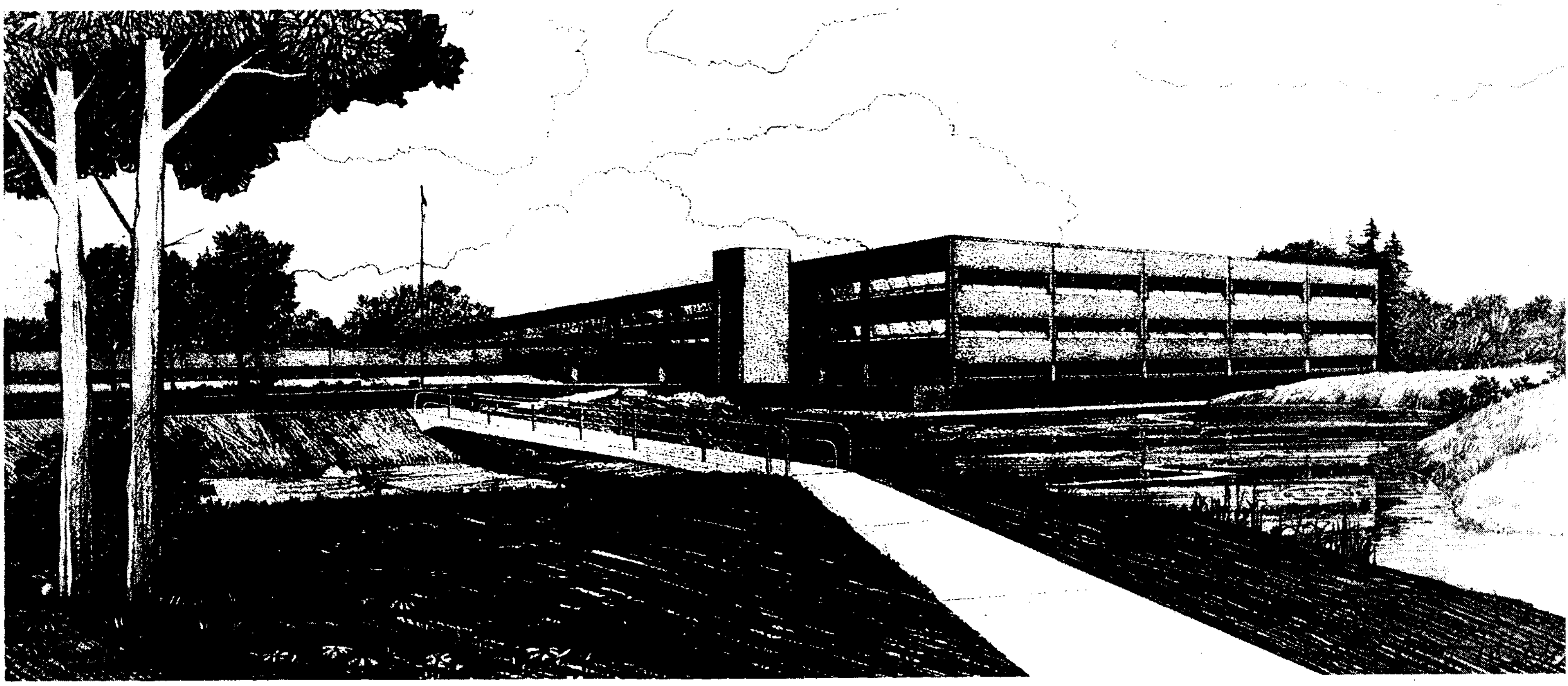

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CONTENTS

Abstract

A survey of wet cooling tower literature was performed to develop a simplified method of cooling tower design and simulation for use in power plant cycle optimization.

In the report the theory of heat exchange in wet cooling towers is briefly summarized. The Merkel equation (the fundamental equation of heat transfer in wet cooling towers) is presented and discussed. The cooling tower fill constant (Ka) is defined and values derived. A rule-of-thumb method for the optimized design of cooling towers is presented. The rule-of-thumb design method provides information useful in power plant cycle optimization, including tower dimensions, water consumption rate, exit air temperature, power requirements and construction cost. In addition, a method for simulation of cooling tower performance at various operating conditions is presented. This information is also useful in power plant cycle evaluation.

Using the information presented in this report, it will be possible to incorporate wet cooling tower design and simulation into a procedure to evaluate and optimize power plant cycles.

NOMENCLATURE

a	-	Specific Transfer Surface (ft^2_{TA}/ft^3_{fill})
A	-	Approach (T_2 - t_{wb}; ^oF)
B	-	Base Area (ft^2_B)
B_d	-	Blowdown (lbs H_2O/hr)
C	-	A constant
C_p	-	Heat Capacity of Water (Btu/lb^oF)
D	-	Drift (lbs H_2O/hr)
E	-	Evaporation (lbs H_2O/hr)
F	-	Air Flow Rate (actual cubic feed per minute; acfm)
G	-	Air Flow Rate (lbs air/hr)
h	-	Enthalpy of Air (Btu/lb dry air)
H	-	Air Humidity (lb H_2O/lb dry air)
H_p	-	Head of Pump (ft)
K	-	Air Mass Transfer Constant (lbs air/hr ft^2_{TA})
Ka	-	Volumetric Air Mass Transfer Constant (lbs air/hr ft^3_{fill})
$\frac{Ka\overline{V}}{\overline{L}}$	-	Tower Characteristic (lbs air/lb H_2O)
L	-	Water Flow Rate (lbs H_2O/hr)
$\bar{L}$	-	Loading Factor (lbs H_2O/hr ft^2_B)
L/G	-	Water-Air Flow Rate Ratio (lbs H_2O/lb air)
M	-	Makeup (lbs H_2O/hr)
P	-	Power (hp)
Q	-	Heat Load (Btu/hr)
R	-	Range (T_1 - T_2; ^oF)
t	-	Air Temperature (^oF)
T	-	Water Temperature (^oF)
TC	-	Tower Characteristic ($Ka\overline{V}/\overline{L}$; lbs air/lb H_2O)

V - Fill Volume, Total (ft^3_{fill})

$\overline{V}$ - Specific Fill Volume (ft^3_{fill}/ft^2_B)

Z - Fill Height (ft)

η - Fan Efficiency (dimensionless; ∿0.80)

ρ - Density (lb/ft^3)

$ - Dollars

Subsymbols

a - Air

B - Base Area

calc - Calculated Value

des - Design

fill - Fill Volume

F - Fan

mix - Mixture of Air and Water Vapor

op - Operation

P - Pump

sa - Saturated Air @ Water Temperature

t - Air Temperature

T - Water Temperature

TA - Transfer Area

w - Water Vapor

wb - Wet Bulb Temperature

1 - Inlet Condition

2 - Exit Condition

I. INTRODUCTION

The design of wet cooling towers is a competitive field of technology, where design methods and constants are proprietary information. However, the approximate design of cooling towers using rules-of-thumb is presented and provides information suitable for use in power plant cycle optimization, including tower dimensions, water consumption rate, exit air temperature, power requirements and construction cost. A method for the simulation of cooling tower performance at various operating conditions is also presented.

Several types of wet cooling tower exist. Wet cooling towers can be natural or mechanical draft. Mechanical draft towers can be either forced or induced draft. Air and water flow can be crosscurrent, countercurrent or both. The fundamentals of wet cooling are presented by McKelvey and Brooke (1). Mechanical draft cooling towers are the predominate types of cooling towers built in the United States. Therefore, the design of mechanical draft cooling towers is the subject of this paper.

In wet cooling towers, air and water are intimately mixed to provide heat transfer. Therefore, psychometry is the basis for analysis of heat transfer in wet cooling towers. Air-water psychometric data and psychometry theory are presented in several references (1, 2, 3).

Heat transfer in cooling towers occurs by two major mechanisms: transfer of sensible heat from water to air (convection) and transfer of latent heat by the evaporation of water (diffusion). Both of these mechanisms operate at the air-water boundary layer. The total heat transfer is the sum of these two boundary layer mechanisms. The total heat transfer can also be expressed in terms of the change in enthalpy of each bulk phase. The heat transfer at the boundary layer is equal to the heat transfer in the bulk phases. After manipulation of the terms, a fundamental equation of heat transfer in cooling towers (the Merkel equation) is obtained.

The Merkel equation, named after F. Merkel who first derived it, is:

$$\frac{Ka\overline{V}}{\overline{L}C_p} = \int_{T2}^{T1} \frac{dT}{h_{sa}-h_a}$$

The theory of heat transfer in wet cooling towers is presented in several references (1, 3, 4, 5). An especially clear derivation of the Merkel equation is presented by Kern (6). The enthalpy of the air stream is h_a. The air stream is in contact with water at a different temperature. The enthalpy of saturated air at the water temperature is h_{sa}. Air does not reach this enthalpy. The driving force in the Merkel equation (Δh_{DF}) is the difference between the enthalpy of saturated air at the water temperature and the enthalpy of air at the air temperature:

$$\text{"Driving Force"} = \Delta h_{DF} = h_{sa} - h_a$$

Strictly speaking, enthalpy difference is not the driving force in wet cooling. The driving force in wet cooling is actually the difference in water vapor pressure between the water and air phases (7). The tower characteristic $\left(\frac{Ka\overline{V}}{\overline{L}}\right)$ or design point is determined by solving the right-hand side of the Merkel equation.

The Merkel equation is presented differently by various authors. Before using the Merkel equation, care is required to determine the units required by K. As defined above, K is an air mass transfer constant. In other sources, K is the water mass transfer constant (2) or the convective heat transfer coefficient (5). Also, C_p is frequently left out of the Merkel equation since its value is 1.0, a convention adopted in this paper. The units of C_p cannot be overlooked, however.

The mass and heat transfer characteristics of cooling tower fill are described by Ka, a volumetric mass transfer constant. Mass and heat transfer occur on the wetted fill surface and on the surface of the drops. As a result, the specific mass transfer surface (a) is difficult to measure. Therefore, Ka is regarded as a single constant. Ka is a measure of the mass and heat transfer rate through the boundary layer per unit of fill volume. Larger values of Ka reflect better mass and heat transfer characteristics of fill.

Cooling tower and fill vendors do not release values of Ka. However, Ka values can be determined by back-calculation for existing towers. For 16 cooling towers built by Research-Cottrell (8), Ka values are between 64 and 140 with an average value of 95 $\pm$ 35 (two standard deviations). For 39 Marley Company cooling towers (9), values of Ka varied from 49 to 152 with 100 $\pm$ 30 as the average value. For conventional types of cooling tower fill, a Ka value of 100 gives reasonable fill heights (6, p.601). This value of Ka applies to Ka as defined in the above form of the Merkel equation. Ka is not strictly a constant, but is a complex function of several operating variables. Ka does remain relatively constant over normal operating variable ranges.

Detailed design of cooling towers is a trial and error iterative procedure. Once a set of design conditions is defined, designs are performed at several outlet air temperatures. These designs are compared to determine the optimum. Optimization requires a trade-off between operating and construction costs. More detailed discussions of cooling tower optimization for use with power plants can be found in Dickey (9) and Clark (10).

II. DESIGN PROCEDURE

A. Problem Statement

For a given cooling tower design, the quantity of water to be treated (L) and its inlet temperature (T_1) are known. The outlet water temperature (T_2) is specified. The difference between the inlet and outlet water temperatures is the range (R). An ambient wet-bulb temperature (t_{wb};t_1) is chosen for design, such that it is exceeded only three to five percent of the time. The difference between the wet-bulb temperature and the outlet water temperature is the approach (A). The outlet water temperature approaches the air wet-bulb temperature, which is the limiting temperature to which water can be cooled. Generally, cooling towers are designed with an approach of 10 to 15 degrees (Fahrenheit). The approach is seldom less than five degrees.

B. Outlet Air Temperature (t_2)/Water-Air Flow Rate Ratio (L/G)

For a given set of cooling tower design conditions, an optimum design (outlet air temperature/water-air flow rate ratio) exists. The optimum design will result in minimum construction and operating costs. A good correlation exists between the optimum outlet air temperature and the inlet and outlet water temperatures:

$$t_2 = \frac{T_1 + T_2}{2}$$

As is apparent in Figures 1 (1, p.177) and 2 (9), the approximated outlet air temperature is very close to the actual design outlet air temperature. The approximation for the outlet air temperature can be used as a first guess for a detailed design or may be considered as the optimum outlet air temperature in a rule-of-thumb design. When the approximation is used, air flow rate will be within $\pm 10\%$ of the optimum design in most cases.

Water is evaporated during the wet cooling process. For each $10^{O}F$ drop in water temperature, approximately 1.0% of the treated water evaporates (3, p.757). The water flow rate is not strictly constant. However, the evaporation rate is small and is commonly neglected, yielding the following energy balance (11, p.589):

$$\frac{L}{G} = \frac{h_2 - h_1}{C_p\,(T_1 - T_2)}$$

The outlet air is usually saturated at the outlet air temperature. The enthalpies of the inlet and outlet are as found from a table or chart of psychometric data. Therefore, by specifying an outlet air temperature, the water-air flow rate ratio is fixed.

c. Tower Characteristic

The tower characteristic $\left(\frac{Ka\overline{V}}{\overline{L}}\right)$ is determined from the Merkel equation:

$$\frac{Ka\overline{V}}{\overline{L}} = \int_{T_2}^{T_1} \frac{dT}{h_{sa} - h_a}$$

The right-hand side of the Merkel equation is difficult to integrate directly because $h_{sa} - h_a$ is difficult to express explicitly in terms of T. However, it can be graphically integrated or solved by Simpson's rule (see sample calculation). The most commonly used computer solution is the Tchebycheff method (2, p.12-13). A nomograph is also available for estimation of the tower characteristic (2, p.12-14).

D. Loading Factor

The loading factor (L), specific water flow rate or water flow rate density is the recommended water flow rate per unit of tower cross-sectional area (base area; B). Through experience with various types of fill, optimum loading factors have been determined as a function of design wet-bulb temperature, range and approach. For difficult cooling jobs (large range and/or close approach), a low loading factor is required and visa versa. Two graphical methods are presented for determining the loading factor (Figures 3, 4, 5 and Figure 6).

The loading factors determined from these two methods agree well, but are lower than the loading factors used with presently-used fills. Methods for determining modern loading factors are not available, however. The back-calculated value of Ka (100) was determined from the available, older loading factors. Therefore, the available, older loading factors must be used when calculations are performed using the recommended value of Ka.

E. Tower Dimensions

The required fill height (Z) is equal to the specific volume ($\overline{V}$) and is determined from the tower characteristic:

$$Z = \overline{V} = \left(\frac{Ka\overline{V}}{\overline{L}}\right)_{Calc} \times \frac{\overline{L}}{Ka}$$

The required base area or cross-sectional area (B) is:

$$B = L/\overline{L}$$

A larger loading factor will result in both a smaller tower height and in a smaller base area. The fill volume (V) is:

$$V = B \times Z$$

F. Water Consumption

Wet cooling towers consume water in three major ways: evaporation, drift and blowdown. The evaporation rate (E) is approximately 1.0% of the water flow rate per each 10^{o}F of cooling range (3, p.757). Drift (D) refers to water which leaves the cooling tower entrained in the exiting air and is approximately 0.2% of the water flow rate (3, p.757). As water evaporates, solids and chemicals concentrate in the cooling water. Blowdown (B_d) is the water removed from the system, and replaced by fresh water, to prevent solids/chemicals buildup in the cooling water. Blowdown is expressed as a percentage of the evaporation rate and depends upon the solids/chemicals concentration which can be tolerated in the process in which the cooling water is being used and the solids/chemicals concentration of the makeup water. Blowdown is usually about 20% of the evaporation rate. Makeup (M) water is required to replace the consumed water:

$$\text{Water Consumption Rate} = M = E + D + B_d$$

$$E = .001 \times R \times L$$

$$D = .002\ L$$

$$B_d \cong .2\ E$$

$$M = (.0012\ R + .002)\ L$$

The evaporation rate can also be determined from a mass balance around the air stream:

$$E = (H_2 - H_1)\ G$$

In this case,

$$M = 1.2G\ (H_2 - H_1) + .002\ L$$

This second method of determining the evaporation rate is more accurate than the first method, but the first method is easier to use because it involves fewer variables.

G. Power Requirements

Pump power (P_p) is determined from the following equation:

$$P_p = \frac{L \times H_p}{1.98 \times 10^6 \times \eta}$$

The head (H_p) is difficult to determine. Dickey (8, p.12) recommends a 75 foot head. However, the power requirement obtained with a 75 foot head is two to two and a half times greater than the requirement obtained from other approximations. McKelvey and Brooke (1, p.178) present the following approximation:

Fill Height (Ft.)	hp/1000 gpm	hp/10^6 lbs/hr
20-24	7	14
24-28	8.5	17

The power requirements obtained from the above approximations tend to be low. From an analysis of data, a good estimate of the pump head has been found to be:

$$H_p = Z + 10$$

This equation is convenient and allows the tower height to affect the pump power requirements.

Fan power requirements can be estimated from the following approximations presented by McKelvey and Brooke (1, p.178):

Fill Height (ft.)	hp/1000 gpm	hp/10^6 lbs/hr
20-24	14	28
24-28	12	24

Fan power requirements can also be estimated from the volume of moist air moved by the fan. For forced draft towers, the volume of the inlet air is used in the calculation. For induced draft towers, use the volume of the exit air. Assuming that one hp is required for each 8,000 actual cubic feet of air per minute (acfm) moved by the fan (1, p.178), the fan power is approximated from the following formula:

$$P_F = \frac{F}{8{,}000},$$

where P_F = Fan Power (hp)

F = Air Flow Rate (acfm)

$$F = \frac{(1 + H_t)}{60\ \rho_{mix,t}} G$$

H_t = Air Humidity @ t (lbs H_2O/lb dry air)

G = Air Flow Rate (lbs air/hr)

$\rho_{mix,t}$ = Density of Moist Air @ t (lbs/ft^3)

$$= \frac{(1 + H_t)(\rho_{w,t} \times \rho_{a,t})}{(\rho_{w,t} + \rho_{a,t})}$$

$\rho_{a,t}$ = Density of Dry Air @ t (lbs/ft^3)

$$= \frac{42.6439}{t + 460}$$

$\rho_{w,t}$ = Density of Water Vapor @ t (lbs/ft^3)

$$= \frac{26.6525}{H_t(t + 460)}$$

The formulas for calculation of $\rho_{a,t}$ and $\rho_{w,t}$ are derived from the ideal gas law. The assumption of 1 hp/8,000 acfm is consistent with data reported by Research - Cottrell (11).

The total power is obtained by adding fan power and pump power. McKelvey and Brooke (1, p.179) present a method for approximating total power requirements from range, appoach, design wet-bulb temperature and water flow rate.

H. Cost Estimation

Zanker (12) has derived an equation for the estimation of cooling tower construction cost:

$$\$_{1967} = \frac{Q}{C \times A + 39.2R - 586}$$

where $\$_{1967}$ = 1967 dollars

Q = Total Heat Load (Btu/hr)

R = Range (^{o}F)

A = Approach (^{o}F)

$$C = \frac{279}{\left[1 + 0.0335\,(85 - t_{wb})^{1.143}\right]}$$

t_{wb} = Design Wet-Bulb Temperature

Multiplication of 1967 dollars by 2.7 $\left[1.08^{13}\right]$ will approximately correct to 1980 dollars.

Dickey (9) presents a method for estimation of cooling tower construction costs. From analysis of 39 cooling towers built by

Marley Co, cooling tower construction cost was found to be \$14.45 (1978 dollars) per cooling tower unit (TU). The number of tower units in a given cooling tower are found as follows:

$$TU = \text{Water Flow Rate (gpm)} \times \text{Rating Factor.}$$

The rating factor is a measure of the cooling job difficulty. For the 39 Marley Co. cooling towers, a linear relationship (correlation coefficient = .9844) was found between the Rating Factor and the tower characteristic (TC):

$$\text{Rating Factor} = .9964 \times TC - .3843$$

A method of estimating cooling tower cost from the tower characteristic is therefore:

$$\$_{1978} = \frac{14.45}{500} L \; (.9964 \times TC - .3843)$$

To convert from 1978 dollars to 1980 dollars, multiply by 1.4. From the tower characteristic equation, a separate construction cost is obtained for each design (outlet air temperature); whereas, the Zanker equation yields only one cost for each set of design conditions.

I. Sample Calculation (Example 1)

Design a cooling tower to cool 120,000 gpm (60×10^6 lbs/hr) from 119^oF to 89^oF, when the wet-bulb temperature is 75^oF. Also, estimate water consumption rate, power requirements and construction cost. Assume Ka equals 100.

Solution:

Estimate outlet air temperature (t_2) and L/G:

$$t_2 = \frac{T_1 + T_2}{2} = 104^oF$$

$h_2(104^oF, \text{sat'd}) = 79.31$ Btu/lb dry air

$h_1(t_{wb} = 75^oF) = 38.62$ Btu/lb dry air

$$\frac{L}{G} = \frac{h_2 - h_1}{C_p\,(T_1 - T_2)} = \frac{79.31 - 38.62}{(1.0)(119 - 89)} = 1.36$$

Calculate tower characteristic by the Simpson rule. Divide range into five equal sections of 6^oF each, then

$$\Delta h_a = \frac{79.31 - 38.62}{5} = 8.139 \text{ Btu/lb air.}$$

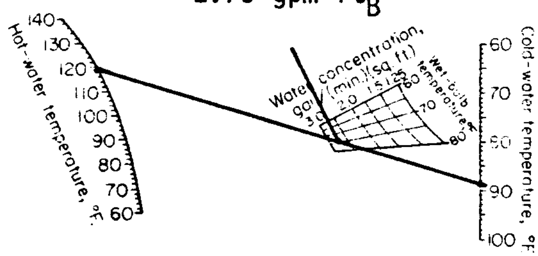
T	h_{sa}	h_a	$h_{sa}-h_a$	$(h_{sa}-h_a)^{-1}$
89	54.85	38.62	16.23	.0616
95	63.34	46.76	16.58	.0603
101	73.58	54.90	18.68	.0535
107	85.59	63.04	22.55	.0443
113	99.74	71.18	28.57	.0350
119	116.50	79.31	37.19	.0269
				.2816
			Ave =	.0469

$$\left(\frac{K_a\overline{V}}{\overline{L}}\right)_{calc} = R \times \left(\frac{1}{h_{sa} - h_a}\right)_{ave} = 30 \times .0469 = 1.408$$

Determine the loading factor:

$$\overline{L} = 2.75\ \text{gpm/ft}_B^2$$

$$= 1375\ \text{lbs/hr/ft}_B^2$$



Determine the tower dimensions:

$$Z = \left(\frac{K_a\overline{V}}{\overline{L}}\right)_{calc} \times \frac{\overline{L}}{K_a} = \frac{1.408 \times 1375}{100} = 19.4\ \text{ft.}$$

$$B = L/\overline{L} = 120{,}000/2.75 = 43{,}636\ \text{ft}_B^2$$

$$V = B \times Z = 844{,}740\ \text{ft}_{fill}^3$$

Estimate pump power:

$$H_p = Z + 10 = 29.4 \text{ ft.}$$

$$P_p = \frac{L \times H_p}{1.98 \times 10^6 \, \eta} = \frac{(60 \times 10^6) \times 29.4}{(1.98 \times 10^6) \times (.80)} \cong 1100 \text{ hp}$$

Estimate fan power (Induced draft; $t_2 = 104^{o}F$):

$$\rho_{a,t} = \frac{42.6439}{t_2 + 460} = .0756 \text{ lbs/ft}^3$$

$$H_t = .0491 \text{ lbs } H_2O/\text{lb air}$$

$$\rho_{w,t} = \frac{26.6525}{(t + 460) \times H_t} = .9624 \text{ lbs/ft}^3$$

$$\rho_{mix,t} = \frac{(1 + H_t)(\rho_{a,t})(\rho_{w,t})}{(\rho_{a,t} + \rho_{w,t})} = .0735 \text{ lbs/ft}^3_{mix}$$

$$G = L \left(\frac{L}{G}\right)^{-1} = 44.23 \times 10^6 \text{ lbs/hr}$$

$$F = \frac{(1 + H_t)\, G}{60 \, \rho_{mix,t}} = 10.52 \times 10^6 \text{ acfm (ft}^3\text{/min)}$$

$$P_F = \frac{F}{8000} = 1300 \text{ hp}$$

Total power requirement is approximately 2400 hp.

Estimate construction cost:

$$\$_{1978} = \frac{14.45}{500} L \quad (.9964 \times TC - .3843) = 1.77 \times 10^6 \text{ dollars}$$

Estimate water consumption:

$$M = (.0012R + .002)\ L$$

$$= 3.36 \times 10^6 \text{ lbs/hr}$$

$$= 6720 \text{ gpm}$$

III. SIMULATION CALCULATIONS

Cooling towers operate most of the time at conditions different than their design conditions. Prediction of cooling tower performance at various conditions is important in the optimization of power plant cycles. A change in one operation parameter changes the performance of the tower. Simulation calculations are required to determine the new operating state.

The parameter which changes most often is the ambient wet-bulb temperature. A change in the wet-bulb temperature will affect the range and the approach of the tower, but the tower characteristic $\left(\frac{Ka\overline{V}}{\overline{L}}\right)$ will remain unchanged. The air flow rate is also frequently changed by reducing the fan speed. When the air flow rate is changed, not only is the approach affected, but the tower characteristic is also changed. Water flow rate also affects the tower characteristic. A change in the inlet water temperature does not affect the tower characteristic, but does change the approach and can change the range. The tower characteristic $\left(\frac{Ka\overline{V}}{\overline{L}}\right)$ is a function of L/G by the following relationship:

$$\frac{Ka\overline{V}}{\overline{L}} = C\left(\frac{L}{G}\right)^{M}$$

or

$$\log_{10}\left(\frac{Ka\overline{V}}{L}\right) = M\log_{10}\left(\frac{L}{G}\right) + \log_{10} C$$

The slope (M) is approximately equal to -0.6 for most conditions (13, p.2.8). If one point on the line is known (for instance, $\frac{Ka\overline{V}}{\overline{L}}$ at design L/G), the intercept ($\log_{10}C$) can be calculated and C determined. Once C is determined, the tower characteristic can be determined for other L/G. The above equation is accurate within the following range: $\frac{1}{2} \times \left(\frac{L}{G}\right)_{des} < TC_{des} < \frac{3}{2} \times \left(\frac{L}{G}\right)_{des}$.

Mathematical solutions to simulation problems must be solved by trial-and-error. Two examples of arithmetically solved simulation problems are given below. The solution of simulation problems using Cooling Tower Institute Performance Curves (14) is given by the Cooling Tower Institute (15, 16).

Example 2

A cooling tower is designed to cool 8950 gpm (4.475×10^6 lbs/hr) from 110^oF to 84^oF (R = 26) at a wet-bulb temperature of 69^oF ($A = 15^oF$) with a tower characteristic of 1.49 (L/G = 1.30 and $t_2 = 97.3^oF$). When the wet-bulb temperature drops to 60^oF, what will T_1 and T_2 be for L/G and range held constant?

Solution (Trial and Error)

$$h_1\ (t_{wb} = 60^oF) = 26.46 \text{ Btu/lb air}$$

Guess 1: $T_2 = 75^oF$ ($T_1 = 101^oF$)

$$h_2 = h_1 + \frac{L}{G}(T_1 - T_2) = 60.26 \text{ Btu/lb air}$$

$$h_{sa}\ (T = 75, \text{ sat'd}) = 38.62 \text{ Btu/lb}$$

$$h_{sa}\ (T = 101, \text{ sat'd}) = 73.58 \text{ Btu/lb}$$

$$\left(\frac{Ka\overline{V}}{\overline{L}}\right)_1 = \int_{T_2}^{T_1} \frac{dT}{h_{sa} - h_a} = 2.34$$

Guess 2: $T_2 = 80^oF$ ($T_1 = 106^oF$)

$h_2 = 60.26^oF$

$$\left(\frac{K_a\overline{V}}{\overline{L}}\right)_2 = 1.43$$

Guess 3: Interpolation - $T_2 = 79^oF$ ($T_1 = 105^oF$)

$h_2 = 60.26^oF$

$$\left(\frac{K_aV}{\overline{L}}\right)_3 = 1.56$$

Guess 4: Interpolation - $T_2 = 79.5^oF$ ($T_1 = 105.5^oF$)

$h_2 = 60.26^oF$

$$\left(\frac{K_a\overline{V}}{\overline{L}}\right)_4 = 1.49$$

$\therefore T_2 = 79.5^oF$ and $T_1 = 105.5^oF$

Example 3

Consider the above cooling tower. If the fan rpm is reduced 50% (an approximately 40% reduction in air flow rate), what will T_1 and T_2 be for $t_{wb} = 60^oF$ and the range held constant?

<u>Solution</u>: (Trial and Error)

$$\left(\frac{L}{G}\right)_{op} = \frac{1}{.60}\left(\frac{L}{G}\right)_{des} = \frac{1.30}{.60} = 2.17$$

$$C = \left(\frac{Ka\overline{V}}{\overline{L}}\right)_{des} \times \left(\frac{L}{G}\right)_{des}^{-M}$$

$$= 1.7442$$

$$\left(\frac{Ka\overline{V}}{\overline{L}}\right)_{op} = C \times \left(\frac{L}{G}\right)_{op}^{M} = 1.096$$

$$\therefore 1.096 = \int_{T_2}^{T_1} \frac{dT}{h_{sa} - h_a}$$

Solve by trial and error as in Example 2.

$T_2 = 89.6^{o}F$ and $T_1 = 115.6^{o}F$.

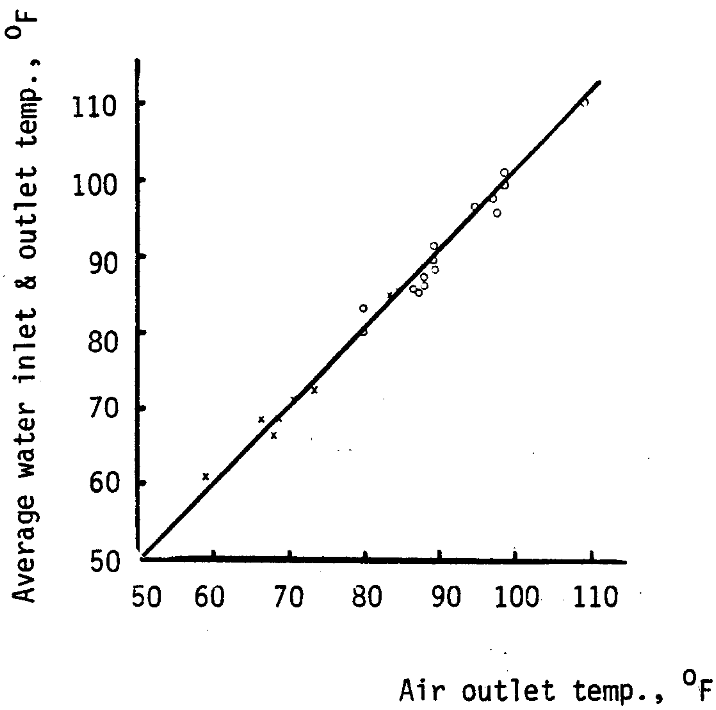


FIGURE 1: Average of water inlet and outlet temperature vs. Design air outlet [From McKelvey & Brook (1, p.177)]

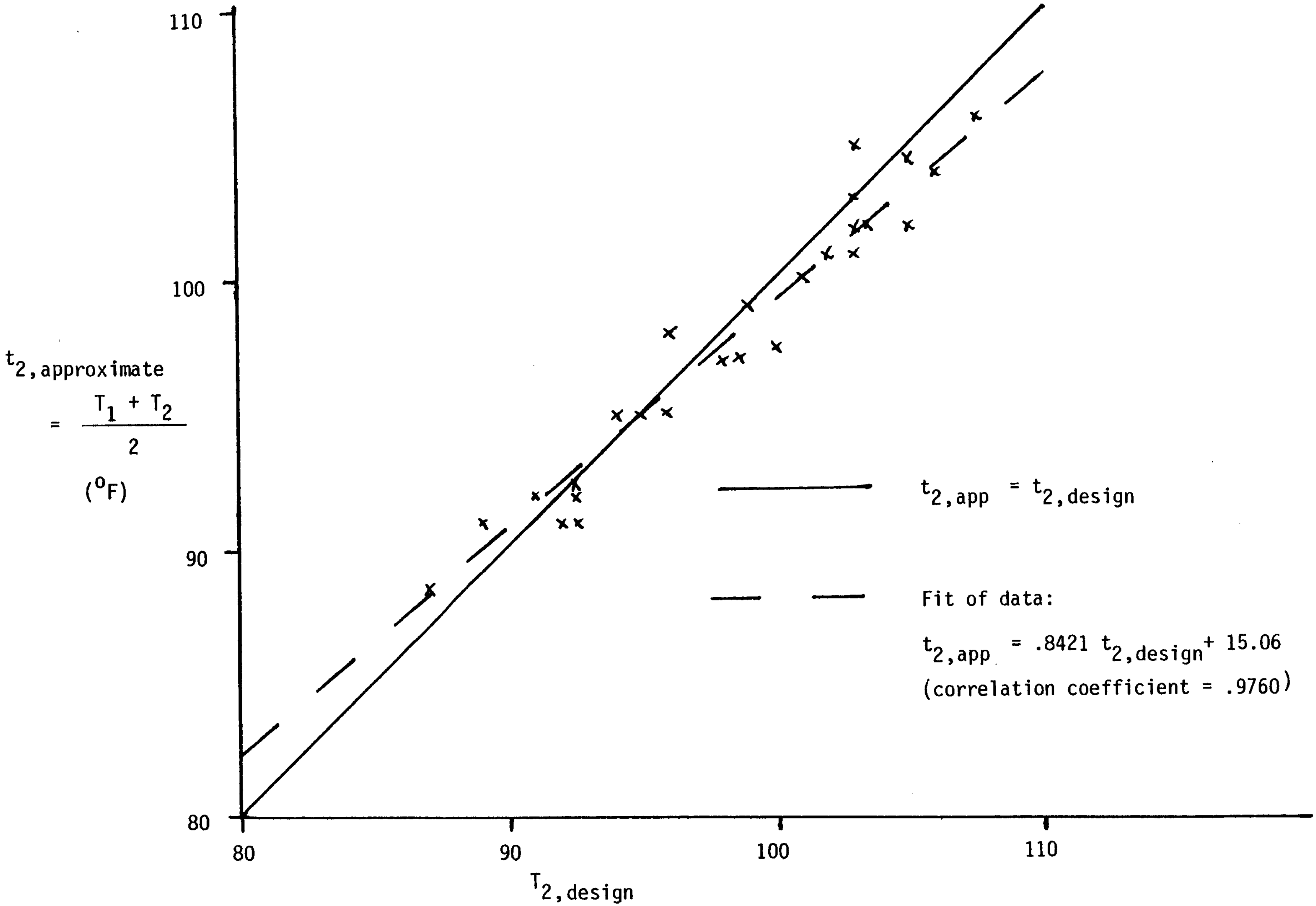


FIGURE 2: Approximate outlet air temperature vs. Design outlet air temperature
[From an analysis of Marley Co. data (9)]

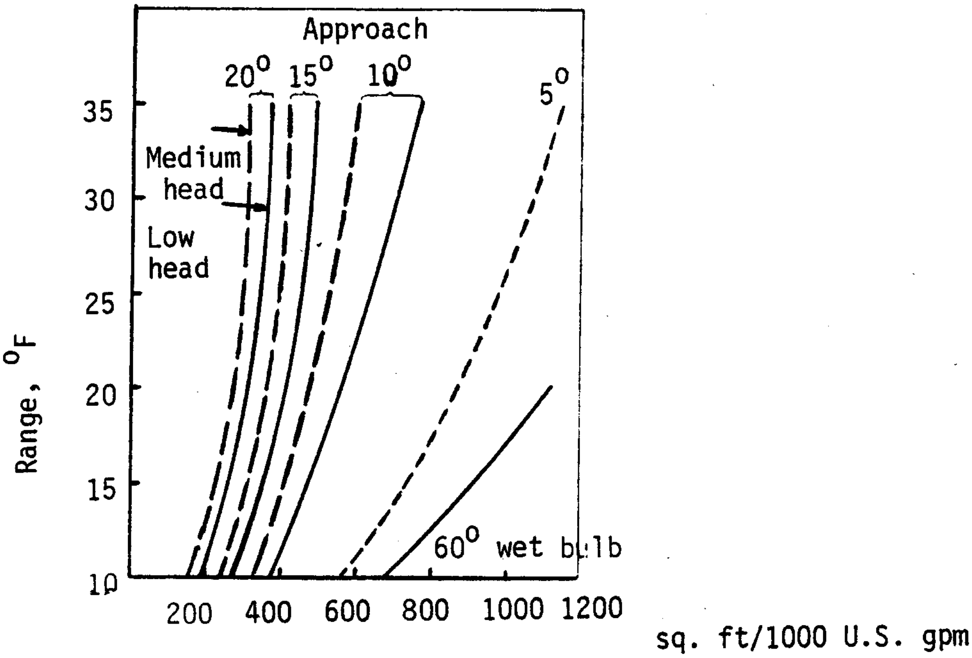


FIGURE 3: 60°F wet bulb - Design chart for mechanical draft towers

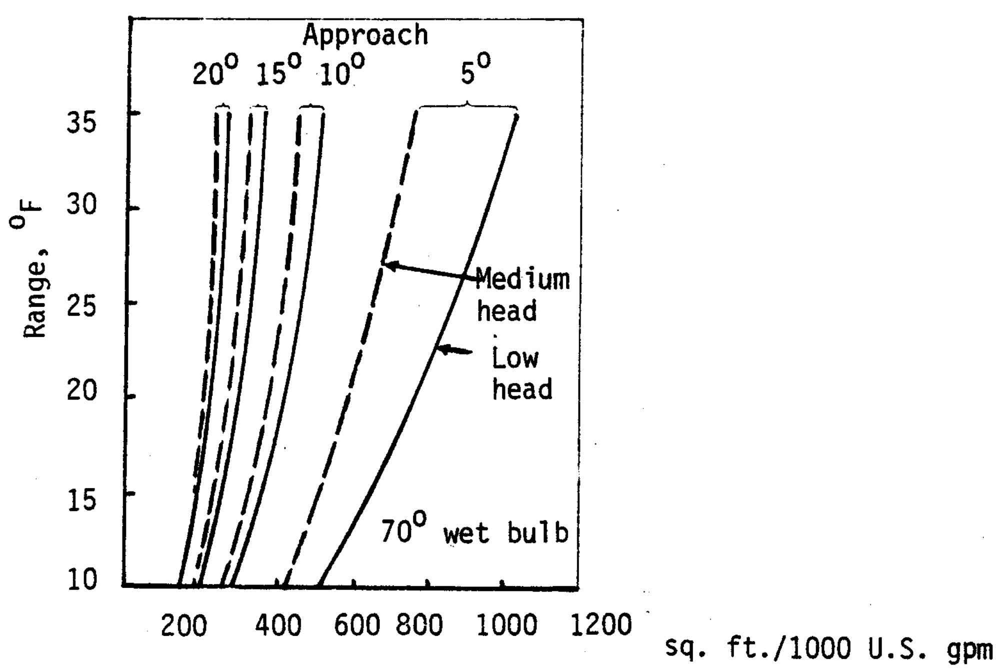


FIGURE 4: 70° wet bulb - Design chart for mechanical draft towers

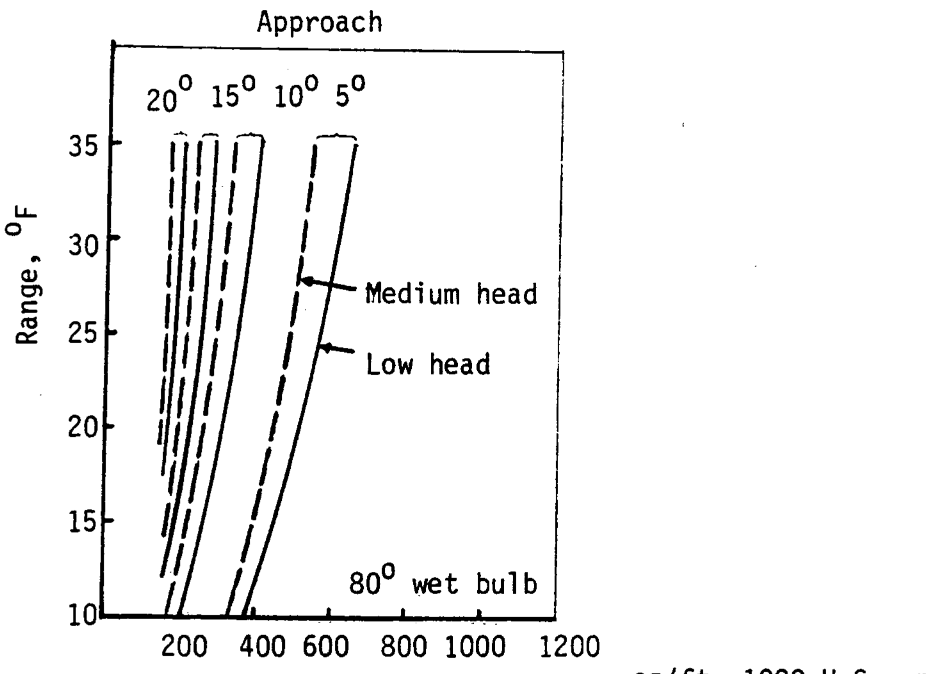


FIGURE 5: 80° wet bulb - Design chart for mechanical draft towers

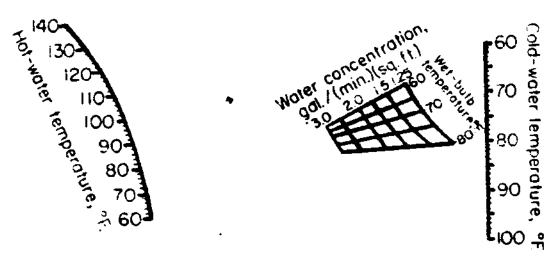


FIGURE 6: Loading Factor (Water Concentration) Determination Chart for Induced-Draft Cooling Towers

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