Cooling towers size and cost as related to their location
by Layne Russell Zimmerman

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Mechanical Engineering
Montana State University
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Abstract:
State of the art methods for sizing evaporative, dry, and hybrid (combination of evaporative and dry) cooling towers were examined. Where appropriate these methods were used to develop computer programs for sizing cooling towers. The well known Merkel theory was incorporated in programs to size wet cooling towers. Where methods had not been developed for sizing cooling towers, as for dry cooling towers, heat exchanger-theory was used to develop methods of sizing these towers. A program was developed to size the dry cooling towers. This program together with the programs for sizing evaporative cooling towers were used to size hybrid cooling towers.

The relationships between the size of the cooling towers and the total power plant cost were examined, and the important variables stressed.

It was shown that accurate methods of sizing the cooling system must be used in conjunction with all plant designs to do any cost optimizations. Due to the complex interaction at a proposed plant site of the weather, water availability, and construction costs, the methods provided herein are indispensable in determining the overall picture when planning a new site.

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ABSTRACT

State of the art methods for sizing evaporative, dry, and hybrid (combination of evaporative and dry) cooling towers were examined. Where appropriate these methods were used to develop computer programs for sizing cooling towers. The well known Merkel theory was incorporated in programs to size wet cooling towers. Where methods had not been developed for sizing cooling towers, as for dry cooling towers, heat exchanger theory was used to develop methods of sizing these towers. A program was developed to size the dry cooling towers. This program together with the programs for sizing evaporative cooling towers were used to size hybrid cooling towers.

The relationships between the size of the cooling towers and the total power plant cost were examined, and the important variables stressed.

It was shown that accurate methods of sizing the cooling system must be used in conjunction with all plant designs to do any cost optimizations. Due to the complex interaction at a proposed plant site of the weather, water availability, and construction costs, the methods provided herein are indispensable in determining the overall picture when planning a new site.

NOMENCLATURE

English Letter Symbols

a	Effective heat transfer area per unit volume in wet tower
A	Total heat transfer area
A_c	Heat exchanger minimum free flow area
A_f	Total fin area of one side of a heat exchanger
A_{fr}	Total frontal area of a heat exchanger
C	Flow-stream capacity rate ($\dot{m}\ C_p$)
C_c	Flow-stream capacity rate of cold-side fluid
C_h	Flow-stream capacity rate of hot-side fluid
C_{min}	Smaller of the two quantities C_c and C_h
C_{max}	Larger of the two quantities of C_c and C_h
C_p	Specific heat at constant pressure
D_h	Hydraulic diameter of internal passage
F_G	Connection factor to log-mean rate equation
f	Flow friction factor within heat exchanger
G	Mass flow rate of air
G_a	Mass flow rate of air per unit area
h	Enthalpy
h	Film conductance coefficient for convection heat transfer
K	Overall heat transfer coefficient from water to air
K_G	Contraction loss coefficient for flow at heat exchanger entrance

K_e	Expansion loss coefficient for flow at heat exchanger exit
K_g	Coefficient of sensible heat transfer per unit of effective heat transfer area
K_L	Coefficient of total heat transfer per unit of effective heat transfer area
K'	Coefficient of latent heat transfer per unit of effective heat transfer area
k	Thermal conductivity
L	Mass flow rate of water in evaporative tower
L_a	Mass flow rate per unit area in evaporative tower
L_e	Evaporation mass flow rate
l	Fin length from root to center
M	Molecular weight
$\dot{m}$	Mass flow rate
P	Pressure
Q_L	Rate of latent heat trnasfer
Q_s	Rate of sensible heat transfer
Q_w	Total heat transfer rate from water
q	Heat transfer rate
R	Gas constant
$\bar{R}$	Universal gas constant
r_h	Hydraulic radius
s	Air speed
T	Temperature of air

t	Temperature of water
U	Overall thermal conductance
V	Volume
v	Specific volume
v_m	Average specific volume

GREEK LETTER SYMBOLS

α	Ratio of total transfer area on one side of the exchanger to total volume
δ	Fin thickness
ε	Heat exchanger effectiveness
η_o	Total surface temperature effectiveness
ϕ	Relative humidity
σ	Ratio of free-flow area to frontal area (A_c/A_{fv})
μ	Viscosity
ρ	Density
ω, ω'	Humidity ratio or absolute humidity of air and air-water film respectively

DIMENSIONLESS GROUPINGS

N_{Nu}	Nusselt number ($h4r_h/k$)
N_{Re}	Reynolds number ($4r_h m/\mu A_c$)
N_{st}	Stanton number ($hAc/\dot{m}C_p$)
N_{Pr}	Prandtl number ($\mu C_p/k$)

N_{tu}	Number of transfer units of a heat exchanger
NTU_L	Number of transfer units (KaV/L)
NTU_G	Number of transfer units (kaV/G)

All primes refer to water-air interface properties.

INTRODUCTION

Disposing of vast quantities of waste heat, due to the increasing demand for electrical power has become a major enterprise, and a major concern for groups of technically oriented people. Environmentalists are concerned about how power generating plants and cooling tower design will affect the site selection. Divisions within the state governments are given the job of regulating the power companies and satisfying the majority of concerned people. Because of these concerned people, the Montana Department of Natural Resources and Conservation funded a research assistantship in the area of cooling tower sizing. This thesis presents work done by the research assistant in the areas of evaporative, and dry, water to air cooling and combinations of these, all of which require cooling towers as the instruments of heat dissipation.

The evaporative or wet cooling towers are composed of a series of tower units each of which is independent and capable of functioning alone. The different types and methods of sizing each are discussed in Chapter 2. The methods presented enable a technical person to size evaporative cooling towers for any ambient temperature, barometric pressure, and relative humidity. The different designs of evaporative towers each have their own particular advantages and limitations. The counterflow tower is more efficient thermally, but the crossflow tower has less air resistance, and will develop higher air flow velocities than in the counterflow tower using equal fan horsepower.

Due to the availability of water and construction costs, dry cooling towers are presently not feasible in many areas. A larger dry cooling tower is required to do a job comparible to an evaporative cooling tower. This is expensive, but the cost of water is rising also. Large generating plants cannot be constructed in water short areas with evaporative cooling towers, because of the makeup water required. If water supplies become more limited, the difficulty of power plant site selection will intensify. A 1000 MW fossil-fuel fired generating plant will require around 4000 gpm of make-up water due to evaporation alone if evaporative cooling is employed. Construction of large dry cooled generating plants has been done only in England thus far. The knowledge gained from these show that it is possible to construct plants of almost any size using dry cooling. Chapter 3 presents methods of sizing dry cooling towers to meet year round ambient weather conditions and additional information on analyzing heat exchangers under off-design conditions.

Various chemical production plants in dryer sections of the country have used dry cooling for years. One reason for this is that it is easier to use air for cooling remote parts of the plant where piping of water can be inconvenient and costly.

A more modern answer to the cooling problem is the use of the hybrid or wet/dry cooling tower. If plume control or water availability are major concerns, they may be alleviated with the combination cooling. This idea is delt with in Chapter 4. The use of dry or wet/dry cooling

becomes practical as the selection of construction sites becomes more difficult due to necessary environmental protection restrictions, and limited water availability.

Chapter 2

WET COOLING TOWERS

The general development of wet cooling tower theory is attributed to Merkel and is called the Merkel formulation. It is presented by Gurney and Cotter (1) in terms of water droplets with air flowing past them as shown in Figure 2.1.

It is assumed that the air in contact with the water is moving very slowly with respect to the droplet, and is saturated. The bulk air moving past the water always has a lower vapor pressure, and the vapor at the higher pressure in the water film attempts to equalize with the vapor pressure in the air. This results in a flow of water vapor, evaporation, and the water looses heat. This combined with direct conduction provides the necessary heat transfer.

To begin, assume the water droplet to be falling through a counterflow tower, with water fed into the top, and air into the bottom as shown in Figure 2.2. Wet cooling tower calculations are normally based on the pack plan area which contains the area around the free droplets as well as the contact area of the packing. Equations can be developed to find the sensible heat transfer, and the latent heat transfer. The sensible heat transfer is that due to conduction, and the latent heat transfer is that due to evaporation.

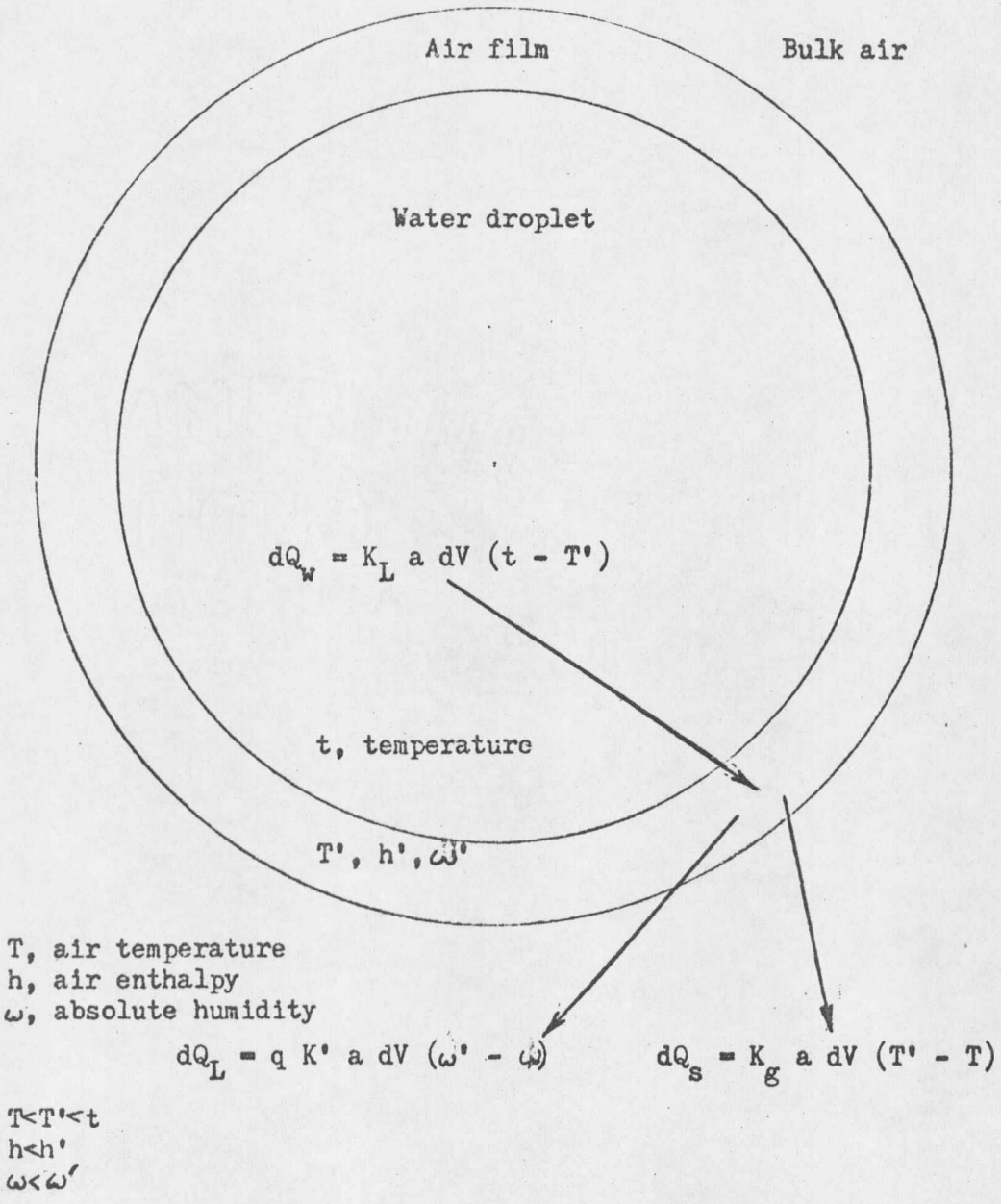


Figure 2.1 Heat transfer from water droplet.

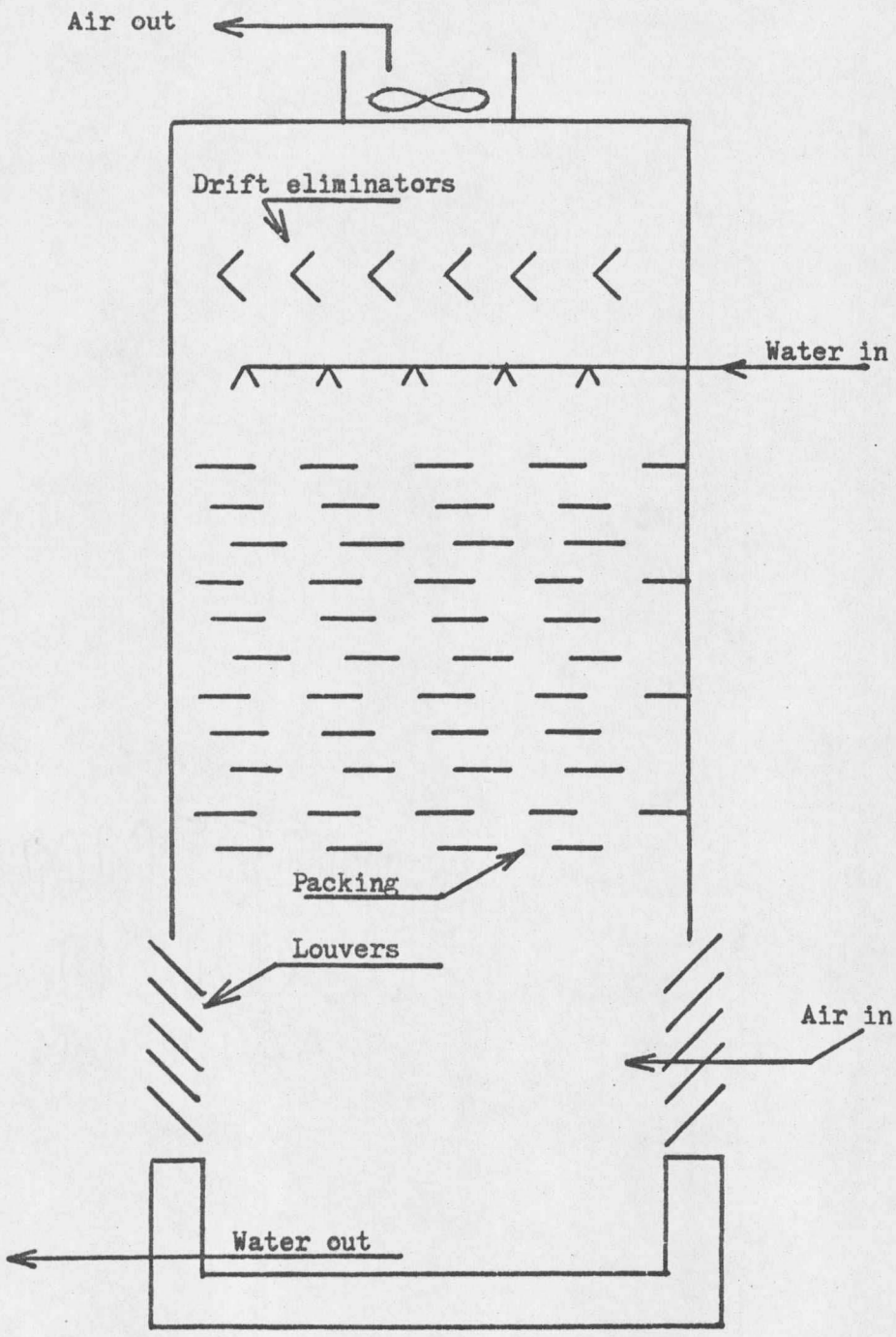


Figure 2.2 Counterflow induced-draft tower draws air directly counter to the water flow.

Rate Equations

The total heat transfer from the water to the interfacial film is

$$dQ_w = L\ dt = K_L\ a\ dV\ (t - T') \tag{2.1}$$

The rate equation for the sensible heat from the interfacial air to the main air stream is

$$dQ_s = K_g\ a\ dV\ (T' - T) \tag{2.2}$$

The mass transfer from the film to the air is of the form

$$dm = K'\ a\ dV\ (\omega' - \omega) \tag{2.3}$$

Let q = latent heat of evaporation, and the rate equation for the latent heat transfer becomes

$$dQ_L = qdm = q\ k'\ a\ dV\ (\omega' - \omega) \tag{2.4}$$

Mass and Energy Balances

The rate of mass leaving the water equals the rate of humidity increase in the air

$$dm = G\ d\omega \tag{2.5}$$

The heat lost by the water equals the heat gained by the air

$$G\ dh = L\ dt \tag{2.6}$$

The enthaply of moist air per pound mass of dry air is

$$h = C_{pa}(T-T_o) + \omega\ [q + C_{pv}(T-T_o)] \tag{2.7}$$

Differentiating this equation gives:

$$dh = C_{pa} + \omega\, C_{pv} dT + [q + C_{pv}(T-T_o)]d\omega \qquad (2.8)$$

T_o is the datum temperature that enthalpy is based on, and the water vapor is considered to be an ideal gas.

Collecting terms in equation 2.8 gives

$$dh = (C_{pa} + \omega C_{pv})dt + [q + C_{pv}(T-T_o)]d\omega \qquad (2.9)$$

On a 1bm of dry air basis the specific heat of moist air is

$$C_{pm} = Cp_a + C_{pv}\omega \qquad (2.10)$$

Recognizing the specific heat of moist air in equation 2.9 and substituting gives

$$dh = C_{pm}\, dT + [q + C_{pv}(T-T_o)]\, d\omega \qquad (2.11)$$

Substituting equation 2.11 into the energy balance equation 2.6 yields

$$L\, dt = G\, C_{pm}\, dT + [q + C_{pv}(T - T_o)]G\, d\omega \qquad (2.12)$$

L dt = sensible heat + latent heat

Equating the sensible heat rate equation 2.2 to the sensible heat in the energy balance equation 2.12 gives

$$dQ_s = K_g\, a\, dV(T' - T) = G\, C_{pm}\, dT \qquad (2.13)$$

Equating the mass transfer rate equation 2.3 to the mass balance equation 2.5 gives

$$dm = K' a\, dV(\omega' - \omega) = G\, d\omega \qquad (2.14)$$

Use is now made of the Lewis relationship (2), a dimensionless number which for air/water mixtures is $K_g/(K'C_{pm}) \doteq 1$. Since Lewis presented the relationship in 1922 work has been done which shows that for air/water mixtures the value is nearer 0.9, but for simplicity this small deviation from unity is ignored.

Substituting $K_g = K' C_{pm}$ in equation 2.13 yields

$$dQ_s = K' C_{pm}\, a\, dV(T' - T) = G\, C_{pm}\, dT \qquad (2.15)$$

The conduction coefficient is eliminated from the sensible heat equation in this expression.

Now substituting equation 2.15 for the sensible heat part of the energy balance in equation 2.12 and equation 2.14 for G dH in equation 2.12 gives

$$Ldt = K'a\, d\, v[C_{pm}(T' - T) + [q + C_{pv}(T-T_o)](\omega' - \omega)] \qquad (2.17)$$

By eliminating the temperature of the bulk air and the temperature of the air film, equation 2.17 can be expressed in terms of temperature of the water, enthalpy of the air and air film, and absolute humidity of the air and air film.

From equation 2.8 the enthalpy of moist air is

$$h = C_{pa}(T - T_o) + \omega[q + C_{pv}(T - T_o)]$$

Substituting $C_{pm} = C_{pa} + \omega C_{pv}$ and solving for $C_{pm}T$ yields

$$C_{pm} T = h + C_{pa} T_o - \omega(q - C_{pv} T_o) \tag{2.18}$$

Similarly for the interfacial air film,

$$C_{pm} T' = h' + C_{pa} T_o = \omega'(q - C_{pv} T_o) \tag{2.19}$$

Substituting equation 2.18 and 2.19 into the last form of the energy balance equation 2.17 one obtains an equation which reduces to

$$L\, dt = K' a\, dV(h' - h) + C_{pv}(\omega' - \omega) \tag{2.20}$$

Since the air in wet cooling towers is always nearly saturated the second term is small and is omitted. Using the basic energy balance, equation 2.6 leaves

$$L\, dt = K' a\, dV(h' - h) = G\, dh \tag{2.21}$$

This equation refers to the interfacial air film. By assuming that the air film is at the same temperature as the water, the film coefficient becomes an overall coefficient, and the means is provided for finding the enthalpy of the moist air in the interfacial film.

Equation 2.21 can be integrated to yield

$$NTU_G = (K\ a\ V)/G = \int_{h_1}^{h_2} [dh/(h' - h)] \tag{2.22}$$

and

$$NTU_L = (K\ a\ V)/L = \int_{t_2}^{t_1} [dt/(h' - h)] \tag{2.23}$$

Given the plan area of the packing and the volumes, the height of the packing from equation 2.22 is

$$H = G/(K\ a\ A) \int_{h_1}^{h_2} [dh/(h' - h)] \qquad (2.24)$$

where A = pack plan area

H = pack height

Neglecting the small mass change due to evaporation, integration of equation 2.6 gives

$$L\ C_{pw}\ (t_1 - t_2) = G(h_2 - h_1) \qquad (2.25)$$

Heat lost by water = Heat gained by air

or

$$h_2 = h_1 + L\ C_{pw}(t_1 - t_2)/G \qquad (2.26)$$

This shows that the enthalpy change of the air is linear with respect to temperature. The temperature change of the water is also linear. The enthalpy change of the air in the interface is not linear however.

The enthalpy potential or driving force for the heat transfer is shown in Figure 2.3.

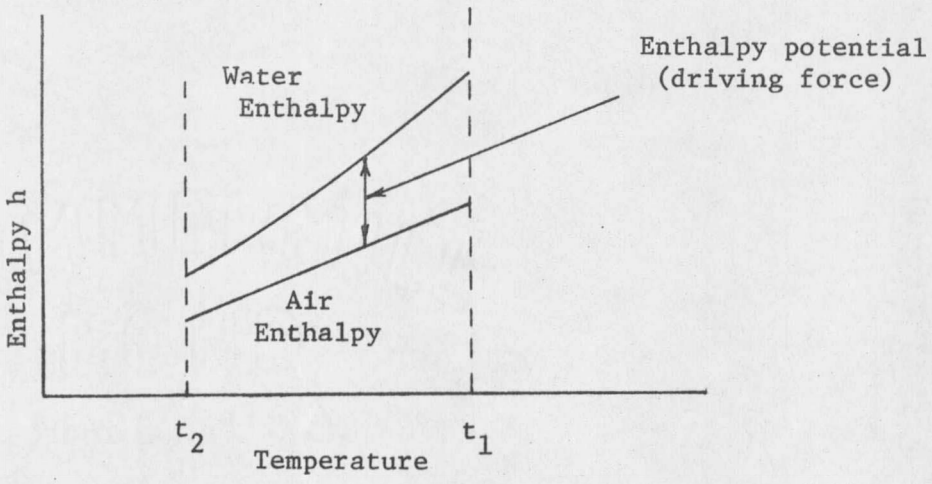


Figure 2.3 Enthalpy Potential

The simplifications used so far are as follows:

Evaporation was ignored where the change in the flow rates were concerned. The resistance to mass transfer through the interface was ignored. The latent heat of evaporation was taken as constant, and the specific heat of water was taken as unity. The Lewis relationship was also taken as unity.

Explanation of the tower coefficient found from equation 2.23 , modifications to Merkel's theory, and methods of integration are presented by Baker and Shryock (3).

When doing the calculations for a theoretical set of conditions, one obtains a number of transfer units, NTU, which gives a measure of the degree of difficulty of the cooling. The NTU corresponding to

these calculations is called the 'required' coefficient. The same calculations applied to a given tower yields what is called the 'available' coefficient of the tower involved.

The required coefficient is given as a function of range, approach, wet-bulb temperature, and the water flow rate that will be cooled at these temperatures. The remaining parameter is the air flow rate or the L/G ratio.

The driving force for a given set of conditions, and variable air rate is shown in Figure 2.4. Equation 2.6 shows that L/G is the slope of the line corresponding to the enthalpy of the air.

$$dh = L/G\ dt$$

Assuming an infinite air rate, L/G = 0, the air enthalpy line becomes the horizontal line AB. This maximizes the driving force, and minimizes the tower coefficient or NTU. Increasing the L/G ratio increases the slope of the air enthalpy line until the limiting case is reached where the air enthalpy line, AD, becomes tangent to the water film enthalpy line, and the required NTU is essentially infinite.

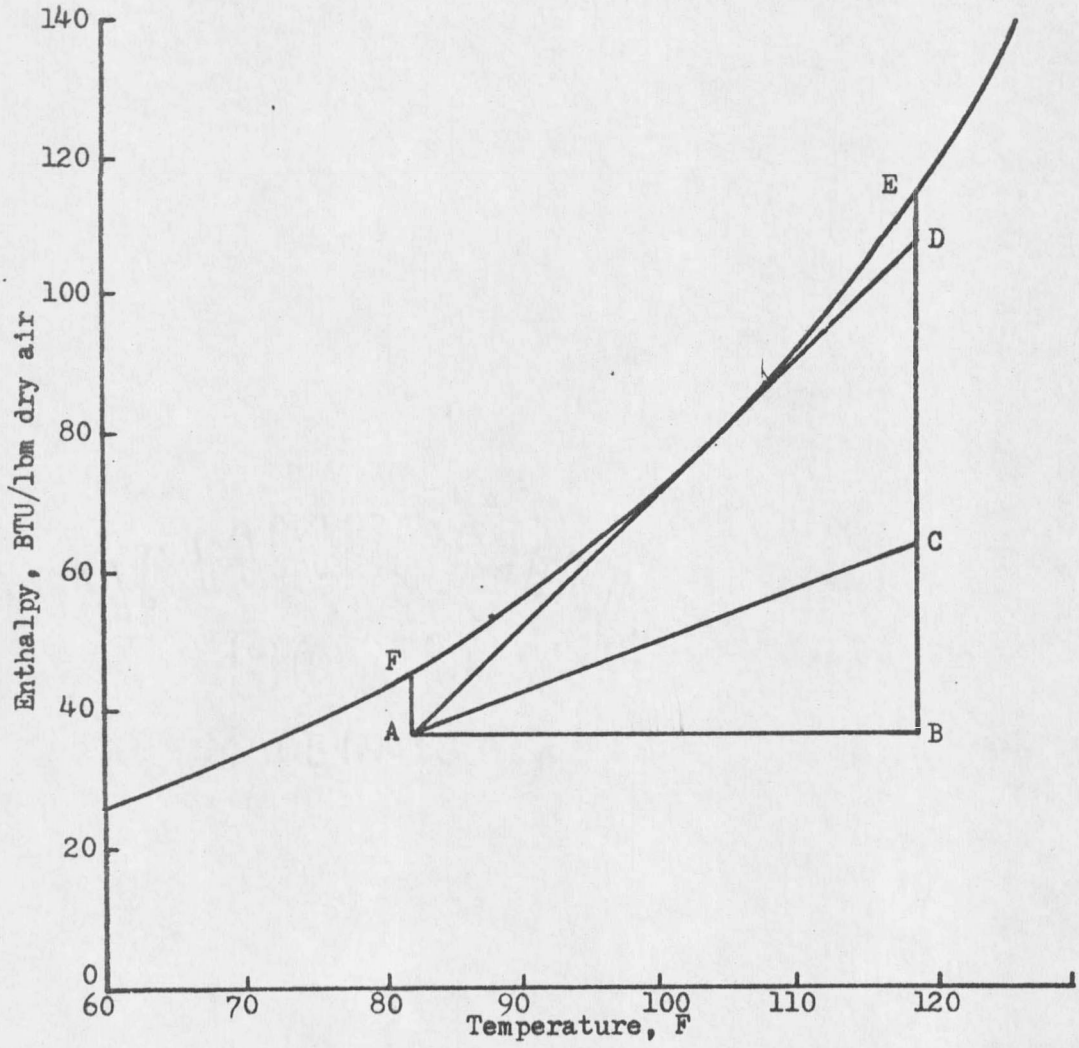


Figure 2.4 Driving force for heat transfer as controlled by the L/G ratio.

Counterflow Integration

The counterflow tower theoretically has no temperature or enthalpy variations with respect to the length or width of the tower, and the calculations are done on a square foot of plan area basis. L and G are used as mass flow rates per unit of plan area. The volume, V, is the volume per square foot of plan area.

A given tower is usually designed for a fixed water rate. A tower can handle only a certain amount of water per square foot of plan area. If the flow of water is too large, the water runs off the packing in sheets and not droplets which greatly decreases the surface area for evaporation and heat transfer. At the same time the pressure drop requirements for a large air flow rate through the tower can become larger than the fans are capable of delivering. If the flow of water is too small channeling may occur. This has an effect similar to the sheeting action except in channeling the water runs off the slats in the packing in rivulets, and this also decreases the heat transfer area. For certain packings this minimum flow is about 70 gal/(hr ft^2) of tower plan area (1). The air rate is more easily varied, but usually air speed is from 400-600 ft/min. From this it follows that the L/G ratios for normal operation lie between 0.8 and 1.3. A closer look at the results of the integrations will show why these constraints occur.

The potential difference function to be integrated, shown in

Figure 2.5 generates a smooth curve. Such a curve lends itself to accurate integration using Simpson's rule with a small number of subdivisions. When six of these subdivisions are used the accuracy will be as good as any of the quadrature or tabular methods commonly used, and the labor involved is no greater.

Most of the calculations involved are straightforward, but some such as finding the temperature of the air exiting the tower require an iterative solution. The next examples are included to illustrate the methods of comparing sizes, water and air flow rates, and one method of handling the iterative solution when it becomes necessary.

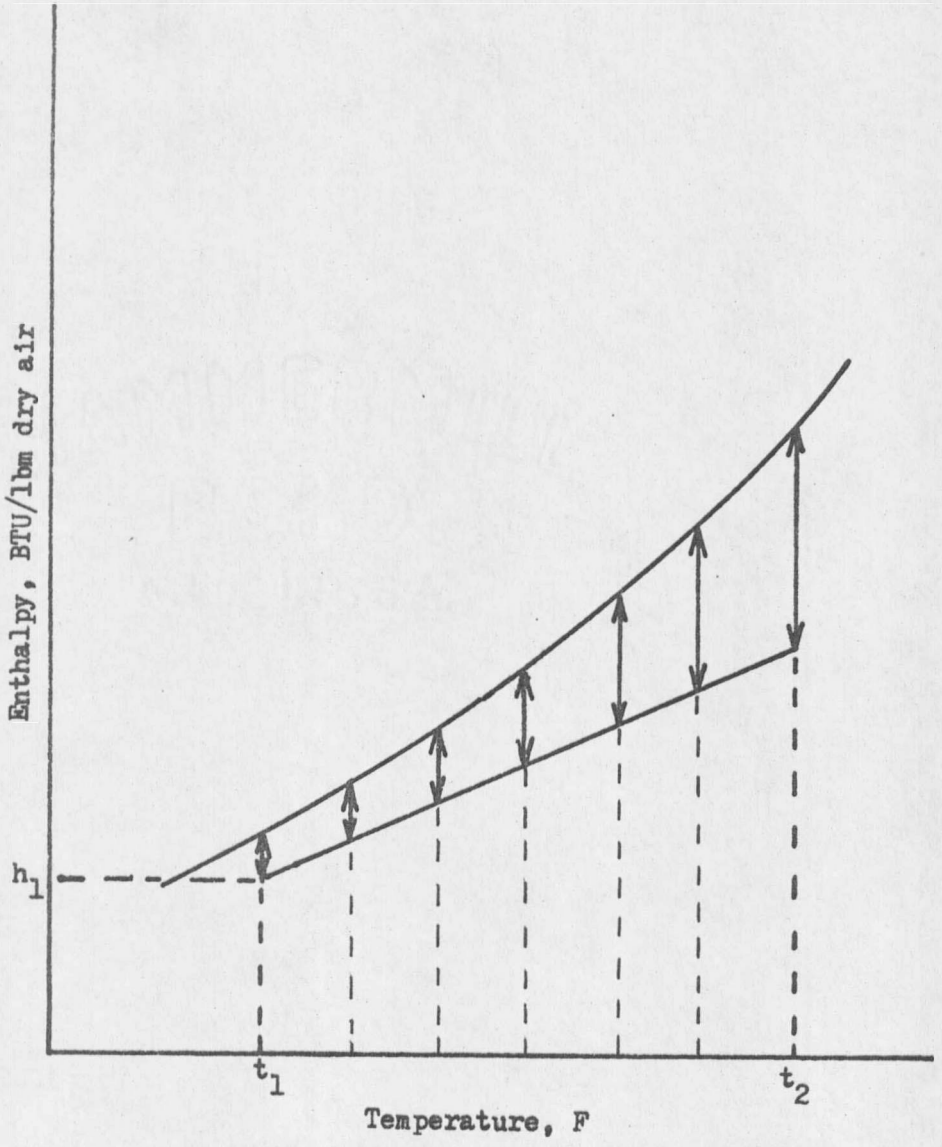


Figure 2.5 Subdivisions for the numerical integration of the driving force.

Example 1

Consider a 1000 MW power plant on a warm day with the following conditions known:

Atmospheric pressure = P = 30.0 in. Hg

Relative humidity = ϕ = 0.28

Dry bulb temperature = T_{db} = 94.0°F

Wet bulb temperature = T_{wb} = 68.0°F

Temperature of the water entering the tower = t_2 = 108.0°F

A power plant will have an efficiency in the neighborhood of 40%. Therefore the heat to be expelled from such a plant will be 60% of the total heat entering the water in the boiler.

The total heat entering the circulating water is Q.

$$0.40 \times Q = 10^9 \text{ watts}$$

$$Q = 10^9/0.4 \text{ watts}$$

The heat to the tower is

$$q = 0.60Q = 0.60(10^9)/0.4 \text{ watts}$$

$$q = 0.60(10^9)/0.4 \text{ watts} \times 3.414(\text{BTU/hr})/\text{watt}$$

$$q = 5.121(10^9) \text{ BTU/hr}$$

Ignoring the water vapor present, density of the air entering is

$$\rho = P/RT \tag{2.27}$$

$$\rho = 30.0 \times 70.53/[53.35(94.0 + 459.69)]$$

$$\rho = 0.0716 \text{ lbm/ft}^3$$

Assuming the water flow is 70 gal/(hr ft^2)

$$L_a = L/A = 70 \text{ gal/(hr ft}^2\text{)} \times 8.337 \text{ lbm/gal}$$

$$L_a = 583.6 \text{ lbm/(hr ft}^2\text{)}$$

The number of transfer units as a function of L/G ratios for fixed conditions is required. The necessary water flow rate is found in terms of the range from

$$q = L\, C_{P_w} \text{ (Range)} \tag{2.28}$$

and letting $C_{pw} = 1.0$ BTU/(lbm°R)

$$L = q/\text{Range} \tag{2.29}$$

Then, a range of 10°F gives

$$L = [5.129(10^9)[/10.0 = 5.121(10^8) \text{ lbm/hr}$$

The plan area necessary to handle the water is

$$\text{Area} = L/(L/a) = 5.121(10^8)/583.6$$

$$A = 8.775(10^5) \text{ ft}^2$$

The temperature of the water leaving the tower is

$$t_1 = t_2 - \text{Range} \tag{2.30}$$

$$t_1 = 108.0 - 10.0 = 98.0°F$$

and the approach is

$$\text{Approach} = t_1 - T_{wb} = 98.0 - 68.0 = 30°F$$

By letting the air flow rate vary the integrals are evaluated to generate a curve relating NTU to the L/G ratio for a range of 10°F. Letting L/G = 0.1 the air rate becomes

$$G = L/0.1$$

$$G = 5.121(10^9) \text{ lbm/hr}$$

and the air speed is

$$s = G/(\rho \times \text{Area}) \tag{2.31}$$

$$s = 5.121(10^9)/(0.0716 \times 8.775(10^5)) = 8.152(10^4) \text{ ft/hr}$$

Referring to Figure 2.5, h_1 is the enthalpy of saturated water vapor and air at the wet bulb temperature of 68°F.

$$h_1 = C_{pa}(T-T_o) + \omega(q + C_{pv}(T - T_o)) \tag{2.7}$$

If the reference temperature, T_o, is taken as zero degrees Farenheit, then

$$h_1 = 32.43 \text{ BTU/lbm}$$

From the energy balance

$$h_2 = h_1 + L(t_2 - t_1)/G \tag{2.26}$$

$$h_2 = 32.43 + 0.1(108 - 98)$$

$$h_2 = 33.43 \text{ BTU/lbm}$$

The rate of water evaporation is given by

$$L_e = G(\omega_2 - \omega_1) \tag{2.32}$$

where ω_1 and ω_2 are the humidity ratios of the entering and exiting air, respectively. The humidity ratios can be computed from

$$\omega = 0.622 \frac{\phi P_g}{P - \phi P_g} \tag{2.33}$$

The entering humidity ratio can be found directly from the information available, but the exiting humidity ratio is a function of the saturated water vapor pressure at the temperature of the air leaving the tower. This temperature is not known, but the enthalpy of the exiting air is. Since the functions relating the two are complex it is easiest to find the temperature by an iterative technique. The relative humidity is assumed to be close to 0.95. Two reasonable temperatures are chosen, and the enthalpies corresponding to each are found. A linear interpolation is done with the known enthalpy to estimate correct temperature. Since there will be some error after the first interpolation the process can be repeated until the desired accuracy is obtained. For this example case, the temperature of the air exiting and the associated saturation pressure are

$$T_a = 69.9°F; \quad P_g = 0.363 \text{ lbf/in}^2$$

Therefore from equation (2.33)

$$\omega_2 = 0.01491 \text{ lbm}_v/\text{lbm}_a$$

and

$$\omega_1 = 9.965(10^{-3})\ lbm_v/lbm_a$$

The water evaporation rate found using equation 2.32 is

$$L_e = 5.121(10^9)(0.01491 - 9.965(10^{-3}) = 2.532(10^7)\ lbm/hr$$

Since the air enthalpy and the water temperature are linear with respect to each other they may be easily divided into six equal parts. The enthalpy of a water vapor and air mixture is found corresponding to each of the water temperatures. These correspond to the top line in Figure 2.5. Evaluating the integral gives the number of transfer units.

$$NTU = KaV/L = \int_{t_1}^{t_2} [dt/(h' - h)]$$

$$NTU = 0.2303$$

Choosing different values for L/G changes the value of G, h_2, and the NTU giving the lower of the curves in Figure 2.6

Assuming the heat load must always be transferred, the water flow rate must change to correspond to each different range. The water flow rate is inversely proportional to the Range.

$$L = q/Range \qquad (2.28)$$

Since the water flow per square foot of plan area is fixed the plan area is directly proportional to the water flow rate and inversely proportional to the Range. When the range is large the water leaving

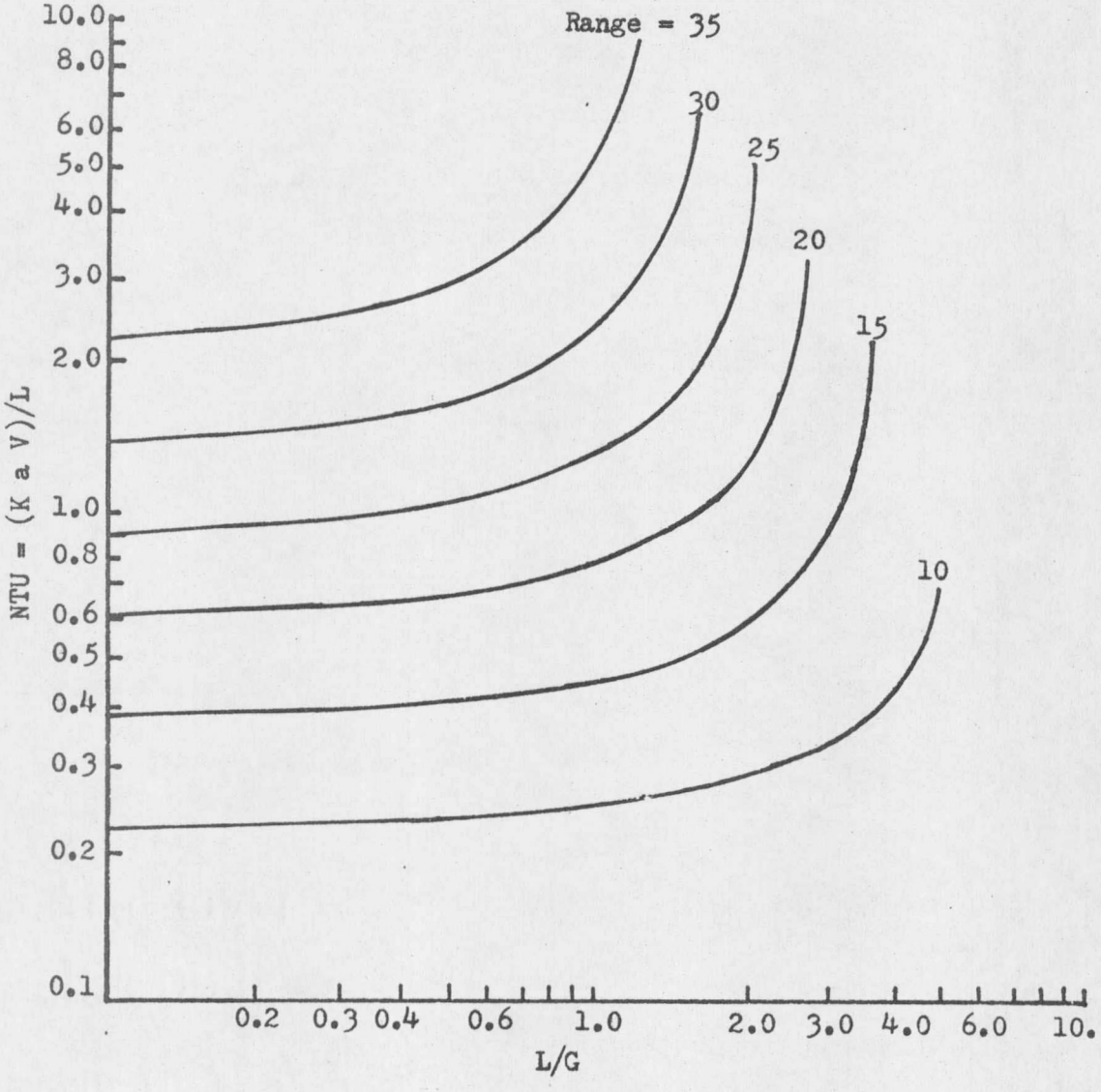


Figure 2.6 Results of the integration using Simpson's rule for Example 2.1.

the tower approaches more closely the wet bulb temperature of the air. Figure 2.6 shows that a linear decrease in the approach gives an exponential increase in the tower size. The limits on the lower ranges come from the plant itself. The condenser requires larger flow rates for water delivered at higher temperatures to obtain the same temperature for the water leaving the condenser. If the temperature of the water leaving the condenser is allowed to rise, the back pressure to the turbines rises, and the efficiency of the plant drops. The size of the condenser then limits the amount of water that can be handled, which in turn limits the minimum range which is acceptable.

The reason for using L/G ratios in the neighborhood of 1.0 now becomes apparent. One might be tempted to increase the air rate, and decrease the L/G ratio to make the tower smaller. Figure 2.6 shows that the reduction in the tower size is minimal for L/G ratios less than 0.6. This reduction is offset by the increase in the fan power necessary to delivery the larger mass flow rates of air.

Once a cooling tower has been designed to do a job on a particular day, it is still not known how a particular tower will function under different conditions. The number of transfer units provided by a tower under a given set of conditions is called the available coefficient. Baker and Shyrock (3) show that the available coefficients, plotted on the same NTU versus L/G graph used for the required coefficient, form a straight line as shown in Figure 2.7. The points where the available

coefficient line intersect the required coefficient curves give the tower operating conditions for various air rates. A set of curves such as these can be formed for any conditions of interest.

The available coefficient line in Figure 2.7 suggests a relation between the available coefficient, and the L/G ration of the form

$$NTU \sim \left(\frac{L}{G}\right)^n \qquad (2.33)$$

The exponent n varies within a range of about -0.35 to -1.1 and the average value is between -0.55 and -0.65.

Crossflow Cooling

A typical 1000 MW fossil-fueled generating plant would require an evaporative cooling tower measuring approximately 70 ft wide, 50 ft high and 350 to 400 ft long. This tower would circulate about 150,000 to 400,000 gpm from which 1-1/2 to 2-1/2% would be lost to evaporation, 0.05 to 0.2% would be lost to drift, and 0.5 to 3% would be lost in blowdown. In typical locations the plant would be designed to operate on days when the wet bulb temperature ranged from 66 to 82°F. The tower would operate with a 12 to 30°F cooling range, and a 10 to 20°F approach.

For crossflow cooling the theory developed is still applicable, but the methods of application differ. Unlike the counterflow type of tower the conditions through a horizontal section of the tower are not constant. The air enters the side of the tower at constant enthalpy,

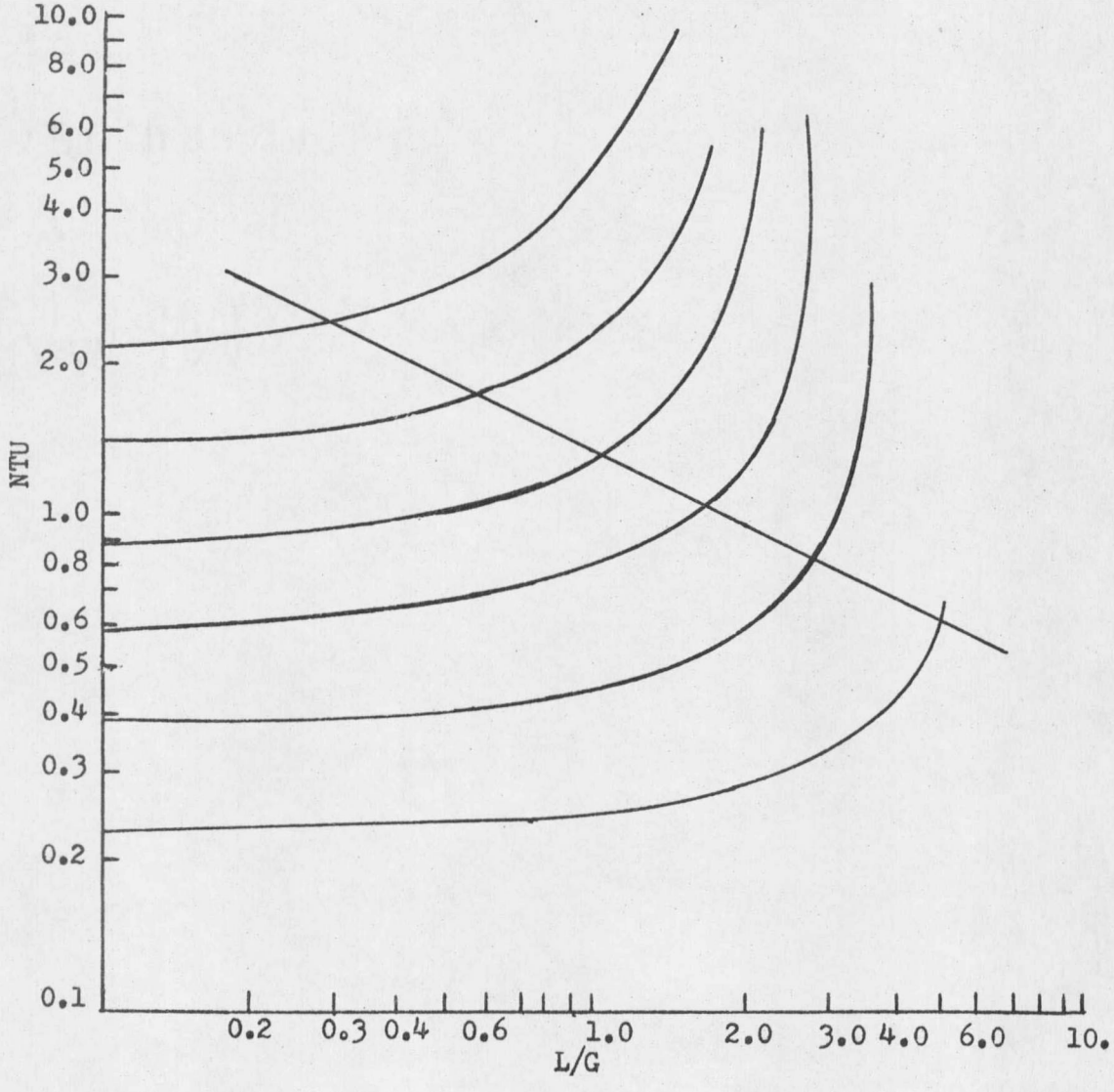


Figure 2.7 Example of an operating line superimposed on the required coefficient curves.

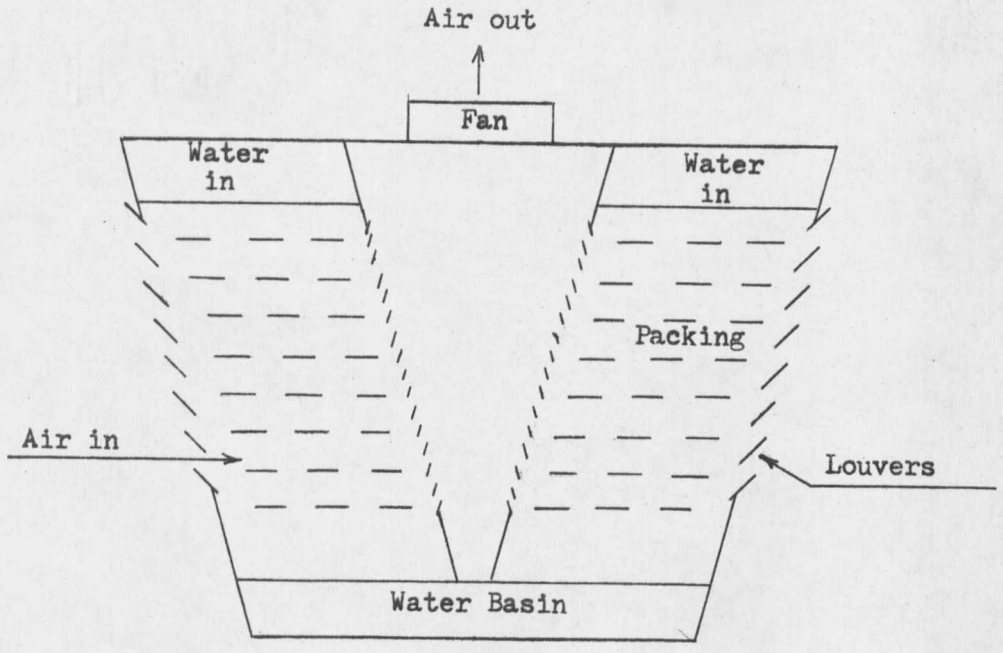


Figure 2.8 Crossection of the crossflow cooling tower.

as shown in Figure 2.9, but as it passes through the packing it becomes less effective for cooling. This results in isotherms in the water as shown.

In sizing, the tower the governing equation to be integrated is equation 2.21.

$$L\ dt = K\ a\ dV(h' - h) = G\ dh$$

Since the fluid properties in the tower are constant in one dimension, it is possible to work with a section of unit length. By assuming that L and G are on a unit area basis the governing equation becomes

$$L_a\ dt\ dx = K\ a\ dy\ dx(h' - h) = G_a\ dh\ dy \qquad (2.33)$$

Integrating yields:

$$\frac{KaZ}{L_a} = \int_{t_1}^{t_2} \frac{dt}{h'-h} \qquad (2.34)$$

and,

$$\frac{KaW}{G_a} = \int_{h_1}^{h_2} \frac{dh}{h'-h} \qquad (2.35)$$

More than one method for handling the integration has been presented. One was presented in 1962 by Zamuner (4). In doing the integrations by approximate methods it is possible to increment either on the water temperature or on the transfer units. Incrementing the transfer units is the most popular method, and the one used by Zamuner. By letting $1/\alpha$ be the fractional part of the transfer unit assumed, the integral (2.34) becomes

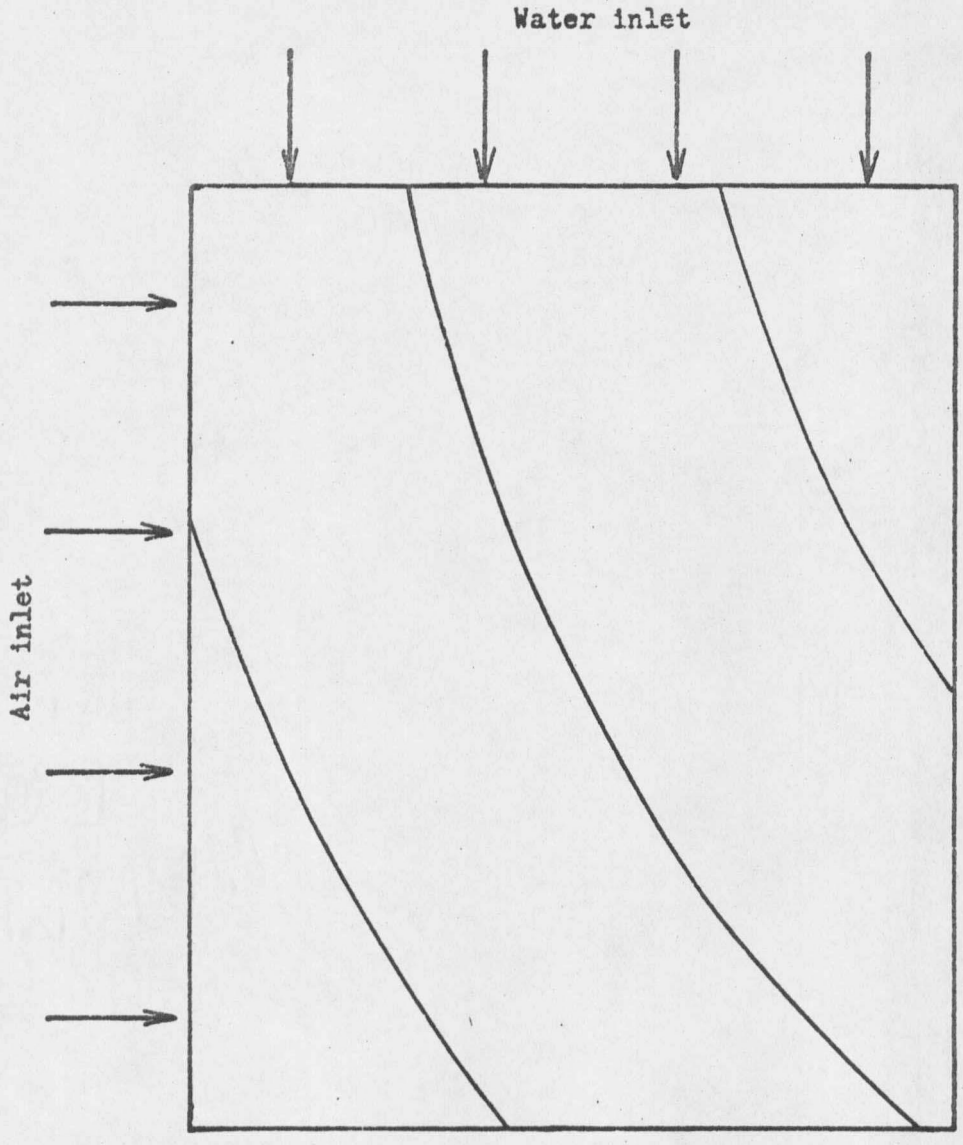


Figure 2.9 Isotherms of the water temperature for a crossflow cooling tower.

$$\frac{1}{\alpha} = \int_{t}^{t + \Delta t} \frac{dt}{h' - h} \qquad (2.36)$$

or approximately:

$$\frac{1}{\alpha} = \frac{\Delta t}{h' - h} \qquad (2.37)$$

where $(h' - h)$ is the mean driving force in the increment Δt.

Equation 2.33 shows that the energy balance for an incremental volume is

$$L_a \, dt \, dx = G_a \, dh \, dy \qquad (2.38)$$

Since the temperature of the water, and the enthalpy of the air entering the tower are known, all the information necessary to do the integration is known at one corner of the packing. Using equation 2.37, the number of fractional vertical transfer units necessary to obtain the desired temperature change can be found by doing the integration step by step down the side of the packing. By letting the horizontal step length equal the vertical step length, $\Delta x = \Delta y$, equation 2.38 becomes approximately

$$L_a \Delta t = G_a \Delta h \qquad (2.39)$$

For an assumed fractional transfer unit the integration can now be done for incremental volumes. Starting in the upper corner where all the entering conditions are known, Δt can be found from equation 2.37. For the horizontal air enthalpy change equation 2.39 is used. Proceeding down and across, a grid is formed as shown in Figure 2.10. The

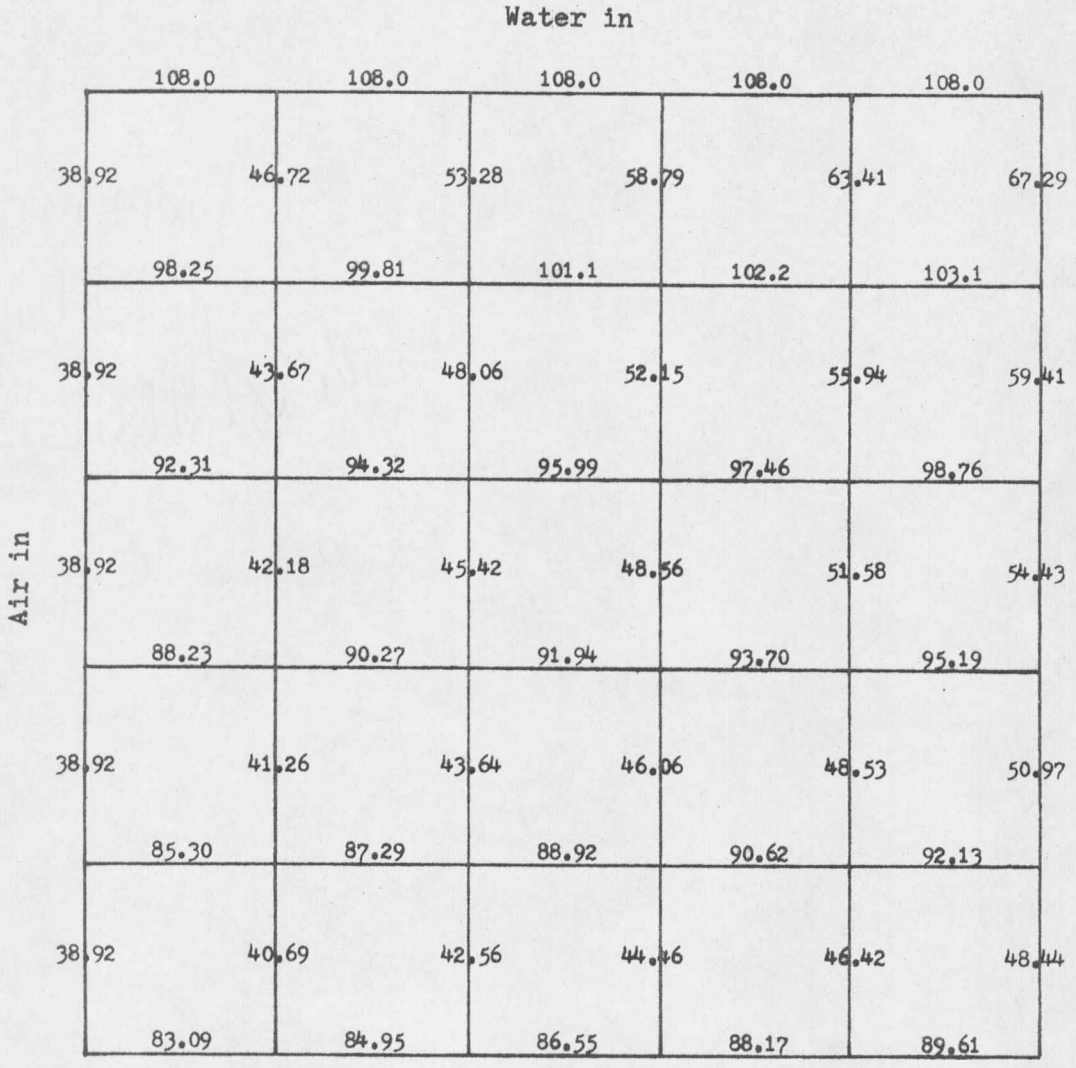


Figure 2.10 Results of crossflow integration using Zamuner's method under conditions stated in Example 2.1 and letting L/G = 0.8 in each unit.

performance of different sizes and shapes of towers can now be predicted. By choosing an arbitrary number of fractional units in height and width, not necessarily equal, the temperature of the water leaving the tower can be found by averaging the temperatures of the water leaving each increment at the base of the section chosen. Each increment in Figure 2.10 represents 0.2 transfer units. If the tower were 0.8 transfer units tall, and 0.6 transfer units wide the average exiting water temperature would be (85.30 + 87.29 + 88.92)/3 = 87.2°F. If the tower were 1.0 by 1.0 transfer units the average exiting water temperature would be (83.09 + 84.95 + 86.55 + 88.17 + 89.61) = 86.5°F. Once the number of transfer units necessary for the desired performance is found, the actual dimensions of the tower can be found from:

$$\frac{K\ a\ Z}{L_a} = NTU \qquad (2.40)$$

Ka is found experimentally for different types of packing, and is usually a function of the water flow rate.

It is important to carefully consider the flow rates when doing the integration by this method. The ratio of the flow rates, L/G, is kept constant within each volume increment, but the total flow rates are proportional to the number of increments in each direction. The overall ratio of flow rates for the cross section is dependent upon the shape of the tower, or the number of horizontal and vertical increments.

Zamuner did some work correlating average exiting water temperatures, and air enthalpies for towers with different numbers of horizontal and vertical transfer units.

In 1960 Baker and Shryock (3) presented a slightly different method of evaluating the same integrals. Their method differs from the previous one in two aspects. Baker and Shryock let each volume increment be similar in shape to the whole cross section. This keeps the L/G ratio the same in each increment as it is in the cross section as a whole. The steps down then equal the steps across regardless of shape.

The second difference in the methods is the way the mean driving force is calculated within the volume increment. The simplest method is the one used by Zamuner. It is assumed that the entering potential difference remains constant through the unit volume. Since the enthalpy of the saturated air at the entering water temperature is greater than it is at the temperature leaving the increment, and the enthalpy of the air entering the increment is less than it is leaving, the driving force computed by Zamuner's method will be larger than the true value in each increment. This could result in under designing the tower. Referring to equation 2.34, it is seen that the driving force is the reciprocal of the enthalpy difference. Therefore a method of finding the driving force would be to average the recriprocals of the entering and leaving conditions. But this method corresponds to

parallel flow, and the value obtained would be lower than the true value. The true value is between those found by the previous two methods. If the enthalpy differences leaving and entering are averaged, and then the recriprocal found the value will be smaller than that found by the first method, and larger than found by the second method. This value should be closer to the true value than the value found in the other two methods. Baker and Shryock recommend this method, and it is the method used in the next example problem, and the corresponding program in Appendix II. Since the enthalpies leaving the volume increment are not known, they must be assumed, and the solution becomes an iterative solution.

Example 2.

Consider the same power plant and the same conditions as treated in Example 1.

Atmospheric pressure = P = 30.0 in Hg

Relative Humidity = 0.28

Dry bulb temperature = T_{db} = 94.0°F

West bulb temperature = T_{wb} = 68.0°F

Temperature of the water entering the tower = t_2 = 108.0°F

The heat to be expelled by the tower is

$$q = 5.121(10^9) \text{ Btu/hr}$$

Assuming the water flow rate is, as before

$$L_a = L/A = 583.6 \text{ lbm/(hr ft}^2\text{)}$$

The number of transfer units necessary for the cooling can be plotted as a function of the L/G ratio for a fixed cooling range. For a range of 10°F the water flow rate is

$$L = q/\text{Range} = 5.121(10^8) \text{ lbm/hr}$$

The plan area necessary to handle this amount of water is

$$\text{Area} = L/(L/A) = 8.775(10^5) \text{ ft}^2$$

The temperature of the water leaving the tower is

$$t_1 = t_2 - \text{Range}$$

$$t_1 = 108.0 - 10.0 = 98.0°F$$

and the approach is

$$\text{Approach} = t_1 - T_{wb} = 98.0 - 68.0 = 30°F$$

Varying the air flow rate to obtain different L/G ratios and evaluating the integrals by an iterative technique will generate the curves wanted.

Letting L/G = 0.1 the air rate becomes

$$G = L/0.1 = 5.121(10^9) \text{ lbm/hr}$$

The enthalpy of the air entering the tower is approximately the enthalpy of saturated air at the wet bulb temperature.

$$h_1 = 32.43 \text{ BTU/lbm}$$

Arbitrarily choosing the tower to have a shape with the height/width ratio to be 3/2, equation 2.33 yields the governing equations

$$NTU_y = \frac{Ka\Delta y}{L_a} = \frac{\Delta t}{h' - h} \tag{2.41}$$

and

$$NTU_x = \frac{Ka\Delta x}{G_a} = \frac{\Delta h}{h' - h} \tag{2.42}$$

Using equation 2.39 together with 2.41, and 2.42 there is enough information available to do the integrations. Choosing the step size to be a fractional part of a transfer unit the procedure is as follows. As explained, the enthalpy difference in equation 2.41 is approximated by the average of the enthalpy differences entering and exiting the volume increment. Since the enthalpies exiting are unknown the average difference is also unknown. Thus the problem lends itself to solution by successive substitution or other iterative methods. A method of linear interpolation is used in the program, and is formulated as follows. The enthalpies h_1' and h_2' are functions of the water temperature, and the atmospheric pressure

$$h_2' = enth(t_2,P) \tag{2.43}$$

$$h_1' = enth(t_1,P)$$

The average enthalpy difference is

$$\Delta ha = [(h_2' - h_2) + (h_1' - h_1)]/2 \tag{2.44}$$

From equation 2.41 the temperature of the water leaving the increment is

$$t_2 = t_1 - \Delta NTU \times \Delta h_a \tag{2.45}$$

From equation 2.39 the enthalpy of the air leaving the increment is

$$h_2 = \frac{L}{G}(t_1 - t^2) + h_1 \tag{2.46}$$

Substituting 2.45, 2.46, and 2.43 into 2.44 yields

$$\Delta h_a = [enth(t_1 - \Delta NTU \times \Delta h_a, P) - (\frac{L}{G} \times \Delta NTU \times \Delta h_a + h_1)]$$

$$+ (enth(t_1, P) - h_1) /2 = F \tag{2.47}$$

Choose a reasonable Δh_a, and solve equation 2.47 for F. Since there will likely be an error, equation 2.48 becomes

$$\Delta h_{a,1} - F_1 = E_1 \tag{2.49}$$

Choosing another reasonable Δh_a yields

$$\Delta h_{a,2} - F_2 = E_2 \tag{2.50}$$

Assuming the functional relation between the error and h_a is linear a better choice for the average difference is

$$\Delta h_a = \Delta h_{a,1} - E_1(\Delta h_{a,1} - \Delta h_{a,2})/(E_1 - E_2) \tag{2.51}$$

By repeating the steps the desired accuracy can be obtained, and the integration proceeds step by step. By using equal numbers of

horizontal and vertical steps the L/G ratio in each increment remains the same as the L/G ratio for the tower. By varying L/G for different ranges the relations shown in Figure 2.11 are found.

The temperature of the water is checked to determine if it has reached the desired tempearture after each successive row and column of increments are added. The temperature is found by averaging the water temperatures leaving the increments at the base of the tower. In this case, for a Range of 10°F this temperature must be 98.0°F. The exiting enthalpy of the air is found similarly. Knowing this enthalpy the water being evaporated can be found as shown in Example 2.1.

Assuming experimental data is available for the packing used, the actual dimensions of the tower are found from equation 2.41 as

$$\text{Height} = \frac{NTU_y \times L_a}{K \times a}$$

where Ka is usually a function of the L/G ratio.

$$\text{Height} = ((0.3)(583.6))/44.0 = 4 \text{ ft}$$

The width is

$$\text{Width} = 2/3 \times \text{Height} = 2.7 \text{ ft}$$

The length of the tower is

$$\text{Length} = \text{Area}/\text{Width}$$

$$\text{Length} = 8.775(10^5) \text{ ft}$$

Obviously this is an impractical tower. It is necessary to examine

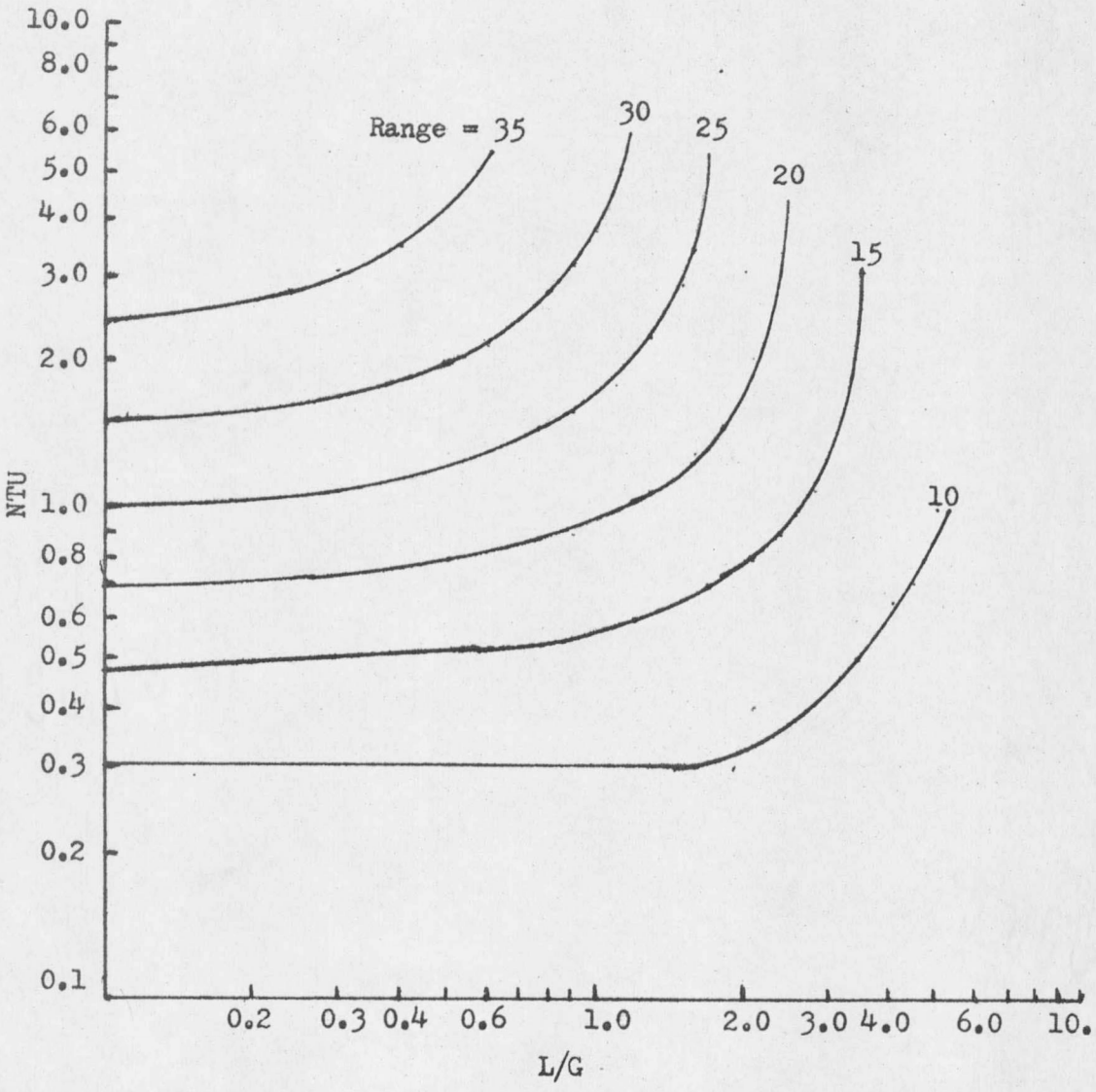


Figure 2.11 Results of integrating the governing equation for the crossflow tower in Example 2.2.

different ranges, shapes, and L/G ratios. Increasing the range or L/G ratio will make the tower taller, wider and shorter in length. An efficient tower has a tendency to be shorter and narrower. As the width decreases the length is increased to handle the water flow rate. This indicates the necessity of trying different height/width ratio designs. The larger ranges and smaller approaches require less water to be cooled. This in turn decreases the plan area necessary to handle the water flow rate. For normal tower configurations the length calculated includes both sides of the tower, so the tower will be half as long and about twice as wide as stated. Such a tower configuration is illustrated in Figure 2.8.

Since the integration is done in integer steps the plot in Figure 2.11 is an approximation to the results. If this plot is compared to the one obtained for counterflow cooling it is seen that the crossflow tower is less efficient for all ranges.

Finding the available coefficient of a particular crossflow tower is done in the same way as for the counterflow towers.

Chapter 3

DRY COOLING TOWERS

The idea of using dry cooling for power plants is relatively new, but because of the present state of heat exchanger theory, the problem is not overly formidable. Kays and London (5) have done extensive analytic and experimental work with heat exchangers, and Johnson and Dickinson (6) and Moore (7) have presented papers on sizing dry cooling towers.

Heat exchangers having a gas for one working fluid and liquid for the other present special problems to a designer. Since the gas has a much lower density than the liquid, larger velocities for the gas are necessary to obtain similar mass flow rates. This combined with the lower thermal conductivities of most gasses results in low heat transfer rates per unit of area of the gas side of the heat exchanger. One of the objects in designing a liquid to gas heat exchanger is to maximize surface area on the gas side of the heat exchanger. Since one of the major costs of operating a dry cooling tower is the cost of operating the fans which must overcome the air flow friction, it is necessary to balance between the power necessary to overcome the flow friction, and the heat transferred per unit area. According to Kays and London, the heat transfer rate per unit area varies roughly with the first power of the velocity, but the friction power varies by as much as the cube of the velocity, and by at least the square of the velocity. Kays and London agree with Johnson and Dickinson that the material, weight,

and thereby the volume of a heat exchanger, for a given load, should be minimized. Their work shows that a thin heat exchanger with a large frontal area will provide a minimum volume.

Moore, Johnson and Dickinson all emphasize the necessity of designing for entrance and exit effects on the gas side of the heat exchanger. Since air entering the heat exchanger is accelerated from nearly zero velocity to that existing within the heat exchanger, the entry region must be designed to minimize the turbulance in this region if the pressure loss is to be kept to a minimum.

There are two methods generally used in analyzing heat exchangers. One method uses the log mean rate equation

$$q = U\ A\ F_G\ \Delta t_m \tag{3.1}$$

where U is the overall conductance for heat transfer in BTU/(hr °F ft^2), A is the surface area on which U is based, Δt_m is the log-mean temperature difference corresponding to counterflow. F_G is a correction factor used for arrangements other than counterflow. If the exchanger is actually a counterflow unit, the nondimensional factor F_G is unity. For all other flow arrangements F_G is less than unity. This factor has been calculated for a variety of flow configurations and was presented in graphical form by Bowman, Mueller, and Nagel (8). Since then Kays and London have done extensive experimental work with most common, heat exchanger geometries (5). Their results are used in the example which follows later, and in the corresponding program in Appendix III.

Kays and London use a method for analyzing heat exchangers that is different than the log-mean temperature difference method. The analysis is formulated in nondimensional variables. The heat transfer rate equation is

$$\frac{dq}{dA} = U\,(t_h - t_c) \tag{3.2}$$

where U is related to the side with area A. U is an overall thermal conductance which is a function of the hot-side film convectance, the wall conductance, the effects of fins or extended surfaces on either side, and fouling factors on both the hot and cold sides. By combining the rate equation with the energy equations, Kays and London develop the parameters discussed below.

The hext exchanger effectiveness compares the heat transferred to the maximum possible heat transfer as limited by thermodynamics.

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_h(t_{h,in} - t_{h,out})}{C_{min}(t_{h,in} - t_{c,\,in})} = \frac{C_c(t_{c,out} - t_{c,in})}{C_{min}(t_{h,in} - t_{c,in})} \tag{3.3}$$

The effectiveness compares the heat transferred by a heat exchanger to the heat transferred by an infinitely long counter flow heat exchanger.

The number of heat transfer units is defined by

$$N_{tu} = \frac{A\,U_{av}}{C_{min}} = \frac{1}{C_{min}} \int_o^A U\,dA \tag{3.4}$$

This nondimensional number is a parameter relating the effective size

of exchangers. For a particular flow arrangement, it is proportional to the surface area on one side of the exchanger, but is a function of other variables as well. The fluids being heated and cooled, material the exchanger is made of, flow rates of the fluids, and geometric design all affect U_{av}.

The last parameter needed in the Kays and London analysis is the capacity-rate ratio

$$\frac{C_{min}}{C_{max}} \tag{3.5}$$

The three parameters defined above contain all the independent variables involved and it is possible to relate them as

$$\varepsilon = F(N_{tu}, \frac{C_{min}}{C_{max}}, \text{flow arrangement}) \tag{3.6}$$

For some flow arrangements there are definite derivable algebraic relations between these variables. For other arrangements there are graphical relations that have been found experimentally. Kays and London present results from experimental work for finding the film coefficients in the exchangers. These are presented in tabular and graphical form as

$$N_{St} N_{Pr}^{2/3} = \Phi(N_{Re}) \tag{3.7}$$

The product of the Stanton number and the two thirds power of the Prandtl number is a function of the Reynolds number.

Friction Effects. As mentioned before the friction effects on the

gas side of an exchanger are important considerations to a designer. Kays and London present the friction factor, f, as a function of heat exchanger core geometry and the Reynolds number. The pressure drop relation is

$$\frac{\Delta P}{P_1} = \frac{G^2}{2g_c} \frac{v_1}{P_1}\left[(K_c + L - \sigma^2) - 2\left(\frac{v_2}{v_1} - 1\right) + f\frac{A}{A_c}\frac{v_m}{v_1} - (1 - \sigma^2 + K_e)\frac{v_2}{v_1}\right] \qquad (3.8)$$

$\frac{\Delta P}{P_1}$ = entrance effect + flow acceleration + core friction + exit effect

K_c and K_e are given for many geometries as a function of Reynolds number. For fow normal to tube banks, entrance and exit loss effects are accounted for in the friction factor and $K_c = K_e = 0$.

Formulation of the Problem

When analyzing heat exchangers, usually the major concern is the amount of heating or cooling they will provide under certain conditions. If enough is known about the exchanger, and the fluids, the performance can be predicted. Assuming the following conditions and properties are known, the method of prediction is outlined by the following example.

Operating conditions:

Air flow rate . . . . . $\dot{m}_a$

Air humidity. . . . . ϕ

Air entering temperature. . . . . T_i

Air entering pressure . . . . . P_1

Water flow rate . . . . . $\dot{m}_w$

Water inlet pressure. . . . . t_i

The overall dimensions of the heat exchanger are known, and the following surface characteristics are known for a given exchanger.

Air side:

Hydraulic radius. . . . . $r_{h,a}$

Total air-side transfer area/total volume. . . α_a

Fin area/total area . . . . . A_f/A

Free-flow area/finned area. . . . . σ_a

Fin thickness . . . . . δ

Fin material conductivity . . . . . k

Fin length (one-half distance between tubes). . . ℓ

Water side:

Free-flow/frontal area. . . . . σ_w

Water side transfer area/total volume . α_w

Water side flow passage hydraulic radius . . . $r_{h,w}$

The following core dimensions are known:

Air-side frontal area . . . . . $A_{fr,a}$

Water-side frontal area . . . . . $A_{fr,w}$

Total heat exchanger volume V

To find the gas viscosity and the specific heat of the gas, a bulk average air temperature must be estimated. This is necessary to find the Stanton, Prandtl, and the Reynolds numbers. To do this one must make a reasonable assumption for the temperatures of the fluids exiting the heat exchanger, t_o and T_o. The average temperatures are calculated using a suitable method. The log-mean method is preferred.

$$t_w = (t_i + t_o)/2 \tag{3.9a}$$

$$T_o = t_w + \frac{(t_i - t_o) - (t_o - T_i)}{\ln \dfrac{t_i - T_o}{t_o - T_i}} \tag{3.9b}$$

The viscosity, Prandtl numbers and specific heats are found in property value tables. The specific heat of the air should be corrected for humidity. In terms of the humidity ratio, the specific heat of moist air is

$$C_p = \frac{1}{1+\omega} C_{pa} + \frac{\omega}{1+\omega} C_{p\mu} \tag{3.10}$$

where

$$\omega = .622 \frac{\phi P_g}{P - \phi P_g} \tag{3.11}$$

The Reynolds numbers are found using the mass flow rate per unit of free flow area.

$$G_a = \frac{\dot{m}_a}{A_{c,a}} = \frac{\dot{m}_a}{A_{fr,a}\sigma_a} \tag{3.12}$$

$$N_{Re,a} = \frac{4r_{h,a}\, G_a}{\mu_a} \tag{3.13}$$

and

$$G_w = \frac{\dot{m}_w}{A_{c,w}} = \frac{\dot{m}_w}{A_{fr,w}\sigma_w} \tag{3.14}$$

$$N_{Re,w} = \frac{4r_{h,w}\, G_w}{\mu_w} \tag{3.15}$$

The film coefficient is found from the Nusselt number, or a similar correlation that is given as a function of the Reynolds number. Kays and London use the correlation shown in equation 3.7. The friction factor for most common geometries is also given as a function of the Reynolds number. The film conductance is given by

$$h = N_{St}\, G\, C_p \tag{3.16}$$

or

$$h = \frac{N_{Nu}\, k}{4\, r_h} \tag{3.17}$$

The overall coefficient of heat transfer based on the air side area, and ignoring the small thermal resistance of the wall is

$$\frac{1}{U_a} = \frac{1}{\eta_o h_a} + \frac{1}{(\alpha_w/\alpha_a)\, h_w} \tag{3.18}$$

Thermal resistance due to fouling has also been ignored. Methods for determining the fin effectiveness, η_o, are given in most convective heat transfer books.

The capacity rates are now found as

$$C_a = \dot{m}_a c_{pa} \tag{3.19a}$$

and

$$C_w = \dot{m}_w c_{pw} \tag{3.19b}$$

The number of exchanger transfer units are found using equation 3.4.

$$N_{tu} = \frac{U_a A_a}{C_{min}}$$

The effectiveness of the heat exchanger is given as a function of the number of transfer units, and is now determined. After simplifying equation 3.3 becomes

$$\varepsilon = \frac{T_{wi} - T_{wo}}{T_{wi} - T_{ao}} \tag{3.20a}$$

if C_{min} is on the water side. If C_{min} is on the air side the equation becomes

$$\varepsilon = \frac{T_{ao} - T_{ai}}{T_{wo} - T_{ai}} \tag{3.20b}$$

Equation 3.20 combined with the energy balance

$$C_a(T_{ao} - T_{ai}) = C_w(T_{wi} - T_{wo}) \tag{3.21}$$

gives two equations and two unknowns, the exiting temperatures, which can then be determined. Initially the temperatures were assumed in order

to determine the fluid properties. If the temperatures found are much different than those assumed the procedure should be repeated using the new fluid properties.

The pressure drop of the air is of major concern in this type of heat exchanger. Equation 3.8 has the pressures involved in both sides of the equation due to the terms involving the specific volumes. It is necessary to iterate to obtain the values of P_2, the exiting pressure. The specific volumes are found from the perfect gas relation

$$v = \frac{RT}{P} \tag{3.22}$$

The amount of water vapor in the air has an effect on the value of R and v. The gas constant in terms of pressures and molecular weights is

$$R = \frac{\bar{R}}{M} = \frac{1545}{\frac{P - \phi P_g}{P} M_a + \frac{\phi P_g}{P} M_v} \tag{3.23}$$

The specific volume after substituting for the gas constant is

$$v = \frac{1545T}{(P - \phi P_g)M_a + \phi P_g M_v} \tag{3.24}$$

All the terms involved in finding the pressure drop have been determined except the relative humidity of the air exiting the exchanger. Since the humidity ratio or moisture content of the air is constant, a relation between the relative humidities can be determined algebraically. The correlation between the relative humidites at any two points in or around the heat exchanger is

$$\phi_2 = \frac{P_2}{P_1} \frac{P_{g1}}{P_{g_2}} \phi_1 \qquad (3.25)$$

Equation 3.8 can now be solved by a suitable iterative technique. The program listed in Appendix III uses a linear interpolation scheme.

In electrical power generating plants, it is important to deliver the water to the condenser at a suitable temperature. Since the temperature of the water leaving the cooling tower is of primary importance, the analysis can be formulated with this temperature specified. If a cooling unit is chosen, and the amount of water that can be cooled by this unit is determined, it is then known how many similar units are necessary to handle the cooling load. The temperature of the water that is delivered is obviously limited by the dry bulb temperature of the air. For a particular unit the geometry is fixed, and the only unknowns are the flow rates, the temperature of the air exiting, and pressures of the fluids exiting. Since the flow rate of the air is an important parameter it can be used as the independent variable. When the Reynolds number of the air is fixed, the amount of water it is possible to cool and the exiting temperature of the air are fixed, although they are still unknown since the energy balance equation must be solved simultaneously with the heat transfer rate equation. The simultaneous solution is obtained by an iterative method described in the following three paragraphs.

The effectiveness of the heat exchanger is found by one of two

methods. One is to use the definition of the effectivness, $\varepsilon=q/q_{max}$. The other is to use the functional relation, $\varepsilon = \Phi(N_{tu})$, for a particular flow configuration. For a heat exchanger operating at steady state these methods must give the same results.

Since providing the air flow rate generates the highest operating cost for a dry cooling tower, the Reynolds number of the air is the most logical choice for the independent parameter. Therefore the object is to determine the performance of the heat exchanger for a fixed air flow rate. If a reasonable assumption is made for the exiting air temperature, an estimate of the Reynolds number of the water, or its mass flow rate, can be determined from an energy balance using equation 3.21. The effectiveness can then be found using $\varepsilon = q/q_{max}$ or for simplicity using equation 3.20. This effectiveness will be denoted as ε_t. Using the assumed temperature, the fluid properties can be found using the suitable average temperatures. The film conductances are then determined using equations 3.16 and 3.17 or other similarly applicable relations found in the literature. Using the film conductances, the overall coefficient of heat transfer is found from equation 3.18. Then using equation 3.4 the number of transfer units is found. From algebraic relations or from graphical relations the effectiveness can then be found from the number of transfer units. This effectiveness will be denoted as ε_g. Call the difference between the two, $\Delta\varepsilon_1$. By assuming a different temperature and repeating the process a second

difference, $\Delta\varepsilon_2$, is determined. From those two differences values, it is possible to use linear interpolation to find a more accurate temperature for the exiting air. That temperature is given by

$$T_o = T_{o,1} + \frac{(T_{o,1} - T_{o,2})\ \Delta\varepsilon_1}{\Delta\varepsilon_2 - \Delta\varepsilon_1} \tag{3.26}$$

Progressively more accurate air exit temperature estimates are obtained by repeating the process. Convergence of this iterative scheme is rapid.

The pressure drop can then be found using equations 3.8, 3.22, 3.23, and 3.24. That pressure differential will determine what fan is necessary to produce the required air flow rate.

Example of dry cooling. The object is to accomodate the cooling load for the same 1000 MW plant as before, but the operating conditions are changed to make the plant more suitable for dry cooling. For simplicity, assume the cooling tower has to expel the same amount of heat as before.

$$q = 5.129(10^9)\ \text{BTU/hr}$$

Let the operating conditions be as follows:

Dry bulb temperature = T_{db} = 70.0°F

Entering water temperature = t_i = 140.0°F

Exiting water temperature = t_o = 110.0°F

Since the water is cooled through a 30°F range the water flow rate to be handled is

$$\dot{m}_w = q/(t_i - t_o) = 1.710(10^8) \text{ lbm/hr}$$

The heat exchanger chosen is a fin tube arrangement with continuous fins. The tubes are spaced 1.50 inches center to center and normal to the flow. They are spaced 1.75 inches from center to center parallel to the flow, and have an outside diameter of 0.676 inches. The whole unit is 10 ft long by 5 ft tall perpendicular to the air flow and is 2 ft thick parallel to the air flow. The remaining significant physical parameters are as follows:

$r_{h,a} = 2.85 \times 10^{-3} \text{ft}$

$\alpha_a = 169.0 \text{ ft}^2/\text{ft}^3$

$\sigma_a = 0.481 \text{ ft}^2/\text{ft}^2$

$\delta = 1.333 \times 10^{-3} \text{ ft}$

$k = 140.0 \text{ BTU/(hr ft °F)}$

$\ell = 6.25 \times 10^{-2} \text{ ft}$

$\sigma_w = 0.5469 \text{ ft}^2/\text{ft}^2$

$\alpha_w = 19.42 \text{ ft}^2/\text{ft}^3$

$r_{h,w} = 1.408 \times 10^{-2} \text{ ft}$

$A_{fr,a} = 50.0 \text{ ft}^2$

$A_{fr,w} = 10.0 \text{ ft}^2$

$\phi = 50\%$

By following the second method outlined in the formulation, the water flow rate, the heat transferred, the exiting air temperature, and the pressure drop are found. The heat exchanger used has a fin arrangement such that there is no pronounced laminar to turbulent transition effects on the air side. As a result the amount of water it is capable of cooling increases with the air Reynolds number, but not as rapidly as it might increase with another type of heat exchanger. This is shown in Figure 3.1. Some heat exchangers are made with dimpled tubes and wavy fins to promote the turbulent effects.

The amount of heat transferred per heat exchanger unit is directly proportional to the amount of water that is cooled through the fixed range as shown in Figure 3.2. Under these conditions with an air Reynolds number of 3000, the heat transferred by each cooling unit is 1.17×10^6 BTU/hr; thus 4400 heat exchangers are required to transfer the 5.129×10^9 BTU/hr to the air.

As the flow rate of the air is increased, the temperature rise of the air will be decreased since it will be in the exchanger for a shorter period of time. Figure 3.3 shows this relation, although due to viscosity variation the air Reynolds number is not directly proportional to the mass flow rate. The shape of the curve in Figure 3.4 suggests that effectiveness follows a linear relation with the exiting air temperature, since the effectiveness curve is shaped similar to the temperature curve in Figure 3.3. Equation 3.20b shows

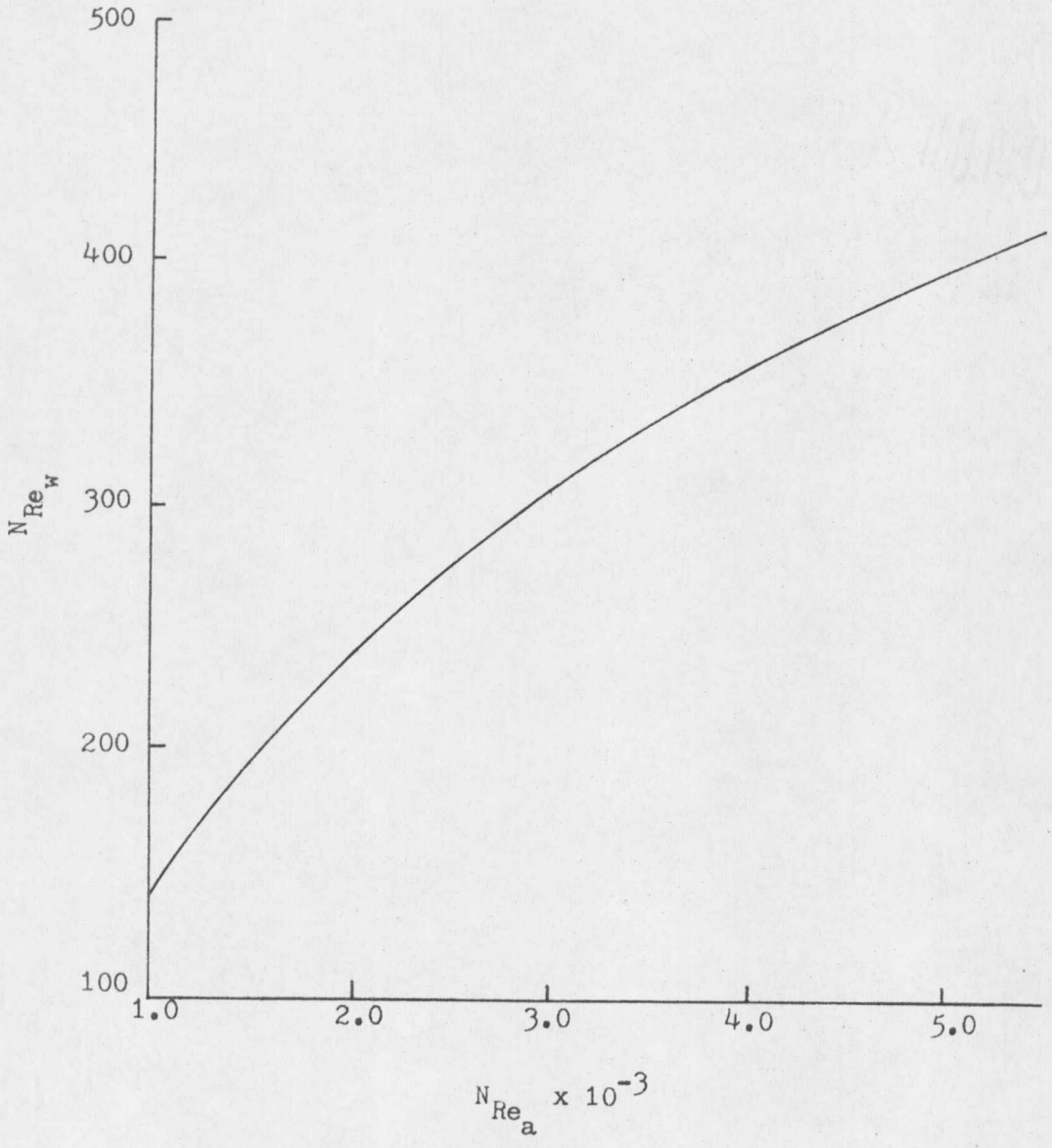


Figure 3.1 Reynolds number of the water as a function of the Reynolds number of the air for fixed water temperatures.

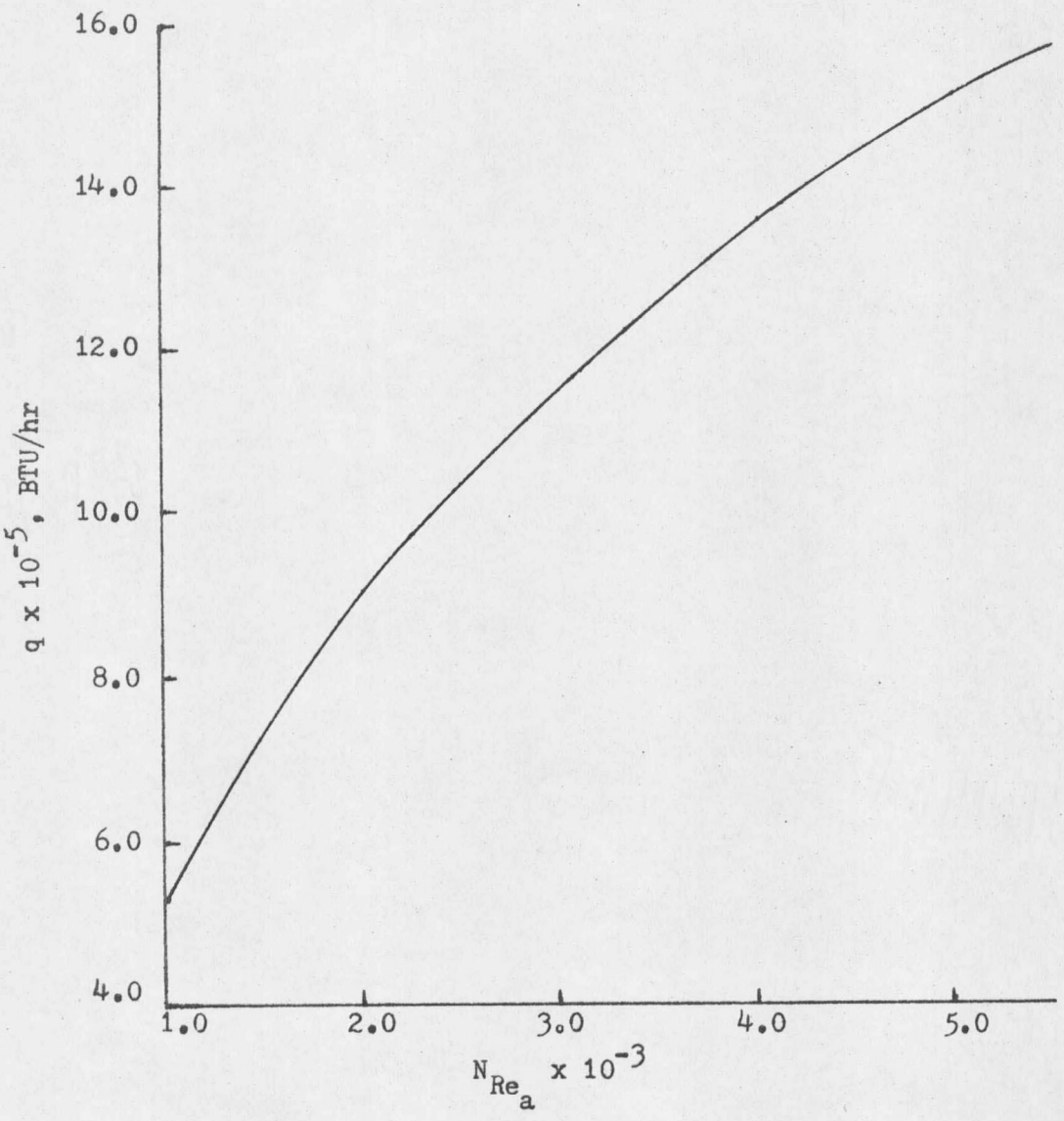


Figure 3.2 Heat transferred from the water to the air in one cooling unit.

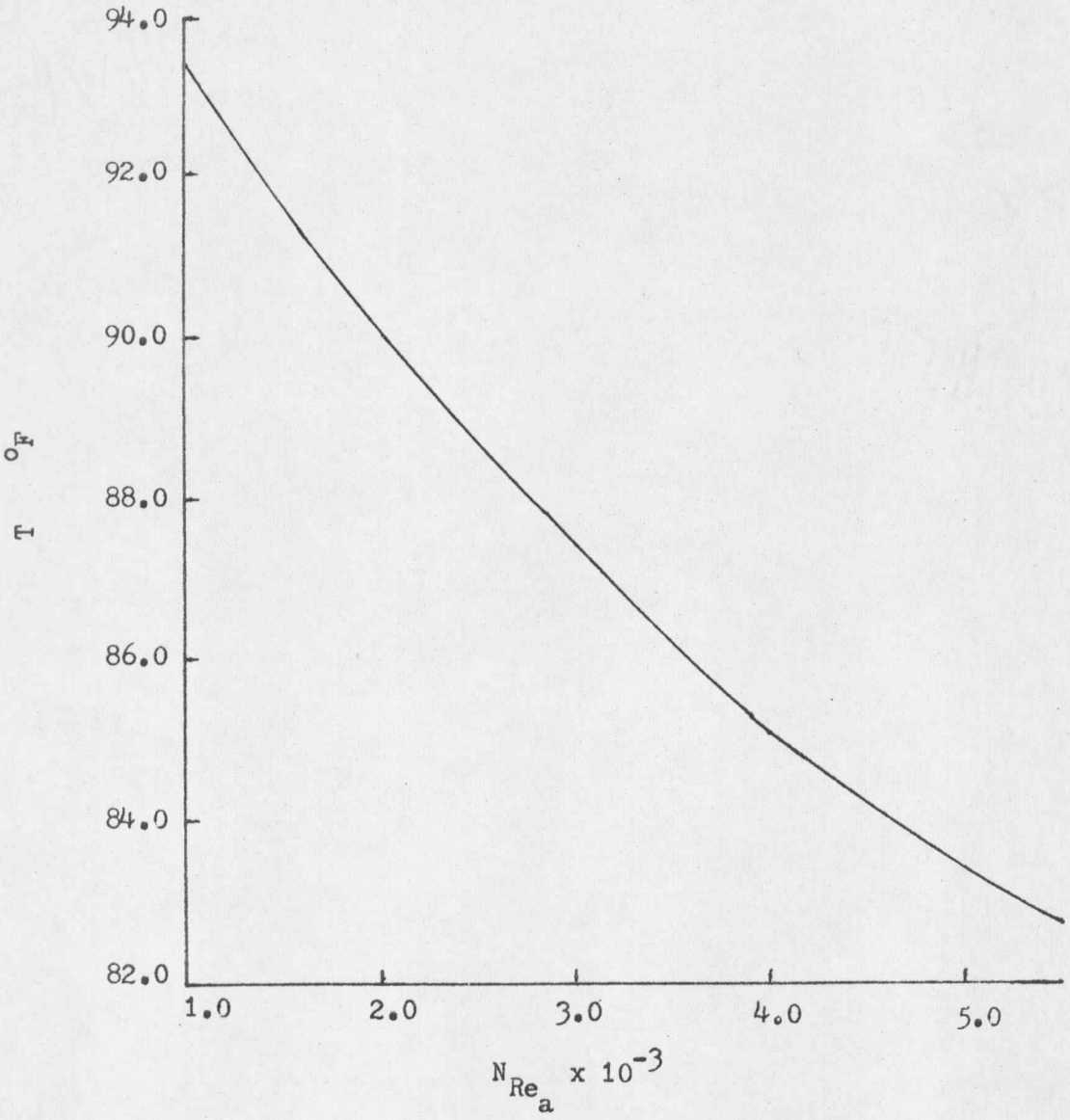


Figure 3.3 Temperature of the air exiting the heat exchanger related to the air Reynolds number.

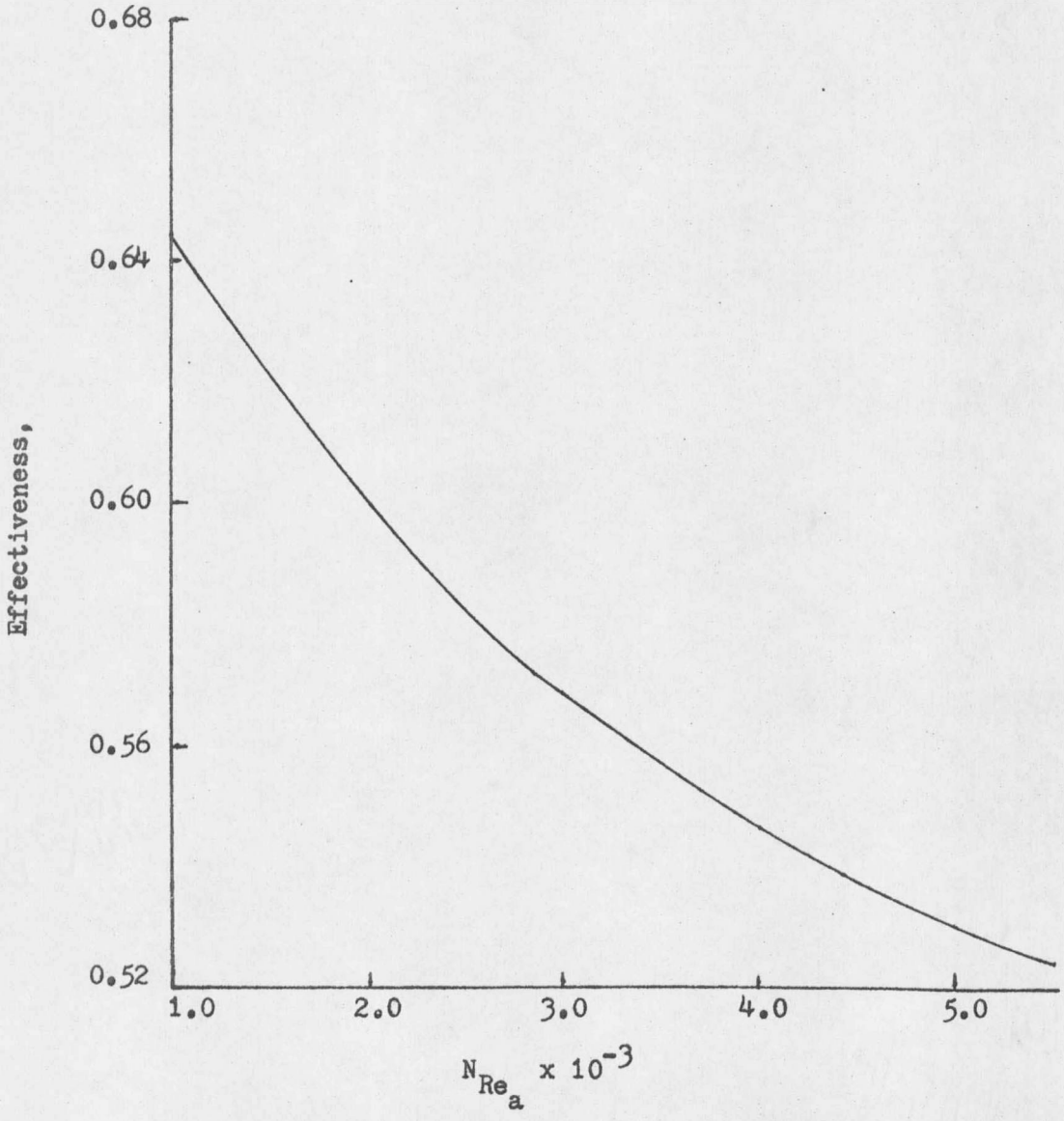


Figure 3.4 Heat exchanger effectiveness as a function of the air Reynolds number.

that for three temperatures fixed, the effectiveness is a linear function of the exiting air temperature.

Kays and London suggest a higher than first order relation between the air velocity, and the friction loss. The results shown in Figure 3.5 agree with this. A heat exchanger designed to promote turbulent flow will have a more pronounced pressure drop than the one studied in this example. Knowing the air pressure drop through the heat exchanger as a function of the Reynolds number allows one to determine the fan size necessary to deliver the desired mass flow rate.

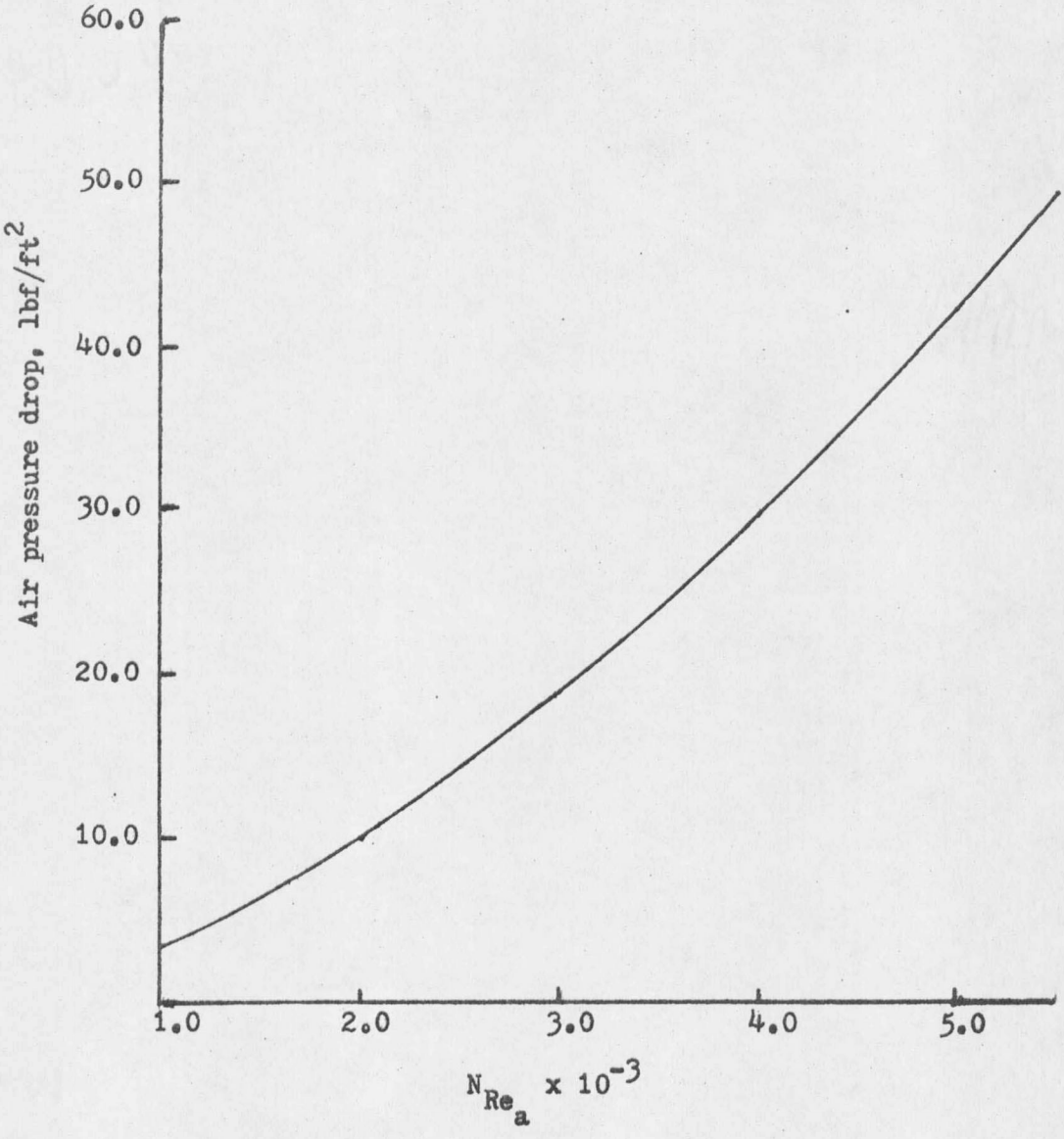


Figure 3.5 Air pressure drop through a heat exchanger as a function of the air Reynolds number.

Chapter 4

WET/DRY COOLING TOWERS

The wet/dry, or hybrid, cooling towers are a step towards solving many cooling tower problems. The wet/dry cooling tower operates by cooling the water delivered to it in two stages. The water first enters a series of heat exchangers, the dry section of the cooling tower. The water is cooled through part of the range in this section. Water exit temperature from this section is limited by the ambient dry bulb temperature. Then the water is cooled using the wet section of the system where water exit temperature is limited by the ambient wet bulb temperature.

The wet and the dry cooling towers have a number of drawbacks that may be overcome by the hybride cooling tower. A wet cooling tower cooling a 1000 MW power plant will evaporate about 5.4×10^5 gal/hr. Along with this loss a part of the water in the system must be replaced so that the mineral and particulate concentrations do not reach a level that will cause excessive corrosion in the system. Replacing this water is called blow down. Using a hybrid system to cool the same power plant allows the wet part of the system to be much smaller. This results in less evaporation and less water lost to blow down.

When cooling towers are located close to residential areas or highways in cold climates there is danger of icing. When the air enters the tower at a temperature below freezing, it is heated, and a certain amount of water is evaporated. As it exits from the tower it mixes with

outside air, and is cooled to the ambient temperature. By examining the saturation vapor pressure as it approaches these lower temperatures, it is found that virtually no water vapor can remain in the air. As this water vapor condenses from the air it can form ice on roads and streets. Using a hybrid system to reduce the evaporation will reduce the icing.

The dry cooling tower alone does not have to contend with the problems mentioned. However there are two drawbacks to the use of dry cooling towers that seriously limit their use. One is the immense size of the tower, and number of heat exchangers required to handle the water. In the previous example, where the operating conditions were favorable, it was necessary to use 4398 heat exchanger elements to cool the water with an air Reynolds number of 3000. This provides a total frontal area of 2.20×10^5 ft^2. For dry cooling of a 1000 MW plant this is not overly large, but it will not provide adequate cooling above 70°F, and the turbines must operate against an absolute back pressure of about three inches of mercury. The second major drawback is that the exiting water temperature is limited by the air dry bulb temperature. If the temperature was 90°F it would take 7406 of the heat exchanger units to handle the cooling load. This structure would be roughly 400 x 925 square feet, which is a total frontal area of 8.5 acres.

For locations with large changes in temperatures, such as Montana,

the dry cooling is not as yet practical. The hybrid cooling system offers a solution to many of the problems relating to the availability of water, and large temperature fluctuations. Laszlo Heller published results of wet/dry cooling towers he designed for the Soviet Union (10). The design work was done for climates with cold winters and hot summers. In this climate the water run off follows the seasons. The water discharge is extremely high at the commencement of the thawing period, and decreases in stages as winter sets in. It reaches a minimum during the period before thawing. The Soviet Union has found that the maintenance of wet towers in temperatures below -5°F presents enough problems to prohibit their use during this period. The wet/dry system was designed to alleviate these problems. During the periods when cooling water is not available the air temperature is extremely low, and a comparatively smaller sized, and therefore less costly air cooling system can be employed. During the winter season the extremely cold atmophere allows the dry heat exchanger to provide the vacuum required in the condenser, even with a heat exchange surface of about one half of the normal surface area. In the summer when ambient temperatures reach 85 to 95°F, abundant water is available, and the cooling water passed through the heat exchange surface of the dry section of the cooling tower is further cooled in the wet section of the cooling tower.

Heller designed the system shown in Figure 4.1. The figure shows that the wet and dry sections of the system are connected in parallel.

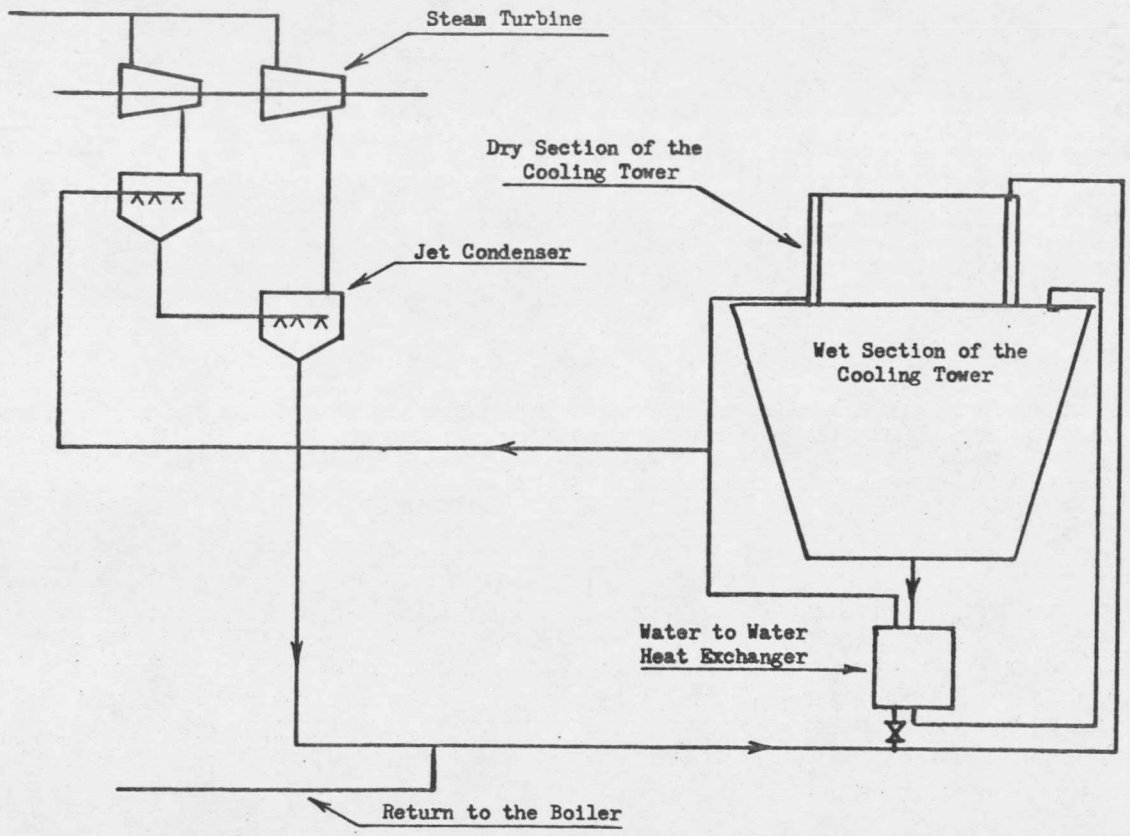


Figure 4.1 The Wet/Dry Cooling Tower utilizing the parallel flow arrangement.

The water drawn from the jet condensers is delivered both to a water-to-water heat exchanger and a water-to-air cooling tower section. In times of cold weather the water may be cycled completely through the water-to-air section. The water in the wet section of the cooling tower is in a cycle by itself, and it collects heat in the water-to-water heat exchanger. By keeping the water in the wet part of the tower from mixing with the rest, any concentrates accumulated in that section will not affect the rest of the system. Thus, the remaining water is maintained at boiler quality and can be used in jet condensers.

A surface condenser will condense the steam at a temperature of about 5 to 8°F greater than the temperature of the water delivered from the cooling tower. Alternately a jet condenser will heat the water from the cooling tower to within 1 to 2°F of the condensing steam temperature. This allows for the water entering and exiting the tower to be about 5°F greater than the same water temperatures associated with a surface condenser for the same turbine back pressure. Normally increasing these temperatures would decrease the size of the cooling tower necessary to handle the heat load, but the water-to-water heat exchanger linking the wet cooling section to the system is not 100% effective, and this forces the cooling tower to cool the water through a slightly larger range in the wet section. Therefore what is gained by increasing the operating temperatures through use of jet condensers may be lost by the use of one more heat exchanger.

The wet and dry sections may also be connected in series as shown in Figure 4.2. For this arrangement all of the condenser water is cooled in the dry section of the tower to a temperature limited by the air dry bulb temperature. It then passes through the water-to-water heat exchanger, and is cooled to a temperature limited by the air wet bulb temperature and the effectiveness of the heat exchanger. During cold periods the wet section may be shut down, and the water-to-water heat exchanger will transfer essentially no heat.

Sizing the Hybrid Tower

Adequate theory has been developed in the previous chapters to size a hybrid tower. Consider the parallel system shown in Figure 4.1. For a 1000 MW plant, the cooling system must dispose of about 5.13×10^9 BTU/hr of heat. If the water enters the condenser at 110°F and exits at 140°F the turbines must operate at a back pressure of 2.89 lbf/in^2 or 5.89 inches Hg.

These conditions, though being far from optimum, can be used to compared the different cooling towers. There are an infinite number of combinations of wet and dry sections that will provide the necessary coooling, and only a few will be examined. For a total optimization it is necessary to consider the climate throughout the year, the operating conditions, water availability, and construction, material, and operating costs.

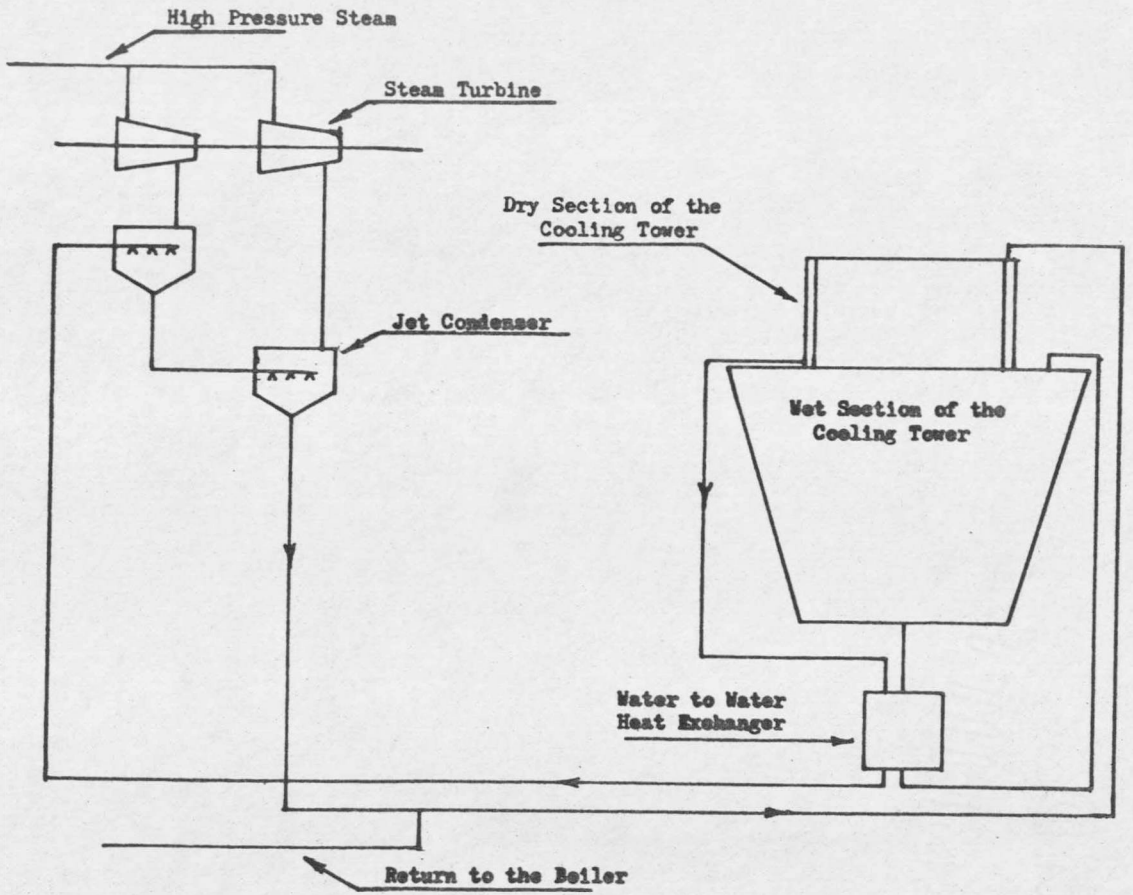


Figure 4.2 The Wet/Dry Cooling Tower utilizing a series flow arrangement.

Consider the plant described on a day when the temperature is 90°F, and the wet bulb temperature is 66.5°F at standard pressure. This corresponds to a relative humidity of about 28%. The flow rate of water to be cooled is 1.71×10^8 lbm/hr. To cool this water in a dry cooling tower with an air Reynolds number of 3000 requires 7535 of the 10 x 5 x 2 ft sections described in the previous chapter. This is a frontal area of 3.77×10^5 ft^2. Cooling this amount of water in a wet cooling tower evaporates 4.76×10^6 lbm water/hr. Three cases of hybrid arrangements will be compared for size and the amount of water evaporated. In two of these cases the wet and dry sections will be parallel. A series arrangement will be used in the third case.

Example 4.1. Rferring to Figure 4.3 consider the case where half the water to be cooled is delivered to the dry section, and half to the water-to-water heat exchanger. The water in the dry section is cooled to t_1 = 120.0°F. Since the wet bulb temperature is 66.5°F, which is the temperature shown as t_3, and is about the lowest temperature that it is practical to deliver at this point. The water-to-water heat exchanger will deliver water to the condenser at a temperature t_2, which is 5 to 10°F greater than the temperature t_3. This limits the minimum temperature of the water the wet section of the cooling tower can actually deliver to about 80°F. This is below the dry bulb temperature. If the humidity is such that the wet bulb temperature is within 10°F of the dry bulb temperature it will be difficult to cool the condensing

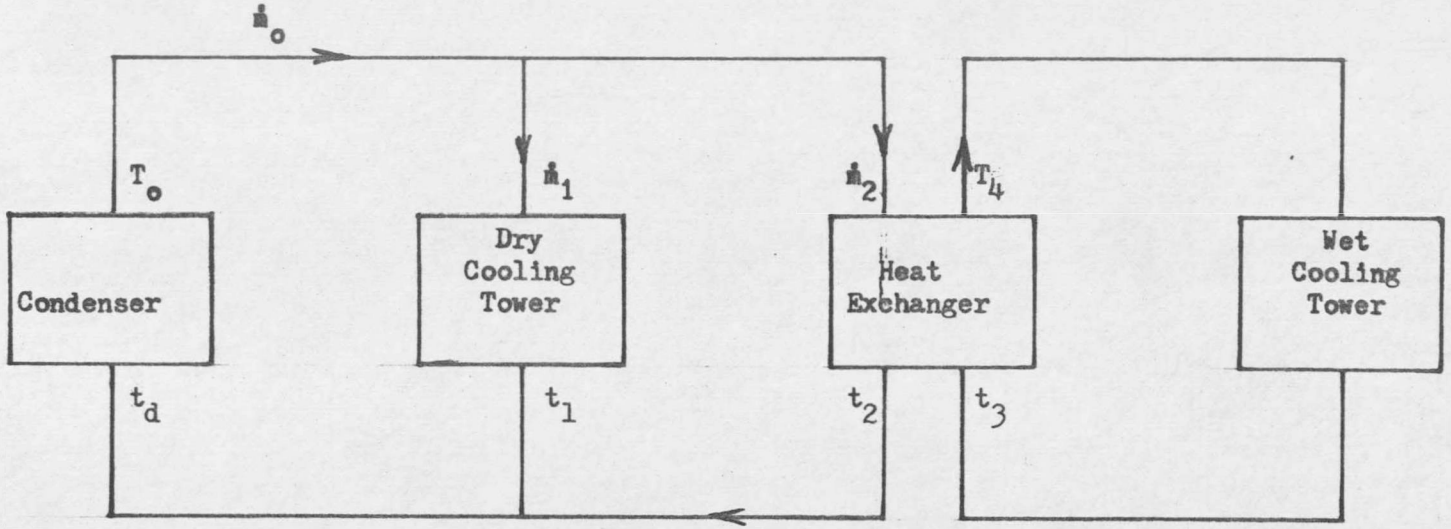


Figure 4.3 Diagram showing parallel flow arrangement.

water below the dry bulb temperature.

From the energy balance

$$\dot{m}_a t_d = \dot{m}_1 t_1 + \dot{m}_2 t_2 \qquad (4.1)$$

t_2 can be obtained, and for this example t_2 = 100.0°F. As the amount of water cooled in the dry section is increased, the temperature of the water leaving the water-to-water heat exchanger, t_2, must be decreased to obtain the desired temperature, t_d, of the water delivered to the condenser. Since t_2 has a minimum of 10 to 20°F greater than the wet bulb temperature, the amount of water that can be cooled in the dry section has a maximum if it is not cooled through the complete range.

Knowing the heat to be transferred in each section and somewhat arbitrarily letting the wet tower cool from 130°F to 90°F, the same programs used in the previous chapters are sufficient to do the heat transfer and sizing analysis. It takes 1996 of the dry heat exchanger sections to cool half the water through 2/3 of the range as compared to 7535 to handle the full load. The water is evaporated in the wet section at the rate of 2.87×10^6 lbm/hr as compared to 4.76×10^6 lbm/hr if all the heat is expelled in the wet tower. The wet/dry tower in this example would reduce the amount of makeup water needed by about 40% during the hot weather periods and by more during the cooler periods.

In locations in Montana where much of the year has temperatures at

about 10°F it would be desirable to have a tower that would operate on the dry part alone during the winter months. To obtain the same cooling at 10°F requires 1945 of the dry heat exchanger elements. Therefore the combination of wet and dry cooling just examined could cool completely with dry cooling through much of the winter in many locations.

Example 4.2. When construction costs are of major importance in plant design the tower can be designed to use some water all year around. Letting the dry section cool a fourth of the water, 4.275×10^7 lbm/hr, to 120°F and using equation 4.1 the temperature of the water delivered from the water-to-water heat exchanger is found to be 107°F. The dry cooling section for this example requires 998 of the hext exchanger elements to handle the load with an air Reynolds number of 3000. With the wet tower handling the majority of the load, and cooling from 130 to 90°F it will evaporate about 3.6×10^6 lbm/hr under the conditions of 90°F and a relative humidity of 28%. This will result in a savings in makeup water of 24% compared to a wet tower. When the temperature is 10°F the dry section will handle 51% of the load, and the amount of water evaporated is greatly decreased due to the smaller load to the wet section, and the colder air which can accomodate much less water vapor.

Example 4.3. There is but one sensible way to make the series

connection, and that is to cool in the dry section first and then the wet section. Taking the same load and cooling to 125°F in the dry section and cooling through the remainder of the range to 110°F in the water-to-water heat exchanger requires 2757 of the heat exchanger elements and evaporates 2.2×10^6 lbm/hr when the wet tower cools from 115 to 90°F. Since this results in many more elements than is needed to dry cool the full load in the winter, the dry section of the tower may be oversized depending upon the design criteria. Although half the heat is handled in the dry section, it requires only 37% as many elements as is required for dry cooling the full load.

The series arrangement has the advantage over parallel arrangements that it can cool further below the dry bulb temperature. The parallel system must cool to lower temperatures in the wet section than the series system must as only part of the water is cooled, and mixed with the water that was cooled in the dry section.

There are many plant sites available in the northern states and Canada that have some water available for cooling, but not sufficient for a totally wet cooling system. For these sites a wet/dry cooling tower should be considered. In locations where there is little or no runoff during the colder periods, wells could be used in combination with the dry section of a wet/dry tower to cool the condensers, and both sections used in conjunction during the warmer seasons when runoff is available. If the dry section is sized accordingly the only water

needed is what is used to replace leakage, and maintain the boiler feed water within the system. A tower sized to operate on the dry section alone during the winter may require as much as 40 to 50% less water during the summer months than a wet tower, and tributaries leading to major rivers may provide sufficient water for siting.

Chapter 5

ANALYZING COSTS

Due to the expenditure involved extensive consideration must be given to all the variables affecting the initial and operating costs of a power plant and cooling tower. The power plant designer must consider the conditions that exist in the water systems during all operating modes. He must consider variation in heat loads and power demands and their relation to expected weather conditions. Also important are the performance variables of the condenser and cooling towers, and the changes in efficiency of the steam turbine as various demand loads occur at different back pressures.

The ultimate procedure for selection of the cooling tower would involve examining all possible variations of water system equipment, and evaluating operating and first-cost economics for all combinations. Operating cost calculations can be made for short time increments. The initial expenditure can be put on a uniform annual cost basis and added to annual operating cost to obtain a total annual cost. Total annual cost for all the variations can then be compared to find the least expensive option.

Once the combination of components is chosen the designer can specify the condenser and cooling tower equipment that is necessary to meet his design criteria. These design conditions are generally established so that they represent a condition of operation that is expected during a reasonably large number of hours. The cooling system

equipment must be optimized with each combination of components being studied. Optimization of plant cooling water system equipment involves the selection of the pumps, piping, tower and condenser for the most favorable year-round plant operation. In each case the design criteria must be established from the plant components chosen before the cooling system can be optimized. The methods presented in the previous chapter for physically sizing the cooling towers become essential to the overall optimization.

The turbine can be operated most efficiently against a back pressure corresponding with a saturation temperature of about 85 to 100°F for full load conditions. Standard wet cooling towers have traditionally been designed to maintain condenser temperature in that range for about half the operating year (9). Reasonable efficiency and reliability is assured at back pressures ranging up to 4-1/2 inches of mercury pressure. Most turbine equipment is guaranteed for operation up to about 5 inches of mercury, or 134°F saturation temperature.

Surface condensers are selected by means of establishing an overall heat transfer coefficient with foul factor, and are sized to produce a given mean temperature difference between the saturation steam temperature and the hot and cold water circulating within the tubes of the condenser. The terminal temperature difference, that is the difference between temperature of the cooling water exiting the condenser and the saturated steam temperature, must exceed 5°F at

design conditions. This is a minimum level established by the Heat Exchange Institute.

When plant designs are evaluated to determine relative costs, studies are made to determine year-round operating economics by examining different combinations of water system components. The pumps, piping system, condensers and cooling towers are considered in these evaluations. In addition the changes in KW output capability and changes in the fuel costs must be considered. Uncertainties in cost estimates occur due to changing costs of materials used to make various components. The influence of complexity of the design on labor cost have a large bearing on the initial cost of a plant. When the turbines are operated at a higher back pressure, the fuel consumption must increase to maintain the same electrical power output. More heat is expelled in a cooling tower when it is operated at higher temperatures, therefore as the turbine back pressure is increased the size of the cooling tower may be decreased. If equipment and tower costs have more effect than the fuel costs on total expenses it is more economical to operate at these higher back pressures.

It is important to define the type of operation that is to be expected at a given plant. For example, if a plant is to be operated under conditions which cause high tower cost such as operating full load during the summer when wet bulb temperatures are highest, it is possible that optimum economic conditions will indicate using a

turbine back pressure beyond the turbine manufacturers' point of guarantee. Since this cannot be done the choice must be made, in order to evaluate the costs, as to whether the plant can be operated with some degree of unloading, relying on peaking plants or other systems to pick up the difference in loads. This would not be acceptable in locations that have summertime peaks throughout the whole area, as all the plants within the area will be dealing with the peak loads simultaneously. When dry cooling is considered it may economically call for trubine back pressures far above the limits of what is available today.

When economically optimizing the range and approach for operating wet cooling towers, it is necessary to consider the expected variation of load and simultaneous weather condition variations at the location being studied. The most accurate study is one which uses the smallest time increment possible. However, hourly studies on a year-round basis can be too time consuming and expensive even for todays computers, because there are a wide variety of condensers, pumps,and cooling towers than can be included in such a study. Such studies are usually done after turbine specifications have been fixed. Each combination of tower, pump, and condenser can be evaluated separately in conjunction with probably changes in fuel costs to indicate relative economic advantages of one system versus the other.

Estimating the expected weather conditions and load is difficult

for a proposed site. Past weather conditions for a proposed site are not usually completely recorded, and load demand will be influenced by cost, future population levels, and other factors which are as difficult to forecast as the weather. The average temperature of a monthly time period is usually far below the maximum temperature, making it possible to misinterpret the maximum back pressure that might be encountered during the hottest periods of the summer. In actual operation, as the high wet bulb conditions occur within a monthly period, plant fuel consumption rates rise due to turbine inefficiencies at high back pressure. The result is that the fuel cost component of the evaluation will be somewhat low. These are factors which should not be overlooked, if full load conditions occur at the same time as extreme high wet bulb temperatures.

Choosing components for dry cooling involves many of the same considerations as for evaporative cooling, but due to the increased costs of equipment, the economics will optimize at higher back pressures. If fuel costs increase radically during the life of the plant, the optimum back pressure will decrease, and this possibility should be considered in any analysis done. In Chapter 3 examples were given where the amount of cooling, number of units necessary, and pressure drops were given as a function of the air Reynolds number. Given typical fan performance curves and costs, the optimum tower can be designed for particular conditions. Since these conditions are not constant

the overall system and meteorological conditions must be considered simultaneously.

As with evaporative systems, the use of peaking plants or other methods of handling loads during high dry bulb temperatures must be considered. Trade-offs involve an increased initial investment in equipment in return for future savings in fuel because of improved efficiency or lower operating and maintenance costs. Other means of maintaining generating capacity at high ambient temperatures include augmenting heat transfer with a fine spray of water on the exchanger fins and tubes and precooling the air by evaporative cooling (10). In mechanical draft installations, the selection of multispeed fans offers the option of operating the fans at the higher speed to improve heat rejection at high ambient temperatures.

In areas where water is available the problem of handling peak loads during the summer may be solved by the use of hybrid towers. The dry cooling is economical in cold weather, and the evaporative cooling assistance is the most economical during the hot periods. The use of hybrid towers further compounds the difficulty of optimization due to the number of combinations available. These additional combinations must also be analyzed for operation throughout the year using the smallest time increments possible.

Chapter VI

CONCLUSIONS

Methods previously developed for sizing evaporative cooling towers were studied, and when appropriate used in developing programs for sizing cooling towers. Merkel's method of obtaining the equation governing the size of evaporative cooling towers was presented along with techniques for integrating this equation. These techniques of integration were presented for both counterflow and crossflow towers. The programs developed were designed to use a minimum of input, with the thermodynamics and fluid properties contained within the programs.

When developing a program to size dry cooling towers, standard heat exchanger theory was used to do the heat transfer analysis. The program that was developed to size dry cooling requires the most input due to the number of physical heat exchanger parameters which must be predetermined. Parameters to be aware of when minimizing the size of water-to-air heat exchangers were brought forward.

The methods for sizing evaporative and dry cooling towers were used in combination to size the hybrid towers. The adaptability of the hybrid tower to hostile environments was stressed. It's ability to overcome the problems of water shortage, and plume abatement were shown. The use of dry or wet/dry cooling expands the number of plant sites available, and allows plants to be located in water-short areas which are often less populated than areas with plentiful water supplies.

The relations between the cooling tower and cooling system and the

economics of the plant as a whole were presented. It was shown that for high labor cost and low fuel cost it is economical to operate at higher back pressures than are currently used. The relative magnitude of fuel, labor, and material costs were shown to control the most economical turbine back pressure and cooling tower size.

APPENDICES

APPENDIX I

PROGRAM FOR SIZING COUNTERFLOW COOLING TOWERS

This program will generate curves relating the number of transfer units to the ratio of the flow rates for a counterflow tower (equation (2.23)). The program was designed to use a minimum of input and contains within it all the thermodynamics necessary to size a counterflow cooling tower. The program was written to handle a specific set of meteralogical conditions, and arranged in a manner that makes the inclusion of year round conditions a minor adjustment.

```
      REAL KA,NTUST,L,LPERA
C**** THIS PROGRAM SIZES COUNTERFLOW COOLING TOWERS FOR
C**** A GIVEN HEAT LOAD WHEN GIVEN; THE DRY AND WET BULB
C**** TEMPERATURES IN FARENHEIT, THE RELATIVE HUMIDITY AS
C**** A DECIMAL, ATMOSPHERIC PRESSURE IN INCHES OF MERCURY,
C**** THE WATER FLOW RATE PER UNIT OF PLAN AREA, AND THE
C**** TOWER PACKING COEFFICIENT, KA, IN LBM PER HR PER
C**** SQUARE FOOT.
C
C**** THE VARIABLES ARE DEFINED AS FOLLOWS.
C
C**** HEAT IS THE TOTAL HEAT TO BE DISSIPATED IN THE TOWER
C**** P IS THE ATMOSPHERIC PRESSURE.
C**** TWB AND TDB ARE THE WET AND DRY BULB TEMPERATURES
C**** RESPECTIVELY.
C**** PHI IS THE RELATIVE HUMIDITY.
C**** T2 IS THE TEMPERATURE OF THE WATER ENTERING THE
C**** TOWER.
C**** LPERA IS THE WATER FLOW RATE PER UNIT AREA.
C
      INPUT HEAT,P,TWB,TDB,T2,PHI,LPERA
      PSAT(TDB)=EXP(-11.6638+45.3179/TDB+2.1423*ALOG(TDB)+.01294337*
     $TDB)
      OUTPUT HEAT
      OMEGA1=.6221*PHI*PSAT(TDB)/(.491*P-PHI*PSAT(TDB))
      RHO=(P*70.53)/(53.35*(TDB+460.0))
      OUTPUT RHO
      LPERA=70.0*8.337
      LPERA=LPERA*1.5
      DO 10 I=10,35,5
C       WILL GENERATE LINES FOR FIXED RANGES, AND FIXED APPROACHES
      RANGE=I*1.0
      L=HEAT/RANGE
      AREA =L/LPERA
C***** T1 IS THE TEMPERATURE OF THE WATER LEAVING THE TOWER
      T1=T2-RANGE
      APRCH=T1-TWB
      WRITE(108,22)
   22 FORMAT('1','RANGE',2X,'APPROACH',2X,'PLAN AREA',7X,'WATER RATE')
      WRITE(108,21) RANGE,APRCH,AREA,L
   21 FORMAT(F4.0,5X,F4.0,4X,1PE11.4,5X,1PE11.4)
      WRITE(108,23)
   23 FORMAT(///,7X,'L/G',13X,'NTU',10X,'AIR RATE',7X,'AIR SPEED',7X,
     $'HEIGHT', 9X,'AIR',12X,'EVAPORATION')
      WRITE(108,43)
   43 FORMAT(82X,'EXIT',11X,'RATE')
      WRITE(108,44)
   44 FORMAT(79X,'TEMPERATURE')
      WRITE(108,45)
   45 FORMAT(35X,'(LBM/HR)',9X,'(FT/MIN)',8X,'(FEET)',5X,'(FARENHEIT)',
     $ 5X,'(LBM/HR)')
C***** L/G=C1
      C1=.1
   40 CONTINUE
      G=L/C1
```

```
      S=G/RHO/AREA
C***** THE FUNCTION ENTH HAS THE ARGUMENTS OF WET BULB TEMPERATURE,
C**** RELATIVE HUMIDITY, AND PRESSURE IN INCHES OF MERCURY
C**** IN THAT ORDER.
      EN1=ENTH(TWB,1.0,P)
C***** EN2 IS THE ENTHALPY OF THE AIR LEAVING THE TOWER
      EN2=EN1+C1*(T2-T1)
      C2=ENTH(T2,1.0,P)
      IF(EN2.GE.C2) GO TO 50
      F0=1.0/(ENTH(T1,1.0,P)-(EN1))
      F1=1.0/(ENTH(T1+.1667*(T2-T1),1.0,P)-(EN1+.1667*(EN2-EN1)))
      F2=1.0/(ENTH(T1+.3333*(T2-T1),1.0,P)-(EN1+.3333*(EN2-EN1)))
      F3=1.0/(ENTH(T1+.5*(T2-T1),1.0,P)-(EN1+.5*(EN2-EN1)))
      F4=1.0/(ENTH(T1+.6667*(T2-T1),1.0,P)-(EN1+.6667*(EN2-EN1)))
      F5=1.0/(ENTH(T1+.8333*(T2-T1),1.0,P)-(EN1+.8333*(EN2-EN1)))
      F6=1.0/(ENTH(T2,1.0,P)-EN2)
      NTUST=(T2-T1)/18.0*(F0+4.0*F1+2.0*F2+4.0*F3+2.0*F4+4.0*
     >F5+F6)
C**** THE ENTHALPY OF THE AIR LEAVING THE TOWER IS KNOWN,
C**** BUT THE TEMPERATURE AND THE HUMIDITY RATIO OF THE AIR
C**** EXITING IS UNKNOWN SO ITERATE TO FIND THESE.
      P2=.491*P
      PHI2=.97
      GT=TWB+10.0
      G2T=TWB+20.0
   42 GOMEG=.6221*PHI2*PSAT(GT)/(P2-PHI2*PSAT(GT))
      GH=.2408*GT+GOMEG*(.43*GT+1061.8)
      G2OMEG=.6221*PHI2*PSAT(G2T)/(P2-PHI2*PSAT(G2T))
      G2H=.2408*G2T+G2OMEG*(.43*G2T+1061.8)
      TAO=G2T+(GT-G2T)*(EN2-G2H)/(GH-G2H)
      OMEGA2=.6221*PHI2*PSAT(TAO)/(P2-PHI2*PSAT(TAO))
      HOUT=.2408*TAO+OMEGA2*(.43*TAO+1061.8)
      ERROR=ABS((EN2-HOUT)/EN2*100.0)
      IF(ERROR.LT.0.05)GO TO 41
      GT=G2T
      G2T=TAO
      GO TO 42
   41 EVAP=G*(OMEGA2-OMEGA1)
C**** KA IS THE TOWER PACKING COEFFICIENT AND MAY BE A
C**** FUNCTION OF THE WATER FLOW RATE.
      KA=44.0
      HEIGHT=NTUST*L/(KA*AREA)
      S=S/60.0
      WRITE (108,20) C1,NTUST,G,S,HEIGHT,TAO,EVAP
   20 FORMAT(7(5X,1PE10.4))
      IF(C1.GE.5.0) GO TO 10
      C1=C1+.1
      GO TO 40
   50 OUTPUT 'NO DRIVING FORCE ANYMORE'
   10 CONTINUE
      END
      FUNCTION ENTH(TDB,PHI,PIN)
      PATM=.491*PIN
      PSAT=EXP(-3.98136+5.2778E-2*TDB-1.4644E-4*TDB**2-4.7138E-7*TDB**3
     >+1.0307E-8*TDB**4-5.0898E-11*TDB**5+8.3956E-14*TDB**6-7.6653E-21
```

```
     *TDB**/-3.5591E-23*TDB**8)
      PVAP=PHI*PSAT
      OMEGA=.6221*PVAP/(PATM-PHI*PSAT)
      ENTH=.2408*TDB+OMEGA*(.43*TDB+1061.8)
      RETURN
      END
```

APPENDIX II

PROGRAM FOR SIZING CROSSFLOW COOLING TOWERS

This program will generate curves relating the number of transfer units to the ratio of the flow rates for a crossflow cooling tower. It performs the same integrations as the previous program except in two dimensions. It includes all the thermodynamics necessary to size the crossflow cooling tower with a minimum of inputs.

```
      DIMENSION TIN(51,51),HIN(51,51)
      REAL NTU1,KA,LPERA,L,LENGTH
      PSAT(TDB)=EXP(-3.98136+5.2778E-2*TDB-1.4844E-4*TDB**2-4.7138E-7*
     $TDB**3+1.0307E-8*TDB**4-5.0898E-11*TDB**5+8.3956E-14*TDB**6-
     $7.6653E-21*TDB**7-3.5591E-23*TDB**8)
C
C**** ALL UNITS ARE FEET, HOURS, FAHRENHEIT,LBF, AND BTU
C**** UNLESS OTHERWISE SPECIFIED.
C
C**** RATIO IS THE RATIO OF THE WIDTH OF THE TOWER TO THE
C**** HEIGHT OF THE TOWER.
C**** TDB AND TWB ARE THE DRY AND WET BULB TEMPERATURES
C**** IN FAHRENHEIT.
C**** PHI IS THE RELATIVE HUMIDITY AS A DECIMAL.
C**** T2 IS THE TEMPERATURE OF THE WATER ENTERING THE TOWER.
C**** PIN IS THE ATMOSPHERIC PRESSURE IN INCHES OF MERCURY.
C**** HEAT IS THE TOTAL HEAT TO BE DISSIPATED BY THE TOWER.
C**** LPERA IS THE WATER FLOW RATE PER UNIT OF PLAN AREA.
C**** KA IS THE TOWER PACKING COEFFICIENT IN LBM PER HOUR
C**** PER SQUARE FOOT.
C
      INPUT RATIO,TDB,TWB,PHI,T2,PIN,HEAT,LPERA
C
C**** FINDING THE HUMIDITY RATIO OF THE AIR ENTERING THE TOWER
      OMEGA1=.6221*PHI*PSAT(TDB)/(.491*PIN-PHI*PSAT(TDB))
      DO 23 I2=1,7
      RANGE =I2*5.0
      L=HEAT/RANGE
      AREA=L/LPERA
      TOUT=T2-RANGE
C     TOUT IS THE AVERAGE WATDR TEMPERATURE LEAVING THE TOWER
      APRCH=TOUT-TWB
      IF (APRCH.LE.4.0)GO TO 18
      WRITE(108,303)
  303 FORMAT('RANGE',5X,'WATER RATE',8X,'AREA',8X,'WATER TEMP OUT',
     $5X,'APPROACH')
      WRITE(108,304)RANGE,L,AREA,TOUT,APRCH
  304 FORMAT(F5.1,4(5X,1PE11.4))
      WRITE(108,16)
   16 FORMAT(6X,'L/G',12X,'NTU',12X,'HEIGHT',9X,'WIDTH',10X,
     $'LENGTH',9X,'AIR RATE',8X,'EVAPORATION RATE')
      DO 21 I3=1,25
      I=1
C***** C IS THE L/G RATIO
      C=0.2+(I3-1)*0.2
      G=L/C
      DO 208 M=1,51
      DO 208 N=1,51
  208 TIN(M,N)=HIN(M,N)=0.0
      DO 201 N=1,51
  201 TIN(1,N)=T2
      HTWB=ENTH(TWB,1.0,PIN)
      DO 202 M=1,51
  202 HIN(M,1)=HTWB
      EN1=HIN(1,1)
```

```
      TW=TIN(1,1)
      CALL ITERAT(EN1,TW,C,PIN,PHI,I)
      IF(I.GE.41)GO TO 21
      HIN(1,2)=EN1
      TIN(2,1)=TW
      IF(TW.LE.TOUT)GO TO 206
      I=2
C     HORIZONTAL PART STARTS HERE
  207 M=I
      DO 203 N=1,I-1
      EN1=HIN(M,N)
      TW=TIN(M,N)
      CALL ITERAT(EN1,TW,C,PIN,PHI,I)
      IF(I.GE.41)GO TO 21
      HIN(M,N+1)=EN1
  203 TIN(M+1,N)=TW
C     VERTICLE PART STARTS HERE
      N=I
      DO 204 M=1,I
      EN1=HIN(M,N)
      TW=TIN(M,N)
      CALL ITERAT(EN1,TW,C,PIN,PHI,I)
      IF(I.GE.41)GO TO 21
      HIN(M,N+1)=EN1
  204 TIN(M+1,N)=TW
      T1=0.0
      DO 205 K=1,I
  205 T1=T1+TIN(I+1,K)
      A=I
      T1=T1/A
      IF(T1.LE.TOUT)GO TO 206
      I=I+1
C**** THE NEXT STATEMENT CONTROLS THE MAXIMUM NUMBER OF UNITS
C**** IN EACH DIRECTION THAT THE CALCULATIONS ARE DONE FOR,
C**** AND THIS NUMBER MAY BE INCREASED IF THE DIMENSIONS
C**** ARE INCREASED.
      IF(I.GE.41)GO TO 22
      GO TO 207
C**** THE NEXT STATEMENT MUST CORRESPOND TO THE STEP SIZE
  206 NTUT=I*0.1
      EN2=0.0
      DO 11 K=1,I
   11 EN2=EN2+HIN(K,I+1)
      EN2=EN2/I
C**** THE ENTHALPY OF THE AIR LEAVING THE TOWER IS KNOWN,
C**** BUT THE TEMPERATURE AND THE HUMIDITY RATIO OF THE AIR
C**** EXITING IS UNKNOWN SO ITERATE TO FIND THESE BY ASSUMING
C**** THE RELATIVE HUMIDITY IS KNOWN.
      P2=.491*PIN
      PHI2=.95
      GT=TWB+10.0
      G2T=TWB+20.0
   12 GOMEG=.6221*PHI2*PSAT(GT)/(P2-PHI2*PSAT(GT))
      GH=.2408*GT+GOMEG*(.43*GT+1061.8)
      G2OMEG=.6221*PHI2*PSAT(G2T)/(P2-PHI2*PSAT(G2T))
```

```
      G2H=.2408*G2T+G2OMEG*(.43*G2T+1061.8)
      TAO=G2T+(GT-G2T)*(EN2-G2H)/(GH-G2H)
      OMEGA2=.6221*PHI2*PSAT(TAO)/(P2-PHI2*PSAT(TAO))
      HOUT=.2408*TAO+OMEGA2*(.43*TAO+1061.8)
      ERROR=ABS((EN2-HOUT)/EN2*100.0)
      IF(ERROR.LT.0.05)GO TO 13
      GT=G2T
      G2T=TAO
      GO TO 12
   13 EVAP=G*(OMEGA2-OMEGA1)
C**** KA IS THE PACKING COEFFICIENT AND MUST BE INPUT TO
C**** CORRESPOND TO THE PACKING BEING STUDIED.  KA IS
C**** OFTEN A FUNCTION OF THE WATER FLOW RATE.
      KA=44.0
      HEIGHT=NTUT*LPERA/KA
      WIDTH=HEIGHT*RATIO
      LENGTH=AREA/WIDTH
      WRITE(108,15)C,NTUT,HEIGHT,WIDTH,LENGTH,G,EVAP
   15 FORMAT(7(5X,1PE10.3))
      GO TO 21
   22 OUTPUT 'THE TOWER IS OVER FOUR TRANSFER UNITS TALL'
      GO TO 23
   21 CONTINUE
      GO TO 23
   18 OUTPUT  'THE APPROACH IS TOO SMALL'
      GO TO 20
   23 CONTINUE
   20 END
      SUBROUTINE ITERAT(EN1,TW,C,PIN,PHI,I)
      REAL NTU
C**** NTU MUST BE CHANGED TO CORRESPOND TO THE SPEP SIZE
       NTU=0.1
C      C IS THE L/G RATIO
       J3=0
       EN1D=ENTH(TW,1.0,PIN)-EN1
       X1=0.4*EN1D
       X2=0.8*EN1D
       Y1=X1-((ENTH(TW-NTU*X1,1.0,PIN)-(C*NTU*X1+EN1))*
      *(ENTH(TW,1.0,PIN)-EN1))/2.0
  101  Y2=X2-((ENTH(TW-NTU*X2,1.0,PIN)-(C*NTU*X2+EN1))*
      *(ENTH(TW,1.0,PIN)-EN1))/2.0
       J3=J3+1
       ENAD=X1-Y1*(X1-X2)/(Y1-Y2)
       ERROR=ABS((ENAD-X2)/ENAD)
       IF(ERROR.LT.0.001)GO TO 102
       IF(J3.GE.8)GO TO 103
       X1=X2
       X2=ENAD
       Y1=Y2
       GO TO 101
  103  ENAD=0.8*EN1D
       WRITE(108,10)I
    10 FORMAT('CONVERGENCE PROBLEMS AT ROW AND COLUMN I =',I3)
       I=41
  102  DELT=NTU*ENAD
```

```
      DELH=C*DELT
      TW=TW-DELT
      EN1=EN1+DELH
C     TW AND EN1 ARE NOW THE TEMPERATURE OF THE WATER AND THE
C     ENTHALPY OF THE AIR, LEAVING THE SQUARE, RESPECTIVELY.
      RETURN
      END
      FUNCTION ENTH(TDB,PHI,PIN)
      PATM=.491*PIN
      PSAT=EXP(-3.98136+5.2778E-2*TDB-1.4844E-4*TDB**2-4.7138E-7*TDB**3
     &+1.0307E-8*TDB**4-5.0898E-11*TDB**5+8.3956E-14*TDB**6-7.6653E-21
     &*TDB**7-3.5591E-23*TDB**8)
      PVAP=PHI*PSAT
      OMEGA=.6221*PVAP/(PATM-PHI*PSAT)
      ENTH=.2408*TDB+OMEGA*(.43*TDB+1061.8)
      RETURN
      END
```

APPENDIX III

PROGRAM FOR SIZING DRY COOLING TOWERS

This program uses conventional heat exchanger thoery in an unusual method for sizing dry cooling towers. The input variables are defined within the program.

This program is designed to be used in conjunction with either of the previous two when sizing wet/dry cooling towers. This program requires more input than the previous two due to the number of physical characteristics of a heat exchanger whcih must be predetermined.

```
      REAL KC,KE
      REAL LOF,MUA,MUW,MDA,MDW,NUW,NTU,M
      PSAT(TDB)=EXP(-3.98136+5.2778E-2*TDB-1.4844E-4*TDB**2
     $-4.7138E-7*TDB**3+1.0307E-8*TDB**4-5.0898E-11*TDB**5
     $+8.3956E-14*TDB**6-7.6653E-21*TDB**7-3.5591E-23*TDB**8)
C
C**** THE NUMBER OF HEAT EXCHANGER ELEMENTS NEEDED IS
C**** OUTPUT AS NUMBER.
C
C***** ALL UNITS IN LBM,FT, HR, AND LBF UNLESS OTHERWISE SPECIFIED
C
C**** THE VARIABLES THAT MUST BE INPUT ARE DEFINED AS
C**** FOLLOWS
C
C**** Q IS THE TOTAL HEAT TO BE DISSIPATED IN THE TOWER.
C**** DELTA IS THE FIN THICKNESS.
C**** DHA AND DHW ARE THE HYDRAULIC DIAMETER ON THE AIR
C**** AND THE WATER SIDE RESPECTIVELY.
C**** SIGMAA AND SIGMAW ARE THE RATIOS OF THE FREE FLOW
C**** AREA TO THE FRONTAL AREA ON THE RESPECTIVE SIDES.
C**** ALPHAA AND ALPHAW ARE THE RATIOS OF THE TOTAL
C**** HEAT TRANSFER AREA TO THE VOLUME FOR THE RESPECTIVE
C**** SIDES.
C**** AFPA IS THE RATIO OF THE TOTAL FIN AREA TO THE TOTAL
C**** HEAT TRANSFER  AREA ON THE AIR SIDE.
C**** LOF IS THE LINGTH OF THE FINS.
C**** CONDM IS THE CONDUCTIVITY OF THE METAL THE HEAT
C**** EXCHANGERS ARE CONSTRUCTED  OF.
C
      INPUT DELTA,DHA,SIGMAA,ALPHAA,SIGMAW,ALPHAW,AFPA
      INPUT LOF,CONDM
C
C**** TAI IS THE DRY BULB TEMPERATURE OF THE AIR ENTERING
C**** THE HEAT EXCHANGERS.
C**** TWI AND TWO ARE THE TEMPERATURES, IN FARENHEIT,OF
C**** THE WATER ENTERING AND EXITING THE HEAT EXCHANGERS
C**** RESPECTIVELY.
C**** PHI IS THE RELATIVE HUMIDITY OF THE AIR AS A DECIMAL.
C**** PATM IS THE ATMOSPHERIC PRESSURE IN INCHES OF MERCURY.
C
      INPUT Q,TAI,TWI,TWO,PHI,PATM
C
C**** X AND Y ARE THE DIMENSIONS ON THE AIR SIDE
C**** Y AND Z ARE THE DIMENSIONS ON THE WATER SIDE
      X=10.0;Y=5.0;Z=2.0
C     ENTRANCE CONDITIONS OF WATER AND AIR
      FLOW=Q/(TWI-TWO)
C     AC IS THE MINIMUM FREE FLOW AREA ON THE AIR SIDE
      AC=SIGMAA*X*Y
      TDB=TAI
      PVAP=PHI*PSAT(TDB)
      OMEGA=.6221*PVAP/(PATM-PVAP)
      CPW=1.0
      DO 15 I=1,10
      J=0
```

```
      REA=500.0+I*500.0
      OUTPUT REA
      TAO=(TWO+TAI)/2.0
C     CALCULATIONS START HERE
      TWA=(TWO+TWI)/2.0
   20 TAA=(TAI+TAO)/2.0
      T=TAA+459.69
C     CPV AND CPA ARE THE SPECIFIC HEATS OF WATER VAPOUR
C     AND HUMID AIR RESPECTIVELY.
      CPV=(19.86-597.0/SQRT(T)+7500.0/T)/18.016
      CPA=.2404/(1.0+OMEGA)+OMEGA*CPV/(1.0+OMEGA)
      J=J+1
      IF(J.EQ.10)GO TO 10
C     FINDING PROPERTIES OF THE FLUIDS AND THE METAL
      MUA=6.419*10.0**(-5.0)*TAA+.03913
      MUW=6.4046-8.8693*10.0**(-2.0)*TWA+5.243*10.0**(-4.0)
     *TWA**2.0-1.1391*10.0**(-6.0)*TWA**3.0
      PRA=.721*EXP(-.0002012*TAA)
      CONDW=.224*TWA**.1077
      MDA=REA*MUA*AC/DHA
C**** THE NEXT TWO EQUATION GIVE THE FILM COEFFICIENT ON
C**** THE AIR SIDE AND THEY MUST BE CHANGED TO CORRESPOND
C**** TO THE HEAT EXCHANGER GEOMETRY BEING STUDIED.
      STPR=.0967*REA**(-.3504)
      HA=STPR*MDA*CPA/(AC*PRA**.66667)
      M=SQRT(2.0*HA/(CONDM*DELTA))
      ETAF=TANH(M*LOF)/(M*LOF)
      ETAO=1.0-APPA*(1.0-ETAF)
      MDW=MDA*CPA*(TAO-TAI)/(CPW*(TWI-TWO))
      REW=DHW*MDW/(MUW*SIGMAW*Y*Z)
      PRW=CPW*MUW/CONDW
      IF(REW.LT.2000)GO TO 1
      NUW=0.0155*PRW**0.5*REW**0.83
      GO TO 2
    1 CONTINUE
      NUW=4.01
    2 CONTINUE
      HW=CONDW*NUW/DHW
      U=1.0/(1.0/(ETAO*HA)+ALPHAA/(ALPHAW*HW))
      C1=MDW*CPW
      C2=MDA*CPA
      IF(C1.LT.C2)GO TO 30
      CMIN=C2;CMAX=C1
      GO TO 40
   30 CMIN=C1;CMAX=C2
   40 RATIO=CMIN/CMAX
      NTU=U*X*Y*Z*ALPHAA/CMIN
C***** THE EFFECTIVENESS TO NTU RELATIONS MUST MATCH THE TYPE OF HEAT
C***** EXCHANGER FLOW
      IF(C1.LT.C2) GO TO 5
      GAMMA=1.0-EXP(-NTU*RATIO)
      EFF=1.0-EXP(-GAMMA/RATIO)
      GO TO 6
    5 GAMMAP=1.0-EXP(-NTU)
      EFF=(1.0-EXP(-GAMMAP*RATIO))/RATIO
```

```
    6 CONTINUE
      EFFG=EFF
      IF(C1.LT.C2)GO TO 11
      EFFT=(TAO-TAI)/(TWO-TAI)
      GO TO 14
   11 EFFT=(TWI-TWO)/(TWI-TAO)
   14 IF(J.EQ.1)GO TO 12
      X2=EFFT-EFFG
      T3=(T2*X1/X2-T1)/(X1/X2-1.0)
      T1=T2
      T2=T3
      X1=X2
      GO TO 13
   12 T2=(TAO+TAI)/2.0
      T1=TAO;X1=EFFT-EFFG
   13 TAO=T2
      IF(T2.LE.TAI)T2=TAI+5.0;TAO=T2
      IF(ABS(X1).LT.0.0001)GO TO 10
      GO TO 20
   10 CONTINUE
      HEAT=CPA*MDA*(TAO-TAI)
      NUMBER=FLOW/MDW+1
      OUTPUT EFF,HEAT ,NUMBER
      WRITE(108,4)
    4 FORMAT (3X,'MDA',8X,'ETAF',7X,'ETAO',7X,'MDW',8X,'REW',8X,'NUW',
     $9X,'U',8X,'RATIO',7X,'NTU',8X,'TAO2')
      WRITE(108,3)MDA,ETAF,ETAO,MDW,REW,NUW,U,RATIO,NTU,TAO
    3 FORMAT(10(1PE11.4))
      OUTPUT 'J IS THE NUMBER OF ITERATIONS TO GET THE EFFECTIVENESSES
     $TO MATCH'
      OUTPUT J
      G=MDA/AC
      GC=32.2*3600.0**2
      J=0
      FRTN=.4143*REA**(-.4518)
C***** ENTRANCE AND EXIT LOSS COEFFICIENTS MUST BE CHANGED TO CORRESPOND
C***** WITH THE HEAT EXCHANGER CORE BEING USED
      IF(REA.GE.2000.0)GO TO 19
      KC=.8063-.0671*SIGMAA-.1868*SIGMAA**2-.1537*SIGMAA**3
      KE=1.008-2.486*SIGMAA+1.213*SIGMAA**2-.1335*SIGMAA**3
      GO TO 21
   19 KC=.4854+.08452*SIGMAA-.5519*SIGMAA**2+.03146*SIGMAA**3
      KE=.9891-1.935*SIGMAA+.9514*SIGMAA**2-4.139E-3*SIGMAA**3
   21 CONTINUE
      P1=PATM*144.0
      P2=.9*P1
      PHI1=PHI
      PG1=PSAT(TAI)*144.0
      PG2=PSAT(TAO)*144.0
      TAIR=TAI+459.69
      TAOR=TAO+459.69
      SV1=1545.0*TAIR/((P1-PHI1*PG1)*28.97+PHI1*PG1*18.016)
   17 SV2=1545.0*TAOR*P1/(P2*((P1-PG1*PHI1)*28.97+18.016*PG1*PHI1))
      SVM=(SV1+SV2)/2.0
      PDROP=P1-P2
```

```
      RHS=G**2*SV1/(2.0*GC)*((KC+1.0-SIGMAA**2)+2.0*(SV2/SV1-1.0)+
     *FRTN*Z*4.0/DHA*SVM/SV1-(1.0-SIGMAA**2-KE)*SV2/SV1)
      PEQ=PDROP-RHS
      J=J+1
      IF(J.EQ.1)GO TO 16
      PEQ2=PEQ
      P22=P2
      P2=P21-PEQ1*(P22-P21)/(PEQ2-PEQ1)
      IF(ABS(PEQ).LT.0.01)GO TO 18
      P21=P22
      PEQ1=PEQ2
      GO TO 17
   16 P21=P2
      PEQ1=PEQ
      P2=.7*P1
      GO TO 17
   18 CONTINUE
      WRITE(108,23)
   23 FORMAT(5X,'ITERATIONS',8X,'P1',12X,'P2',7X,'PRESS DROP')
      WRITE(108,22)J,P1,P2,PDROP
   22 FORMAT(4(5X,1PE9.2))
   15 CONTINUE
      END
```

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