Preliminary Evaluation of Wet/Dry Cooling Concepts for Power Plants

---

January 1976

Prepared for the Energy Research
and Development Administration
under Contract E(45-1):1830

Battelle
Pacific Northwest Laboratories

MASTER

BNWL-1969

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

PACIFIC NORTHWEST LABORATORY
operated by
BATTELLE
for the
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Under Contract E(45-1)-1830

Printed in the United States of America
Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22151

Price: Printed Copy ~~\$6.00~~; Microfiche \$2.25
5.00

PRELIMINARY EVALUATION OF WET/DRY COOLING
CONCEPTS FOR POWER PLANTS

by

W. V. Loscutoff

January 1976

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Battelle
Pacific Northwest Laboratories
Richland, Washington 99352

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

CONTENTS

LIST OF FIGURES	v
LIST OF TABLES	v
1.0 INTRODUCTION	1
2.0 SUMMARY AND CONCLUSIONS	3
3.0 EVALUATION SCOPE	5
3.1 WET/DRY COOLING SYSTEM COMBINATIONS	5
3.2 NOMENCLATURE	8
3.3 METHOD OF ANALYSIS	10
4.0 ANALYSIS OF WATER FLOW SEQUENCES	15
4.1 FACTORS IN ANALYSIS	15
4.2 COMPARATIVE HEAT REJECTION CAPACITIES	16
4.3 CONCLUSIONS	25
5.0 ANALYSIS OF AIR FLOW SEQUENCES	29
5.1 THERMODYNAMIC COMPARISON	29
5.2 CONCLUSIONS	32
6.0 WATER EVAPORATION RATES	35
6.1 METHODOLOGY	36
6.2 ANALYSIS	38
6.3 CONCLUSIONS	49
7.0 SEPARATE AND HYBRID WET/DRY SYSTEMS	51
7.1 COMPARISON OF SEPARATE AND HYBRID SYSTEMS	51
7.2 CLOSED AND OPEN LOOP WATER SYSTEMS	54
7.3 WET/DRY TOWER COMBINATIONS	58
8.0 MAJOR RESULTS AND CONCLUSIONS	63
REFERENCES	67
BIBLIOGRAPHY	69
APPENDIX A	
EFFECT OF DRY AND WET TOWER ITDs ON COMBINED TOWER PERFORMANCE	A-1

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

LIST OF FIGURES

1	Combined Wet/Dry Cooling Systems	6
2	Exemplary Combined Cooling System Represented by Figure 1A	7
3	Operating Characteristics for a Specified Dry Cooling Subsystem Design	12
4	Operating Characteristics for a Specified Wet Cooling Subsystem Design	12
5	Modified Wet/Dry Cooling Systems	20
6	Performances of Systems B, C1, and C2 Compared to System A	22
7	Performance of Systems B, C1, and C2 Compared to System A with Optimized Wet Towers	26
8	Deluge and Adiabatic Cooling Configurations	35
9	Counterflow Cooling Tower Mechanism	38
10	Temperature-Enthalpy Trajectory for Air in System A Wet Cooling Tower	41
11	Temperature-Enthalpy Trajectory for Air in System B Wet Cooling Tower	43
12	Temperature-Enthalpy Trajectory for Air in System F Wet Cooling Tower	45
13	Temperature-Enthalpy Trajectory for Air in an Adiabatic Cooling System	48
14	Closed Water Loop Systems	55
15	Open Water Loop Systems	56
16	Cooling Modules	59
17	Cooling Tower Configurations	61

LIST OF TABLES

1	Cooling Tower Performance Comparisons	39
2	Cooling Tower Configuration Comparisons	53
3	Comparison of Closed and Open Loop Systems	57
4	Hybrid and Separate Tower Comparisons	62

PRELIMINARY EVALUATION OF WET/DRY COOLING CONCEPTS FOR POWER PLANTS

1.0 INTRODUCTION

The combination of a dry tower with a wet tower forms a new cooling system with a number of attractive features for electric power plants. Dry and wet towers differ in almost every important aspect such as size, water consumption, sensitivity to ambient air temperatures, and thermal discharges. The possibility of combining these characteristics in an optimal manner holds promise for an economic and effective cooling system.

Wet and dry cooling towers can be linked in a variety of possible combinations. Classification of combinations based on the flow patterns of water and air--series or parallel connection on the water side, and series or parallel connection on the air side--yields a large number of combined cooling systems. Not all are sequenced to perform in an effective manner. This report documents a thermodynamic analysis of the major wet/dry cooling tower combinations, and presents an evaluation of the effectiveness of each combination in terms of tower operation, water consumption, and other relevant parameters.

The basic questions addressed in this study are:

- What is the most effective water flow sequence for heat rejection and what factors control effectiveness?
- What is the most effective air flow sequence and what factors affect it?
- Which combinations provide maximum heat rejection with minimal water consumption?
- What is the recommended state-of-the-art wet/dry combination for use by power plants?
- What are the fundamentals of wet surface heat exchangers?

- What are advantages and utility of wet surface heat exchangers?
- What are the differences between hybrid wet/dry cooling towers and separate wet and dry towers?

Because of the complexity of cooling systems for electric power plants, all results presented are qualitative in nature. Since the equations used are developed with the accepted principles of mass and energy conservation, it is expected that the final results are not far from what may be obtained in real operation.

2.0 SUMMARY AND CONCLUSIONS

A large number of possible combinations of air and water flow arrangements and sequences were analyzed in this study. First, the heat transfer effectiveness of each was established, based on the assumption of constant size of the dry and wet subsystems. Next, the water consumption required to augment a constant-sized dry subsystem for maintaining cooling capability at a dry bulb temperature in excess of the design conditions for the dry system alone was determined.

Although a high degree of approximation was used in some phases of this preliminary analysis of dry/wet cooling system combinations, the results are qualitatively valid and point to a number of significant conclusions relative to the preferred arrangement of the dry and wet subsystems in dry/wet heat rejection systems. These conclusions are based on the assumption that the wet cooling is used only to augment the dry cooling capability and that minimum consumption of water is a primary consideration.

1. Parallel flow of air through the wet and dry systems is superior to series air flow. This follows from the fact that the most favorable air flow rates in wet and dry towers are considerably different. Thus an excess of air must be pumped through the wet sections when dry and wet sections are used in series. Furthermore, flowing the air in series leads to a decrease in Initial Temperature Difference (ITD) and heat rejection capability of the second subsystem connected in series.
2. The choice between series and parallel water flow configurations is dependent primarily upon economics and is nearly independent of thermodynamic considerations. A series water flow pattern through the dry and wet subsystems allows use of a standard-sized condenser but a larger wet tower. Water should flow through the dry subsystem first. There are two reasons for this sequence. First, the effectiveness of dry before wet is somewhat higher than the reverse, due to the relationship between ITD and dry bulb temperature in dry towers and between ITD and the wet bulb temperature in wet towers. This reason may not be true in a very humid climate. Second,

it is desirable to maintain the full effectiveness of the dry section to minimize water evaporation losses from the wet tower.

3. A parallel water flow pattern is thermodynamically effective if an oversized condenser is used such that any flow necessary for the wet section is obtained from a section of the condenser which is used only for supplemental cooling. This configuration requires a smaller wet tower than used in a series water flow pattern, but at the cost of a larger condenser.
4. There is no difference between separate dry and wet towers and hybrid dry/wet towers with parallel air flow from the standpoint of thermodynamics and water consumption. The primary incentive for a hybrid design is capability to provide plume abatement. The need for plume abatement, as well as the economics of plant construction at the particular site, determine which combination will be used. Separate towers seem to be simpler to construct and avoid potential problems from recirculation of damp air and surface corrosion.
5. Gains in overall heat transfer rates due to surface wetting of dry heat exchangers indicate that wet surface coolers have promise as dry/wet systems, if problems of corrosion and crud deposition can be avoided. Water evaporation rates for the wetted surface are comparable to those of conventional wet towers; however, the overall water consumption is somewhat higher since the dry surface area is reduced, i.e., converted to wet surface area, so the fraction of heat removed by evaporation must be higher than in other cases where the area of dry surface remains constant.

3.0 EVALUATION SCOPE

3.1 WET/DRY COOLING SYSTEM COMBINATIONS

The combinations of wet and dry towers analyzed in this study are shown in Figure 1. Classified on the basis of water and air flow circuits, they are:

1. Parallel path air flow
  - A. series water flow dry-before-wet
  - B. series water flow wet-before-dry
  - C. parallel water flow
  - D. series water flow dry-before-wet with a fraction of water bypassing the wet tower
2. Series path air flow
  - E. air flow dry-before-wet, parallel water flow
  - F. air flow dry-before-wet, series water flow dry-before-wet
  - G. air flow wet-before-dry, parallel water flow
  - H. air flow wet-before-dry, series water flow dry-before-wet

At least two major combinations of wet and dry towers are not included in the group shown in Figure 1. These are the counterparts of F and H shown in Figure 1, with water flow reversed:

3. Series path air flow
  - I. air flow dry-before-wet, series water flow wet-before-dry
  - J. air flow wet-before-dry, series water flow wet-before-dry

Analysis of their equivalents for the parallel air flow combinations should provide sufficient information for their evaluation. Hence, they are excluded from the following general analysis.

PARALLEL PATH AIR FLOW

— WATER FLOW
 - - - AIR FLOW

SERIES PATH AIR FLOW

FIGURE 1. Combined Wet/Dry Cooling Systems

The schematics in Figure 1 represent only heat flow paths and not the means of heat transfer. For example, Figure 1A represents a system in which water flows from the dry tower directly into the wet tower where it is distributed onto tower fill and collected in a basin. Figure 1A also represents a system in which the water from the dry tower flows into a water-to-water heat exchanger where it is cooled by a separate water loop flowing through a wet tower as shown in Figure 2. Each of the above systems has advantages and disadvantages in terms of initial temperature differences, economics of water contamination, etc., which will not be discussed in this report. One fundamental difference--finite terminal temperature resulting from use of heat exchangers--should not affect the linear analysis.

FIGURE 2. Exemplary Combined Cooling System Represented by Figure 1A

Operation of the combined systems is predicated on the assumption that the cooling capacity of dry towers is capable of handling all cooling requirements under normal, or design, conditions. The wet towers are brought into operation only when the design temperatures are exceeded. The primary objective of the combined cooling system is to maintain the turbine back pressure at a level of about 5 in. Hg absolute, which is equivalent to steam temperature of 135°F.

3.2 NOMENCLATURE

The various terms used in this study are defined as follows:

A = area of heat exchange surface, ft^2

a = area of water droplet interface, ft^2/ft^3

b = conductance number, $\text{lb}/\text{ft}^3\text{-hr}$

$C = \dot{m}c_p$, heat capacity flow rate, $\text{Btu}/\text{hr-}^\circ\text{F}$

c_{pa} = specific heat constant of air, $\text{Btu}/\text{lb-}^\circ\text{F}$

c_{pw} = specific heat constant of water, $\text{Btu}/\text{lb-}^\circ\text{F}$

F_c = crossflow correction factor

h = enthalpy of air at wet bulb temperature, Btu/lb

h_i = enthalpy of saturated air at temperature T , Btu/lb

Δh = heat difference between air and water across spray, Btu/lb

K = overall mass transfer coefficient between saturated air and main air stream.

L = latent heat of evaporation, Btu/lb

$\dot{m}$ = flow rate, lb/hr

ρ_w = density of water, lb/ft^3

η = effectiveness of a heat exchanger

Q = heat dissipated, Btu/hr

r = radius, ft

T = temperature, $^\circ\text{F}$

t = time, sec

U' = heat transfer coefficient, Btu/hr-ft²-°F

V = packing volume in wet tower, ft³

w = amount of water, lb

$\dot{w}_{ev}$ = water evaporated, lb/hr

The subscripts used are defined below.

a refers to air

D refers to dry tower

w refers to water

W refers to wet tower

aDi = air into dry tower

aDo = air out of dry tower

aWi = air into wet tower

aWo = air out of wet tower

wDi = water into dry tower

wDo = water out of dry tower

wWi = water into wet tower

wWo = water out of wet tower

T = total

The superscript $*$ refers to wet bulb temperature.

3.3 METHOD OF ANALYSIS

Cooling system performance can be established thermodynamically by balancing the cooling loads of air and water, and by summing the cooling loads in the wet and dry towers. The operating conditions can be verified by the simultaneous solution of heat and mass balance equations. Two heat

balance equations and a heat transfer equation can be written for each dry section and wet section. Mathematically for the dry section they are:

air heat balance

$$Q_D = \dot{m}_a c_{pa} (T_{aDo} - T_{aDi}) \quad (1)$$

water heat balance

$$Q_D = \dot{m}_w c_{pw} (T_{wDi} - T_{wDo}) \quad (2)$$

heat transfer

$$Q_D = UA_D \frac{(T_{wDi} - T_{aDo}) - (T_{wDo} - T_{aDi})}{\ln \frac{T_{wDi} - T_{aDo}}{T_{wDo} - T_{aDi}}} F_c \quad (3)$$

For the wet section, the equations are:

air heat balance

$$Q_W = \dot{m}_a c_{pa} (T_{aWo} - T_{aWi}) + \dot{w}_{ev} L \quad (4)$$

water heat balance (neglecting evaporation)

$$Q_W = \dot{m}_w c_{pw} (T_{wWi} - T_{wWo}) \quad (5)$$

heat transfer

$$Q_W = bV(\Delta h) \quad (6)$$

and

total cooling load

$$Q_T = Q_D + Q_W$$

Although the above equations involve a degree of approximation in some cases, they do preserve the physics of the system and are tractable in solving for an optimal configuration.

The heat duty of a dry tower may also be expressed as

$$Q_D = \eta C_{\min} (T_{wDi} - T_{aDi}) = \eta C_{\min} (ITD), \quad (7)$$

where $C_{\min}$ is the smaller of $\dot{m}_{wD} c_{pw}$ and $\dot{m}_{aD} c_{pa}$. The effectiveness η is a function of parameters NTU and R, defined as

$$NTU = \frac{AU}{C_{\min}} \quad (8)$$

$$R = \frac{C_{\min}}{C_{\max}} \quad (9)$$

where $C_{\max}$ is the larger of $\dot{m}_{wD} c_{pw}$ and $\dot{m}_{aD} c_{pa}$.

Additional information used in this study is the relationship between the Initial Temperature Difference (ITD) and heat rejection capacities of dry and wet cooling subsystems. Transforming the data used by Croley, et al. (Ref. 1, pp. 34, 43) into the convenient form shown in Figures 3 and 4, the fundamental relationships are found to be

dry tower

$$Q_D = K_1 (ITD)^{K_2} \quad (10)$$

wet tower

$$Q_W = K_3 (ITD)^{K_4} \quad (11)$$

Values used in this study are $K_2 = 1.0$ for the dry tower and $K_4 = 1.8$ for the wet tower. The last corresponds to the $T_{wb} = 65^\circ\text{F}$ curve in Figure 4. The two values represent a mechanical draft dry tower and a natural draft wet tower, respectively. Corresponding values for natural draft dry and mechanical draft wet are 1.3 and 1.4 respectively. The values used for this study were selected to amplify the effect of the ITD power relationship. An additional curve for $K_2 = 1.3$ is plotted in Figure 3 for comparison.

FIGURE 3. Operating Characteristics for a Specified Dry Cooling Subsystem Design(1)

FIGURE 4. Operating Characteristics for a Specified Wet Cooling Subsystem Design(1)

The operation of any cooling system is dynamic by nature. A true steady-state condition is never obtained. For the time intervals considered, quasi-equilibrium conditions are assumed and performance characteristics of the proposed system are predicted on this basis.

Heat losses from piping and structure are small compared with the heat load in the system and are ignored here.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

4.0 ANALYSIS OF WATER FLOW SEQUENCES

In this section the effect of the sequence of water flows into wet and dry towers on the overall performance of a combined wet/dry system is examined. For convenience, only parallel path air flow systems are analyzed. However, the results may be extrapolated to series path air flow systems.

Three configurations were selected for the analysis. Shown in Figure 1, they are:

- System A - series water flow dry-before-wet;
- System B - series water flow wet-before-dry;
- System C - parallel water flow.

The general approach in the study is to assume fixed water and air inlet temperatures and fixed sizes of dry and wet sections. Total water flow rate is treated as a variable under specific restrictions. Thus, only the total heat rejection capacity must be determined and this magnitude serves as a basis of comparison. For convenience, heat rejection capacity of System A is used as a reference. Analysis does not differentiate between dominant and supporting systems. However, any performance variations due to role differences will be pointed out.

4.1 FACTORS IN ANALYSIS

To determine which sequence of water flows is most effective for combined wet/dry systems, two major factors affecting any wet/dry system combination must be considered. Each has a different effect on the performance of a combined system.

The first factor arises from dependence of heat rejection capacity on ITD raised to a power

$$Q_W = K_3 (ITD)^{K_4}$$

where $K_4 > 1$. For $T_{wb} = 65^\circ\text{F}$, Figure 4 yields $K_4 \approx 1.8$. This is due to higher vapor pressure of water at elevated temperatures and is demonstrated

for a specific wet cooling subsystem design in Figure 4. Its implications are that, for a given ITD, a wet system operates more effectively at higher inlet water temperatures. The relationship between Q and ITD for a mechanical draft dry tower is nearly linear, as shown in Figure 3; Li⁽²⁾ gives $K_2 = 1.3$ for a natural draft dry tower. Hence, the natural conclusion is to take advantage of the high effectiveness of wet towers at high water temperatures and to design water flows that proceed into wet, then dry, subsystems sequentially as shown in Figure 1B.

The second factor is due to the difference between wet and dry bulb temperatures and the fact that the ITD for a dry tower is based on dry bulb temperatures while the wet tower ITD is based on wet bulb temperatures. The effect of this difference on effectiveness of wet/dry combinations is shown by direct qualitative comparison of System A and System B in the Appendix. The results are that, on the basis of this factor alone, the more effective combination of systems, in terms of heat rejection capacity, is to flow water first into the dry tower and then through a wet system as shown in Figure 1A. The only condition under which this factor has no effect is when the dry and wet bulb temperatures are the same.

In summary, it is apparent that two factors dominate the performance of a combined wet/dry system. One, the difference in relationships between Q and ITD for dry and wet systems, favors flowing water first into a wet tower, then into a dry tower to take advantage of the effectiveness of wet towers at high ITDs. The second, resulting from difference in wet and dry bulb temperatures, yields a more effective combination when water flows first into the dry tower, then into the wet tower. It is thus necessary to establish conditions at which one factor or the other dominates. This will be undertaken in the section which follows.

4.2 COMPARATIVE HEAT REJECTION CAPACITIES

In this section, specific performance characteristics given by Croley, et al.⁽¹⁾ are assigned to wet and dry cooling towers. The objective is to determine relative magnitudes of the two factors discussed previously for

the parallel air flow combinations shown in Figure 1. Because of the complexity of wet and dry tower interaction, parameters such as flow rates and heat rejection capacities are quantified to generate comparative results.

The method of comparing heat rejection capacities involves the following steps:

- determination of the amount of heat rejection capacity to be supplied by the wet system when the design ambient air temperature is exceeded,
- establishment of the flow and temperature conditions on the wet and dry systems in the configuration of System A, and
- determination of the total heat rejection capacities for other configurations under flow and temperature conditions identical to the capacity of System A. A sample of calculations involved in the process is shown. Results are presented in graphical form.

System C, the parallel water configuration shown in Figure 1, is also evaluated. Two water flow rate conditions are used to evaluate System C. In the first condition, the total water flow rate through the dry and wet sections is equal to the flow rate in Systems A and B. For the second condition, water flow rate is held constant in the dry section; any flow through the wet section is due to additional water being pumped. The total flow rate is the sum of the flows through each section and is greater than that for the other configurations, implying a larger condenser. These test conditions are identified as System C1 and System C2, respectively.

Increasing water flow rate is justified as follows. In Systems A and B, an additional pressure loss is incurred when the wet towers are brought on line. When a wet system is brought on line in a parallel fashion as in System C, there is a decrease in the pressure drop when the water flow is split between the wet and dry towers. Thus, by increasing the total flow rate by maintaining the flow rate in the dry section and supplying additional flow to the wet tower, System C2 matches the water pumping requirements of Systems A and B more closely. System C2, however, does require an oversized condenser.

To complete this analysis, a dry cooling tower with the following characteristics is used:

Heat rejection capacity	6.8×10^9 Btu/hr
Air flow rate	1.0×10^9 lb/hr
Circulating water flow rate	0.3×10^9 lb/hr
Design temperature	75°F
Water inlet temperature	135°F
Design ITD	60°F
Range	20°F

A supplementary wet tower has the following basic design characteristics:

Air flow rate (nominal)	0.15×10^9 lb/hr
Circulating water flow rate	0.3×10^9 lb/hr

The air flow rate is not fixed but varies depending on the dry/wet configuration being examined. For parallel air flow systems, it is at the stated number giving a ratio of water to air of

$$\dot{m}_w / \dot{m}_a = 2.0.$$

However, when a wet tower is put in series with a dry tower, the air flow rate through the wet tower is the same as that through the dry tower. Under these circumstances, the water-air flow rate ratio is

$$\dot{m}_w / \dot{m}_a = 0.3$$

if all air passing through the dry section must pass through the wet section.

Given an ambient temperature and the heat load on the dry tower, the amount of heat rejection capacity which must be supplemented by alternate means may be determined. Assuming the linear relationship given by equation (10), the supplementary heat rejection capacity is

$$Q_w = \left(1 - \frac{135^\circ - T_{amb}}{135^\circ - 75^\circ} \right) Q_{design} \quad (12)$$

Examples of calculation procedures are shown below for one ambient condition. The ambient test condition selected was one with temperatures of 95°F dry bulb, 65°F wet bulb. For this condition, the heat rejection capacity of dry towers in terms of designed heat duty is

$$Q_D = \frac{135^\circ - 95^\circ}{135^\circ - 75^\circ} Q_{\text{design}} = \frac{2}{3} Q_{\text{design}} \quad (13)$$

Thus, $1/3 Q_{\text{design}}$ must be obtained from another system.

System A

Assume that the cooling tower is subjected to the test condition and that a wet tower of the necessary size is available for System A. The following temperature profiles will then describe System A wet and dry towers at test conditions.

In this configuration, the wet tower is not utilized most effectively for the specified air and water flow rates since the temperature range is only 6.67°F. For the amount of heat to be rejected, the wet tower water and air flow rates could be reduced by a factor of three, raising the range to 20°. A schematic for this system, in which part of the water bypasses the wet tower, is shown as System A in Figure 5.

SYSTEM A (MODIFIED)

SYSTEM B (MODIFIED)

SYSTEM C1

SYSTEM C2

FIGURE 5. Modified Wet/Dry Cooling Systems

System B

If the above wet and dry towers are now rearranged into the System B configuration, the heat rejection capacity of the new configuration is not known and must be calculated. The heat rejection capacity of the wet tower is now determined from ITD of $135-65^\circ = 70^\circ\text{F}$ as compared to $\text{ITD} = 56.7^\circ\text{F}$ for System A. From Figure 4, the heat rejection capacity of a wet tower in System B is

$$Q_W = 0.49 Q_{\text{design}}$$

The temperature profiles appear as follows:

Here, Q_D is computed by assuming validity of equation (10):

$$Q_D = \frac{125.2-95}{135-95} \left(\frac{2}{3} Q_{\text{design}} \right) = 0.50 Q_{\text{design}}$$

Thus, the total heat rejection capacity of System B is

$$Q_T = 0.49 Q_{\text{design}} + 0.50 Q_{\text{design}} = 0.99 Q_{\text{design}}$$

and there is a 1 percent reduction in overall performance of System B when compared with System A.

For other ambient conditions, especially changes in wet bulb temperature, System B is more effective than System A for humid conditions and less effective for very dry conditions. This is shown in Figure 6. It should

FIGURE 6. Performances of Systems B, C1, and C2 Compared to System A

be noted here that reduction in overall performance is larger if Q is a function of $(ITD)^{1.3}$ instead of the linear assumption made here for the dry tower. Q_D for this case is 0.46 and the total heat rejection capacity is $Q_T = 0.95 Q_{\text{design}}$.

System C

When the above wet and dry towers are arranged into the parallel air and water flow configuration of System C, the overall heat rejection capacities are greater or less than that of System A depending upon the total water flow rate. If the water flow rate is kept constant and distributed between the wet and dry towers in some ratio proportional to heat rejection capacity, say 2 to 1, the temperature profiles will be as follows:

For a particular dry tower design, there will be a loss in the heat rejection capacity due to a decrease in water flow rate. From equation (7) and for the specified dry tower characteristics and effectiveness, η , the change in Q_D is from $Q_D = 0.67 Q_{design}$ to

$$Q_D = \left(\frac{2}{3} Q_{design} \right) \left(\frac{\eta_c}{\eta_a} \right) \left(\frac{C_{min\ c}}{C_{min\ a}} \right) = \left(\frac{2}{3} \right) (1.10) (0.74) = 0.54 Q_{design}$$

The wet tower for System C1 will have one-third the flow rate of the tower in System B with identical temperature conditions. Assuming that the water-to-air flow rate ratio is maintained, the heat rejection capacity is assumed to be reduced by one-third to

$$Q_W = \frac{1}{3} \left(Q_W, \text{System B} \right) = 0.16 Q_{design}$$

The total heat rejection capacity for System C1 then becomes

$$Q_T = 0.70 Q_{\text{design}}$$

A better performance of the parallel water flow configuration is obtained when the water flow rate through the dry tower is maintained at design conditions and any water for the wet system is provided by increasing the water flow rate through the condenser. Under these conditions, the heat rejection capacity of the dry tower is the same as that of System A, or

$$Q_D = 0.67 Q_{\text{design}}$$

If the water flow rate through the wet tower is made equal to that of Systems A and B, then the wet tower capacity is equal to that of System B,

$$Q_W = 0.49 Q_{\text{design}}$$

The total heat rejection capacity is

$$Q_T = 1.16 Q_{\text{design}}$$

Note that the tower sizes, air flow rates, and water flow rates are identical for Systems A, B, and C2. The only difference is in the water temperatures since Systems A and B require the water to flow through cooling subsections sequentially while System C2 has water flowing through cooling systems in parallel.

Figure 6 presents the results obtained for ambient conditions of 95°F dry bulb temperature and wet bulb between 45°F and 85°F for the four wet/dry cooling configurations described. The results are given in terms of ratios to performance of System A.

One of the problems of series water or air flow through a wet/dry cooling system is the mismatch in the nominal water and air flow rates for the wet and dry subsystems. For example, the design conditions for System A specify a given water flow rate through the dry tower. When the design temperature is exceeded so that only two-thirds of design heat rejection capacity is available and a wet tower is brought on line, all of the water flowing into the dry tower also flows into the wet tower. The normal range

for both wet and dry towers is about 20°F. For wet subsections in all four cases examined, they are about 6°F. Thus, the wet tower has three times as much water flowing through it as is necessary for the amount of heat to be rejected. A more realistic case is to match water and air flow rates with desired heat rejection capacities and to design wet towers for optimal performance with a range of about 20°F. These configurations are shown in Figure 5, in which the Figure 1 schematics are redrawn to emphasize the dominance of dry subsections. Following the procedure just described, the results show that Systems A and B remain identical relative to each other. System C1, however, takes on the following values:

$$T_{\text{wet bulb}} = 45^{\circ}\text{F}, (Q_T)_{C1} = (0.54 + 0.49) (Q_T)_A = 1.02 (Q_T)_A$$

$$T_{\text{wet bulb}} = 65^{\circ}\text{F}, (Q_T)_{C1} = 1.03 (Q_T)_A$$

$$T_{\text{wet bulb}} = 85^{\circ}\text{F}, (Q_T)_{C1} = 1.06 (Q_T)_A$$

For System C2, the results are

$$T_{\text{wet bulb}} = 45^{\circ}\text{F}, (Q_T)_{C2} = 1.15 (Q_T)_A$$

$$T_{\text{wet bulb}} = 65^{\circ}\text{F}, (Q_T)_{C2} = 1.16 (Q_T)_A$$

$$T_{\text{wet bulb}} = 85^{\circ}\text{F}, (Q_T)_{C2} = 1.19 (Q_T)_A$$

These results are plotted in Figure 7. It must be realized, however, that in this case the wet towers are very large with a packing volume of approximately three times the size of the baseline cases.

4.3 CONCLUSIONS

In this section, the primary factors determining choice of water flow sequences into wet and dry towers in combination have been examined. When System B is compared to System A and Q is a linear function of ITD, the two major factors have opposite and nearly equal magnitudes, thus negating the

FIGURE 7. Performance of Systems B, C1, and C2 Compared to System A with Optimized Wet Towers

effect of each other. However, when the dry tower follows the law $Q = K (ITD)^{1.3}$, System A has distinct advantages over the other combinations discussed in terms of overall effectiveness if the total flow rate is maintained at design level when supplementary cooling is added. If the supplementary system has its own pumps and condenser, then System C2 is most effective in rejecting heat. The choice of System A or System C2 depends primarily on economics of condensers and wet towers.

Comparison of System A with System B suggests that, in very humid climates, water should be directed to flow first into the wet tower, then into the dry tower. At these climatic conditions, the greater effectiveness of wet towers at higher inlet water temperatures provides greater combination effectiveness. In dry climates, water should flow first into the dry tower, then into the wet to take advantage of the greater relative driving force on the combined system due to the large difference in dry bulb and wet bulb temperatures. The latter factor will dominate if Q follows a power law for dry towers.

Compared with System A, System C1 is not used effectively because the dry tower is utilized at less than full capacity when the supplementary system is put on line. For this reason it is not a good candidate for a combined cooling system.

System C2 uses both wet and dry towers most effectively (maximum flows and maximum ITDs) but requires a separate condenser to handle the increased total flow rate. It appears to be an excellent candidate for a combined cooling system if the condenser costs are sufficiently low or if a closed water flow system is required for dry towers.

The conclusions derived in this section can be extended to systems with series air flow configuration. The major conclusions suggest that if similar power relationships are assumed between Q and ITD for both dry and wet towers, the highest possible ITD must be maintained on dry towers by supplying water at the highest temperature to the dry tower. Hence, water must be sequenced to flow first into the dry tower.

For the extreme case, when $Q_W = K(\text{ITD})_W^{1.8}$ and $Q_D = K(\text{ITD})_D^{1.0}$ for wet and dry towers respectively, the foregoing conclusions do not provide a basis for determining water flow sequence. For such conditions, other criteria, such as water evaporation rates, must be developed.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

5.0 ANALYSIS OF AIR FLOW SEQUENCES

The effect of air flow sequence on the heat rejection capacities of combined dry/wet cooling systems is analyzed in this section. The methodology is identical to that used in Section 4. Furthermore, the same example is used for quantitative results with additional assumptions stated when appropriate.

Configurations examined in this section are System E, System F, System G, and System H. All are shown schematically in Figure 1. As in the preceding section, System A is used as a reference or standard for comparison. The analysis approach first assumes fixed water and air flow rates and fixed sizes of wet and dry cooling sections. The total heat rejection capacities are then calculated and compared with the capacity of System A.

5.1 THERMODYNAMIC COMPARISON

The basic parameters of wet and dry units used in this analysis are identical to those in Section 4. The reference system is again System A with all major conditions as given in Section 4 at the test condition:

- $Q_D = \frac{2}{3} Q_{\text{design}}$
- $Q_W = \frac{1}{3} Q_{\text{design}}$
- $T_{aDi} = 95^\circ\text{F}$
- $T_{aDo} = 111.7^\circ\text{F}$
- $T_{aWi} = 65^\circ\text{F}$ (wet bulb).

System F

When the wet and dry subsystems of System A are oriented into System F, the temperature profile describing performance of this configuration is as follows:

Since air flows through the dry tower prior to entering the wet tower, the wet bulb temperature is increased by 5°F; hence, the wet tower ITD is decreased by the same amount. A much larger flow rate of air, which increases the heat rejection capacity of the wet tower, passes through the wet section than in System A. Since a simple relationship between the air flow rate and the heat rejection capacity does not exist, it can be assumed that the wet section will handle the required heat load and is thus equivalent to System A in terms of total heat rejection capacity. However, this air flow rate is not practical for wet towers. If the air flow rate is reduced to that of the wet tower in System A by bypassing all other air from the dry tower, there will be a decrease in overall heat rejection capacity in System F. Using Figure 4, the heat rejection capacity of the wet tower is found to be $Q_W = 0.28 Q_{design}$, and the total heat rejection capacity is $Q_T = 0.95 Q_{design}$. Possible advantages of this system lie in its use as a plume abatement device.

System E

The most straightforward approach to analyzing System E is to examine it qualitatively and compare the results with those obtained in Section 4 and for System F. At best, this configuration can perform only as well as System C1. However, since air is heated by the dry tower, there is a decrease in ITD and a corresponding loss in heat rejection capacity in System E. This system is less effective than System C to the same degree that System F is less effective than System A. Furthermore, air pumping losses are high due to the excessively large air flow rate through the wet tower. There are no apparent advantages in this configuration.

System G

To complete the analysis of System G, a number of additional assumptions must be made. The most important of these is that the air passing through the wet tower increases in enthalpy by increasing in humidity only, with no change in dry bulb temperature. In view of the very high air flow rates and reduced water flow rates through the wet tower (yielding $\dot{m}_w/\dot{m}_a = 0.1$ for the sample system used in Section 4 if one-third of the water is passed through the wet tower), the assumption appears to be reasonable. In actual operation, there may be a small rise in air temperature. A second assumption is that the air side heat transfer coefficients are not affected by nominal changes in humidity.

Given these assumptions, System G should be just as effective as System C in terms of heat rejections. However, it suffers from several handicaps. First, moist air would cause severe corrosion problems in the dry system. Second, air flow rate through the wet tower cannot be controlled since dry tower demands are primary. Finally, there could be some rise in temperature of air as it passes through the wet tower; as a result, there would be a decrease in dry tower ITD.

System H

A relatively simple evaluation of System H is obtained if it is compared to System A in the same manner that System G was compared to System C. Assuming that dry bulb temperature of air passing through the wet tower does

not change and that humidity does not affect the air side heat transfer coefficients, the performance of System H should be equal to that of System A. However, it is subject to the same handicaps as System G: corrosion due to moist air, uneconomic and excess air flowing through the wet tower, and decrease of ITD in the dry tower due to possible rise in dry bulb temperature of air passing through the wet tower. It does not appear to possess any advantages over System A.

5.2 CONCLUSIONS

In evaluation of air flow sequences, the parallel air path configuration has a number of advantages over the series air path design. The primary advantage is due to the fact that nominal air-to-water flow rate ratios for wet and dry tower designs are generally

- $\dot{m}_w/\dot{m}_a = 0.3$ for dry tower,
- $\dot{m}_w/\dot{m}_a = 1.5$ for wet tower,

for conventional cooling systems. If the water flow rate is maintained at the same level for both dry and wet towers used in combination, the air flow rates differ by a factor of 5 for conventionally designed systems. Furthermore, since the wet tower carries as little as one-third the normal design load when used to supplement dry tower heat rejection capacity, the air flow rates could differ by a factor of 15. Hence, most series air flow path configurations are not practical or economical since such systems require unnecessary flow of large quantities of air through wet towers.

A second advantage of parallel air flow systems is due to the fact that the greatest ITD is obtained when ambient air is fed into each dry and wet subsection. Air flowing through a dry or a wet tower increases in temperature and lowers the ITD for any towers following in series. In subsequent analysis, it will also be shown that to minimize water consumption it is advantageous to minimize the air flow rate. In addition, since air flow control is difficult with series systems, parallel path air flow

systems are obviously preferable. Additional problems arise from corrosion and solids deposition as moisture collects on dry heat exchanger surfaces when air flows through wet towers first.

With the exception of plume abatement, there are no apparent advantages to a series air flow configuration in the designs discussed.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

6.0 WATER EVAPORATION RATES

In addition to heat rejection effectiveness, a second important factor in evaluating various cooling system configurations is the amount of water evaporated. In this section, characteristics influencing water evaporation rates are identified and a number of configurations are evaluated for water loss due to evaporation. Water loss due to blowdown and drift losses are ignored since these are functions of design and specifications and are independent of configurations of wet and dry towers.

The configurations selected for analysis in this section are System A, System B, System C, System F, and two new cooling concepts. The two concepts are deluge and adiabatic cooling designs shown in Figure 8.

FIGURE 8. Deluge and Adiabatic Cooling Configurations

When a separate source of water is used for cooling, either deluge cooling of the heat exchanger surfaces or adiabatic cooling of air before it passes through the heat exchangers becomes quite attractive. The advantage is that only a single heat exchanger system is necessary. This system can operate strictly as a dry cooling system when the ambient temperature is below the design temperature. It can also operate as an integral wet/dry system either when water is added to the air or the surfaces are wetted.

All configurations analyzed in this section are designed to reject the same amount of heat under identical ambient conditions. The configurations selected were limited to those expected to be the best designs from the standpoint of thermodynamic effectiveness and water consumption. Performances of other configurations may be inferred from those presented in this section.

6.1 METHODOLOGY

The primary objective of this analysis was to examine Systems A, B, C, and F shown in Figure 1, and the adiabatic and deluge concepts, for rates of water evaporation under conditions resembling actual operation. The design temperature of dry towers was 75°F. The ambient test condition was a dry bulb temperature of 95°F and a wet bulb temperature of 65°F. The stated value of the wet bulb temperature allows the adiabatic air cooling mode to reject the design heat quantity.

One variation in the wet tower conditions was made for the System A configuration. The water flow rates and water-air flow rate ratios for both wet and dry towers were varied to obtain sensitivity of water evaporation rates to air and water flow rates in a wet tower.

To determine the amount of water evaporated by a wet subsection of a cooling system, it is necessary to establish the dynamics of air-water heat and mass transfer within a wet tower. The driving force for rejecting heat in a wet tower is proportional to the enthalpy difference between the bulk of air flow through the tower and the saturated air at water temperature. Thus wet towers can be characterized by two relationships:

- heat balance between air and water

$$\dot{m}_a (h_{aWo} - h_{aWi}) = \dot{m}_w c_{pw} (T_{aWo} - T_{aWi}) \quad (14)$$

- heat transfer rate related to available surface and difference in total heat

$$\frac{KaV}{\dot{m}_w} \int_{T_{aWi}}^{T_{aWo}} \frac{dT}{h_i - h} \quad (15)$$

See Section 3.2 for identification of the various parameters.

The second equation cannot be integrated unless the relationship between the temperature and the change in total heat can be defined. This relationship and other necessary variables will be determined, to the degree of accuracy required, using psychrometric charts in the following analysis.

The basic representation of the cooling mechanism for wet towers is shown in Figure 9. A counterflow cooling tower of constant cross-sectional area through which air at rate $\dot{m}_a$ flows upward and water at rate $\dot{m}_w$ flows downward is considered. The corresponding temperatures are T_{aWi} , T_{aWo} , T_{wWi} , and T_{wWo} for inlet and outlet air and water temperatures, respectively. The temperatures given for air are dry bulb temperatures. An air-water interface with temperature T_i is assumed to exist at saturated conditions. Temperature T_i will be estimated in all subsequent calculations.

The air enthalpy-temperature trajectory may be estimated graphically using the method described by Jakob.⁽³⁾ All calculations for final air temperatures and water evaporation rates which follow are based on that method. It should be noted that an enthalpy-temperature chart is used in Reference 3, while a psychrometric chart is used in this study. The advantage of the latter is availability of humidity and wet bulb temperatures at all times.

FIGURE 9. Counterflow Cooling Tower Mechanism

6.2 ANALYSIS

System A

To establish a baseline, the water evaporation rate is determined for System A for a water-air flow rate ratio of 2.0 for the wet tower. This implies an air flow rate of 1.5×10^8 lb/hr through the wet section since the water flow rate is 3.0×10^8 lb/hr through the condenser. The basic operational parameters, such as water and airflow rates and temperatures, are given in Table 1. The most important of these are:

- ambient temperature, dry bulb   95°F
- ambient temperature, wet bulb   65°F
- design temperature   75°F
- water temperature out   115°F
- water temperature in   135°F

TABLE 1. Cooling Tower Performance Comparisons*

Conditions	System A		System B	System C		System F	Adiabatic Cooling	Deluge Cooling
	High Air Flow Rate	Low Air Flow Rate		Standard Condenser	Oversized Condenser			
T_{aDi} , °F	95.0	95.0	95.0	95.0	95.0	95.0	75.0	95.0
T_{aDo} , °F	114.0	114.0	109.0	111.0	114.0	109.0	94.0	114.0
T_{aWi} , °F (dry bulb)	95.0	95.0	95.0	95.0	95.0	114.0	95.0	95.0
(wet bulb)	65.0	65.0	65.0	65.0	65.0	71.0	65.0	65.0
T_{aWo} , °F (dry bulb)	99.0	104.0	102.0	103.5	101.5	114.5	75.0	106.5
(wet bulb)	82.0	95.0	88.0	94.0	88.0	74.0	65.0	98.0
T_{wWi} , °F	121.7	121.7	135.0	135.0	135.0	121.7	<75.0	135.0
T_{wWo} , °F	115.0	108.3	125.0	103.7	112.3	115.0	<75.0	97.2
$\dot{m}_{aD}$, lb/hr (10^8)	10.0	10.0	10.0	10.0	10.0	10.0	10.0	6.7
$\dot{m}_{aW}$, lb/hr (10^8)	1.5	0.7	1.5	1.0	1.0	10.0		1.0
$\dot{m}_{wD}$, lb/hr (10^8)	3.0	3.0	3.0	2.0	3.0	3.0	3.0	3.0
$\dot{m}_{wW}$, lb/hr (10^8)	3.0	1.5	3.0	1.0	1.0	3.0		1.0
Q_D , Btu/hr (10^8)	45.3	45.3	34.0	36.7	45.3	45.3	68.0	30.2
Q_W , Btu/hr (10^8)	22.7	22.7	34.0	31.3	22.7	22.7		37.8
$Q_D/\dot{m}_{aD}$, Btu/lb (10^8)	4.5	4.5	3.4	3.7	4.5	4.5	6.8	4.5
$Q_W/\dot{m}_{aW}$, Btu/lb (10^8)	15.1	32.4	22.7	31.3	22.7	2.3		37.8
$Q_W/\dot{m}_{wW}$, Btu/lb (10^8)	7.5	15.1	11.3	31.3	22.7	7.6		37.8
$\dot{w}_{evap}$, lb/hr (10^8)	1.9	1.9	2.9	2.6	1.9	2.3	4.6	3.2

*Basic operational parameters - 75°F design temperature with ambient temperatures of 95°F dry bulb, 65°F wet bulb.

With this information, the heat loads on dry towers can be determined, as well as the amount of heat rejection capacity which must be supplemented by alternate means. Assuming a linear relationship between heat rejected and ITD for dry towers, the heat rejection capacity of dry towers at test condition is $\frac{2}{3} Q_{\text{design}}$. Thus, an amount of heat rejection equal to $\frac{1}{3} Q_{\text{design}}$ must be obtained from a supplementary system which, for this study, is a wet tower.

The rate of water evaporated can be determined once the temperature-enthalpy path of air through a wet cooling tower is determined. Assuming counterflow between air and water as shown in Figure 9, the air path in a wet tower can be described for System A. In Figure 10, two different air flow rates are shown with identical water-air flow rate ratios. Two different air flow rates were used to establish sensitivity of water evaporation rate to air flow rate through a wet tower. Sensitivity of water evaporation rate to water flow rate is established in the process of analyzing the other configurations.

For the dry and wet tower parameters shown in Table 1, the water evaporation rates are:

$$\dot{w}_{\text{ev}} (\dot{m}_{\text{aW}} = 1.5 \times 10^8 \text{ lb/hr}) = 1.9 \times 10^6 \text{ lb/hr} \quad (3800 \text{ gpm})$$

$$\dot{w}_{\text{ev}} (\dot{m}_{\text{aW}} = 0.7 \times 10^8 \text{ lb/hr}) = 1.9 \times 10^6 \text{ lb/hr} \quad (3800 \text{ gpm})$$

The results indicate little or no change in evaporation rates due to change in air flow rate for the range of air flow rates shown. From the model shown in Figure 9 and the temperature-enthalpy trajectory in Figure 10, it may be deduced that air flow rate affects evaporation rate when the air flow rate is very high. The dominant factors appear to be air and water temperature. This is because the initial slope of the air temperature-enthalpy trajectory depends on exit water temperature and inlet air temperature, while the final slope depends on the inlet water temperature and exit air temperature. A high exit water temperature attained with high water flow rates provides highest exit air temperatures and minimum water evaporation rates. Low water flow rates and high air flow rates may result in cooling a portion of air and evaporation of a greater amount of water.

FIGURE 10. Temperature-Enthalpy Trajectory for Air in System A Wet Cooling Tower

System B

In the configuration shown in Figure 1B, the wet and dry towers are reversed with respect to the water flow sequence. From results in Section 4, the heat rejection capacity of the dry tower is reduced from $0.67 Q_{\text{design}}$ to $0.50 Q_{\text{design}}$ at the test conditions. Thus, the wet tower must supply $0.50 Q_{\text{design}}$ at these same conditions. This means a slightly larger wet tower than the one specified in Section 4. With the critical variables as shown in Table 1, the water evaporation rate is

$$\dot{w}_{\text{ev}} = 2.9 \times 10^6 \text{ lb/hr}$$

or about 5800 gpm. The greater rate of water evaporation relative to System A is due to the rejection of more heat by the wet tower in System B. The evaporation rate could be decreased if the flow rate of water is increased to twice or more the given value of $3.0 \times 10^8 \text{ lb/hr}$. However, the gain is small. If the water flow rate is doubled, the decrease in evaporation rate is less than 7 percent.

The undesirability of System B is clearly illustrated by the results in Section 4 and this section. Its performance is decreased due to less effective utilization of dry and wet tower ITDs; further, it promotes greater water consumption than its dual configuration given by System A.

The air temperature-enthalpy path for System B is shown in Figure 11.

System C

This configuration presents an interesting variation due to the added dimension of flexibility in flow rates. A conventional approach is with the total flow rate through the condenser held constant and divided between the wet and dry subsections. A second approach, described in Section 4, is to maintain the water flow rate through the dry tower and to use an oversized condenser and separate pump for the wet tower. The rates of water evaporation for the two cases as shown in Table 1 are:

$$\dot{w}_{\text{ev}} (\text{standard condenser}) = 2.6 \times 10^6 \text{ lb/hr}$$

$$\dot{w}_{\text{ev}} (\text{oversized condenser}) = 1.9 \times 10^6 \text{ lb/hr.}$$

FIGURE 11. Temperature-Enthalpy Trajectory for Air in System B Wet Cooling Tower

The difference in evaporation rates occurs because the system with an oversized condenser maintains the dry tower at full heat rejection capacity of $0.67 Q_{\text{design}}$ while the standard condenser, which decreases the flow rate by 33 percent, decreases the heat rejection capacity to $0.54 Q_{\text{design}}$. The rate of water evaporation is directly proportional to the heat rejection capacity of the wet subsection. The air temperature-enthalpy path for air in the wet tower is described approximately by Figure 10.

System D and System E will not be analyzed. System D is a simple extension of System A but designed for more desirable rates of water flow through the wet subsection. Water evaporation rates are identical to those of System A. System E combines the less desirable characteristics of System B and System F, which will be discussed next. It has very high evaporation rates due to inefficient use of the dry subsection and because of the warmer air that enters the wet subsection.

System F

In the analysis of System F, two factors stand out. First, a very large volume of air, much more than necessary, must be passed through the wet subsection. This results in large air flow losses. Second, the temperature of air entering the wet tower is increased. This leads to a greater water evaporation rate and the need for larger towers. Assuming the values of variables stated in Table 1, the rate of water evaporation is

$$\dot{w}_{\text{ev}} = 2.3 \times 10^6 \text{ lb/hr.}$$

The advantages of System F are that it fully utilizes the dry subsection and provides a single structure for both dry and wet subsections. An approximate air temperature-enthalpy path is shown in Figure 12.

Adiabatic Cooling

In adiabatic cooling, inlet air temperature is reduced to its wet bulb temperature when saturated by evaporation of added water. The temperature that may be obtained this way in practice depends on the droplet size and is generally higher than the wet bulb temperature by several degrees. Adiabatic cooling is very useful in dry climates where the wet bulb temperature is substantially lower than the dry bulb temperature.

FIGURE 12. Temperature-Enthalpy Trajectory for Air in System F Wet Cooling Tower

The basic mechanism of adiabatic cooling involves the removal of latent heat by evaporation from a droplet with a surface temperature of T_s^* (the local wet bulb temperature) and which is surrounded by air at temperature T . The heat removed by evaporation must equal heat flowing into a droplet. Thus

$$-4\pi r^2 L_{p_w} \frac{dr}{dt}$$

is the latent heat removed per unit time. Heat flow into the spherical surface of radius z is

$$4\pi z^2 k \frac{dT_z}{dz} \quad (16)$$

where k is thermal conductivity of air. Assuming a temperature distribution, from droplet surface to some distance z , from droplet center, given by

$$T_z = T - \frac{r}{z} (T - T_s^*) \quad (17)$$

heat balance yields

$$-r \frac{dr}{dt} = \frac{k}{L_{p_w}} (T - T_s^*) \quad (18)$$

Integration yields

$$r^2(t) = r_0^2 - \frac{2k(T - T_s^*)}{L_{p_w}} t \quad (19)$$

assuming that $T - T_s^*$ is constant. Here, $r(t)$ denotes droplet radius at time t and r_0 is the initial droplet radius. Thus, a temperature gradient must be maintained in order to evaporate the water.

Assuming water from a separate source as shown in Figure 8, the amount of water evaporated can be calculated from the initial air dry and wet bulb temperatures, the final dry bulb temperature, and latent heat of vaporization for water. Thus, for initial temperatures of 95°F db, 65°F wb and a desired temperature of 75°F db, the amount of water evaporated is

$$\dot{m}_a c_{pa} \Delta T = L \dot{w}_{ev}$$

Using values given in Table 1, and $L = 1050$ Btu/lb,

$$\dot{w}_{ev} = \frac{10 \times 10^8 \times 0.24 \times (95-75)}{1050} = 4.6 \times 10^6 \text{ lb/hr.}$$

The process is plotted on a psychrometric chart in Figure 13, which can be used to arrive at the same result as in the preceding equation. The results indicate that adiabatic cooling consumes water at a higher rate than any of the other systems. This is because the process of evaporation is not used in cooling the primary fluid but is only a mechanism for adiabatic cooling of the air.

As a final comment on adiabatic cooling, Wachtell⁽⁴⁾ found that ideal atomization of water droplets to promote humidification is an energy-consuming process. The alternative, coarse water droplets, leads to solids deposition and corrosion of the dry heat exchanger surfaces.

Deluge Cooling

Momentarily ignoring the problems associated with solids deposition and corrosion, a very promising and effective system for supplementing dry cooling towers is the deluge system or a wet-surface air-cooler. When envisioned as a supplementary heat rejection system, the deluge system uses conventional dry surfaces for cooling at or under design temperatures. At high ambient temperatures, a water system is provided for covering the air side surfaces with an unbroken film of water. The running water is collected in a reservoir and then used again to wet the heat exchanger surfaces. The transfer of heat is increased since the primary mechanism for heat transfer is evaporation of water. In fact, the deluge system, with the heat exchangers acting as the packing, can be treated as a wet tower.

Using principles of wet cooling towers, it is possible to estimate the rate of water evaporation for a deluge cooled system. Since air is in contact only with the water film, the cooling process is essentially evaporative. Thus, it is necessary only to estimate what fraction of the cooling process is through deluge cooling.

FIGURE 13. Temperature-Enthalpy Trajectory for Air in an Adiabatic Cooling System

Assuming that heat transfer rate is enhanced by a factor of 2.5 for a given volume of dry heat exchangers,⁽⁵⁾ the fraction of the total dry system which must be subjected to a film of water can be determined when the ITD is decreased by one-third. Thus, if V is the total volume used in dry heat transfer, the portion which must be deluged is $1/3V$. With this portion, $(4/9)Q_{\text{design}}$ is rejected by the dry section and $(5/9)Q_{\text{design}}$ is rejected by the wet section. The amount of water evaporated can then be determined for the conditions given in Table 1 and is

$$\dot{w}_{\text{ev}} = 3.2 \times 10^6 \text{ lb/hr.}$$

It should be noted that this rate of evaporation is primarily a function of the total heat rejected and is relatively independent of the other parameters. The air flow rate through the deluged section must be reduced to an $\dot{m}_w/\dot{m}_a$ ratio more closely related to that in wet towers. This is reflected in the values given in Table 1. The deluges and dry sections are arranged in a parallel water flow scheme with standard condenser. The deluge system shows a higher rate of water evaporation because of the method of deploying the cooling subsystems. In the deluge system, unlike the others, all heat exchange surfaces are active at or below design conditions. When the design temperatures are exceeded, a portion of the dry surfaces is removed from dry duty and subjected to water flooding. Thus, under test conditions, only two-thirds of the overall system serves as the dry subsystem as compared to unity for all other configurations. Therefore, the wet section must carry a proportionally higher heat load. Nonetheless, the deluge concept offers tremendous advantages over conventional wet towers if problems of corrosion and solids deposition can be solved.

6.3 CONCLUSIONS

The results of this section, previously summarized in Table 1, lead to the following conclusions:

1. In wet cooling towers, loss of water is primarily a function of the amount of heat rejected. Additional factors are temperatures of exit fluid and incoming air.

2. Gains in overall heat transfer rates when surfaces of dry heat exchangers are wetted indicate that wet surface cooling towers have promise as wet/dry cooling towers. The main advantages are that there are no sections inactive by design and the size of dry towers may be reduced drastically.
3. System A and System C2 are most conservative in rates of water consumption.
4. Adiabatic cooling of air has the highest rate of water evaporation.

7.0 SEPARATE AND HYBRID WET/DRY SYSTEMS

7.1 COMPARISON OF SEPARATE AND HYBRID SYSTEMS

The foregoing analysis points clearly to parallel air path cooling towers as the preferred systems for electric power plants. This adds another dimension to the selection of preferred wet/dry cooling systems where the wet system is to be used only for peaking.

The added factor in selecting a preferred parallel air path wet/dry configuration is the choice between wet/dry cooling systems consisting of separate wet and dry towers and wet/dry cooling systems combining wet and dry sections into one module. Both the separate and hybrid combinations have specific advantages and disadvantages for use in electric power plants.

Certain terms used in this section require clarification. Among these are:

- combined - any wet/dry system in which wet and dry cooling are possible simultaneously.
- hybrid - a wet/dry cooling system in which wet and dry units are combined into a single unit.
- integral - cooling towers which utilize surfaces that operate as dry units during normal temperatures and are deluged or sprayed with water when the ambient temperatures rise above design conditions.
- separate - a cooling system utilizing separate wet and separate dry towers.
- closed - the turbine-condenser cooling water flows in a closed loop not exposed to the atmosphere.
- open - the turbine-condenser cooling water is exposed to the atmosphere.

It is important to note that the two basic flow diagrams shown in Figures 1A and 1C do not differentiate between separate and hybrid systems. From the viewpoint of heat and mass transfer, thermodynamic effectiveness,

and water consumption, there is no difference between the two cooling systems. The major difference between the two types of configuration is structural, which leads to a number of economic, maintenance, operation, and functional differences. The advantages and disadvantages of each type of cooling tower configuration are listed in Table 2. The list resulted from discussions with vendors and general analysis.

A number of factors influence the advantages of one system over the other. For large power plants, the separate tower design appears to have advantages over hybrid tower design when the design temperature for dry towers is high. Under these conditions vapor plume formation is not very probable. Furthermore, a large number of cells would be used. One study suggests an optional dry cooling tower system with 141 cells or wet cooling tower with 33 cells for a 1000 MWe LWR power plant.⁽⁶⁾ It would appear that incorporating 16 wet cells side-by-side with about 100 cells is more practical and, possibly, more economical than using more than 100 hybrid wet/dry cells. Another possible approach could be to use 125 all dry cells and 32 wet/dry hybrid cells, each of which is half wet and half dry. In very humid areas, visible vapor plumes arising from wet towers could cause problems. The hybrid dry/wet system would be the preferred design under these conditions.

One condition under which hybrid towers have an advantage is when cooling duty is low so that only a few wet and dry cells are required. Hybrid wet/dry towers would provide flexibility and a more economical alternative than separate wet and dry towers.

Although only hybrid wet/dry towers can be used for total plume abatement, wet towers used in separate wet and dry combinations can be designed for minimal vapor plume formation. Unless relative humidity is near 100 percent, the wet tower air flow rate can be made high enough so that the line on a psychrometric chart drawn between ambient air temperature and tower exit air temperature, which describes air state as it leaves

TABLE 2. Cooling Tower Configuration Comparisons

	<u>Hybrid Wet/Dry</u>	<u>Separate Wet/Dry</u>
Advantages	1. Single integrated structure. 2. Plume abatement. 3. Simplified hydraulics for series water flow.	1. Greater flexibility in design operation, and expansion in facilities. 2. Practical for forced dry systems. 3. Simplified winter operation since wet and dry sections are separate. 4. Established and simpler design. 5. Simplified hydraulics for water flow with separate cooling water for wet and dry towers.
Disadvantages	1. Moist air may cause fouling and corrosion of dry surfaces. 2. More complex structures due to integration of wet and dry sections. 3. Structurally biased to a series water flow system.	1. Not practical for plume abatement.

the tower, does not intersect the saturation line. With this criterion satisfied, the likelihood of a plume forming above a wet tower is small. Estimated increase in air flow rate requirements is approximately five-fold but depends upon ambient conditions. Thus, the design ratio $\dot{m}_w/\dot{m}_a$ may be 0.3 for the wet tower instead of 1.5 currently used in practice.

7.2 CLOSED AND OPEN LOOP WATER SYSTEMS

An additional factor in design of wet/dry cooling towers is whether water flowing in the dry tower also flows through the wet tower (open system) or is completely independent of the wet tower (closed system). Four examples of closed systems are shown in Figure 14 and four examples of open systems are shown in Figure 15. Naturally, there are other combinations possible which could prove to be quite attractive. However, it is more important to realize the differences between open and closed systems and to quantify these differences.

Basically, the practical difference between open and closed systems lies in the contamination of cooling surfaces. Metal surfaces associated with wet cooling systems (open loop) are subject to fouling and scaling much more than closed loop water flow systems normally associated with pure dry cooling systems. Thus, there are possibilities for capital cost savings in design and construction costs of the latter, because less expensive material can be used for heat exchanger surfaces.

The results in Sections 4 and 5 suggest an optimal flow configuration for the open and closed loop designs. All configurations have parallel air flow. For the closed systems shown in Figure 14, the heat transfer surface of the water-to-water heat exchangers in the series flow configuration is about four times that of the tubes of a surface condenser used in the parallel flow configuration. Thus, for closed loop systems, the parallel flow configuration appears to be economically superior. This holds for both separate and hybrid systems.

SEPARATE -- SERIES FLOW

SEPARATE -- PARALLEL FLOW

HYBRID - SERIES FLOW

HYBRID -- PARALLEL FLOW

FIGURE 14. Closed Water Loop Systems

FIGURE 15. Open Water Loop Systems

Examining open loop systems in Figure 15, the parallel flow configurations are less effective thermally since any flow into wet towers must be subtracted from dry towers. Thus the effectiveness of dry towers is reduced and larger wet towers are required. If water flows sequentially from dry to wet sections, the effectiveness of dry towers is maintained, although larger wet towers are required because of reduced ITD. There are advantages to the open loop systems in that the condenser size is minimal.

Comparison of closed and open loop systems is made in Table 3 for a number of considerations. The primary factor appears to be the cost of material used in condensers and heat exchanger tubes. The feasibility of using conventional metals in dry sections of closed loop systems makes these definitely more attractive than other combinations.

TABLE 3. Comparison of Closed and Open Loop Systems

Type of Loop		
Parameter	Closed	Open
Best Option	Parallel Flow	Series Flow
Combination	Hybrid or Separate	Hybrid or Separate
Condenser		
Size	Larger (separate water box)	Standard Size and Fouling Factor
Material	Carbon Steel	Stainless Steel or Titanium
Heat Exchanger		
Size	Standard Size	Standard Size and Fouling Factor
Material	Carbon Steel	Stainless Steel or Titanium
Wet Tower	Standard Size	Larger (smaller ITD)
Down Time	Better	Normal

7.3 WET/DRY TOWER COMBINATIONS

In the final selection of a combined wet/dry cooling system, additional design possibilities arise. Up to this point, only pure hybrid towers and pure separate towers have been discussed. However, there are other possible configurations.

To simplify the discussion, consider only mechanical draft towers and the three modules shown in Figure 16. These modules are assumed to be approximately equal in size, but could vary by a factor of 2 or more. The dry module and wet module used in Reference 6 are approximately equal in dimensions. The number of modules is used to indicate the cooling tower size for each cooling tower configuration.

The most practical combinations of wet, dry, and hybrid modules for a large power plant requiring approximately 120 dry cells or 30 wet cells are the following:

1. Pure Hybrid--120 wet/dry cells and appropriate manifolding distributed in six tower complexes.
2. Mixed Hybrid--80 dry cells and 40 wet/dry cells distributed in some manner in six towers. The modules could be mixed within each tower or separated to yield two hybrid towers of 20 units each and four dry towers of 20 units each. This configuration provides somewhat better flexibility in design of hybrid modules since the number of hybrids could be determined by size of wet section which could be optimized for a given heat load.
3. Side-by-Side--100 dry cells and 20 wet cells distributed equally through six towers. This configuration has potential for plume abatement if the modules are arranged properly.
4. Separate--100 dry cells and 20 wet cells distributed into five dry towers and one wet tower located at some distance from the dry units. This configuration minimizes problems with recirculation and control of air flowing through wet modules.

DRY MODULE

WET MODULE

HYBRID MODULE

FIGURE 16. Cooling Modules

The four configurations are shown schematically in Figure 17 for circular and rectilinear tower configurations. A comparison of these configurations in terms of various parameters is shown in Table 4. Examination of the table suggests no inherent advantages for hybrid systems and possibly some maintenance, operation, and capital cost disadvantages.

There is a strong motivation to prefer separate wet modules for parallel water flow systems. This is due to the fact that wet tower volume and the cost are primarily functions of total heat rejected by the unit and are less dependent on water flow rates. The plan area and, consequently, the packing height is determined by the amount of water flowing through the tower. A high flow rate of water means a large plan area and a small height, excellent for hybrids. However, this also means a much larger condenser for the wet tower and the attendant higher costs. Thus, economic design favors lowest possible flow rates and highest towers with smallest plan areas. This combination yields the lowest cost condenser.

In summary, the separate and hybrid wet/dry cooling towers have advantages over each other under specific conditions. There are no thermodynamic differences between the two schemes. Unless very clever designs are created, the hybrid structure will probably have a more complex and more demanding structure. If plume abatement is a problem, a hybrid wet/dry system must be used. Power plant site and corresponding climate will play a strong part in the choice between the two wet/dry combinations for augmented cooling of a power plant.

PURE HYBRID TOWERS

MIXED HYBRID TOWERS

SIDE-BY-SIDE WET AND DRY CELLS

SEPARATE TOWERS

FIGURE 17. Cooling Tower Configurations

TABLE 4. Hybrid and Separate Tower Comparisons

Option Parameter	Closed Hybrid		Closed Separate	
	Pure Hybrid 120w/d cells	Mixed Hybrid 20w/d, 100d cells	Side-by-Side 20 wet, 100 dry	Separate Tower 20 wet, 100 dry
Piping	Less	Less	Less	Normal
Recirculation	Poor	Some	Minimal Plus	Minimal
Manifolds	Higher	Standard Plus	Standard	Standard
Fill and Drain	Higher(?)	Normal	Normal	Normal
Structures	Higher	Higher	Normal	Normal
Fans	Less	Less	Less-Normal	Normal
Doors and Louvers	Worst or None*	Bad or None*	Small Problem	No Problem
Control Instruments	Worst	Some Problem	Normal	Normal
Maintenance	Highest	Highest	Slightly Higher	Normal

* Marley design for San Juan Unit 3, Public Service of New Mexico, uses no doors or louvers.

8.0 MAJOR RESULTS AND CONCLUSIONS

The results of this preliminary linear analysis of wet/dry cooling system combinations indicate a number of significant conclusions. It should be pointed out that a high degree of approximation was used in some cases. However, the physics of heat and mass balances were preserved and the results are qualitatively valid. The major results and conclusions are as follows:

1. Parallel air path systems are superior to series air path systems. There is a large difference in air flow rates for wet and dry towers. Thus, an excess of air must be pumped through the wet sections when wet and dry sections are used in series. Furthermore, flowing the air through dry, then wet, subsystems, or conversely, leads to a decrease in ITD and heat rejection capacities for the second subsystem. There are other difficulties but these two reasons alone handicap series air dry/wet or wet/dry combinations when a wet system is to be used for augmenting dry cooling tower heat rejection capacity.
2. A series water flow pattern through dry and wet subsystems allows use of a standard-sized condenser but requires a larger wet tower than does a parallel water flow pattern. The flow sequence should be dry subsection before wet as shown in Figure 1A. Two reasons for this sequence are:
  - the effectiveness of dry-before-wet is somewhat higher due to the relationship between ITDs of wet and dry towers and wet and dry bulb temperatures. This may not be true for very humid climates; and
  - it is desirable to maintain the dry section operating at full capacity to minimize water evaporation losses of the wet tower.

3. A parallel water flow configuration is thermodynamically optimal when an oversized condenser is used such that any flow necessary for the wet subsection is obtained from that section of the condenser used only for supplementary cooling. This provides optimal utilization of the wet and dry cooling subsections at the cost of a larger condenser. The choice between the parallel and series water flow configurations is dependent primarily upon economics of cooling system construction.
4. In wet cooling systems, the loss of water is primarily a function of the amount of heat rejected. A lesser factor is the relative temperature of exit fluid and incoming air. Flow rate of water has some effect to the extent that low rates mean low exit temperatures, and, therefore, possible cooling of the air which leads to greater water consumption. Air flow rates have only a small effect on the rate of water evaporation. Very large air flow rates could increase the rate of water evaporation.
5. Gains in overall heat transfer rates due to wetting the surfaces of dry heat exchangers indicate that wet surface coolers have promise as wet/dry systems. Water evaporation rates are comparable to conventional wet towers.
6. There are no thermodynamic or water consumption rate differences between separate wet/dry towers and hybrid wet/dry towers. Vapor plume problems and economics of power plant construction for a particular site will determine which combination will be used. Separate towers are structurally simpler and avoid louver material and noise problems. Furthermore, integrated towers could present surface corrosion problems.

A number of efforts are necessary as a follow-on to this study. First, the economics of hybrid wet/dry cells should be compared to separate wet

and dry cells. Although it does not seem probable that hybrid structure could be less expensive, creative design could yield interesting answers. The second important effort should be in the area of wet surface heat exchangers. Currently there are very little data describing improvement in overall heat transfer coefficients and corrosion with deluge and spray cooling systems. These must be established before such systems can be considered or rejected for power plant cooling systems.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

REFERENCES

1. T. E. Croley, V. C. Patel, and M. S. Cheng, "The Water and Total Optimization of Wet and Dry-Wet Cooling Towers for Electric Power Plants," Iowa Institute of Hydraulic Research, The University of Iowa, Iowa City, Iowa.
2. K. W. Li, "Analytical Studies of Dry/Wet Cooling Systems for Power Plants," Symposium on Dry and Wet/Dry Cooling Towers for Power Plants, ASME, November 11-15, 1973.
3. M. Jakob, Heat Transfer, John Wiley and Sons, New York, 1957.
4. G. P. Wachtell, Atomized Water Injection to Improve Dry Cooling Tower Performance, Report C00-2241-1, UC-12, Franklin Institute Research Laboratories, Philadelphia, PA; 1974.
5. V. M. Altman, et al., "Proektirovanie Sistem Okladjdeniya s Sykhimi-Mokrimi Gradirnami dlia Atomnikh i Teplovikh Elektrostantsii," Leningrad Division of Institute "Teploelektro-project," Moscow, 1975.
6. United Engineers and Constructors, Inc., Heat Sink Design and Cost Study for Fossil and Nuclear Power Plants, Report-WASH-1360, United States Atomic Energy Commission, December 1974.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

BIBLIOGRAPHY

1. Baker, D. R., and H. A. Shryock, "A Comprehensive Approach to the Analysis of Cooling Tower Performance," Trans. ASME, Journal of Heat Transfer, 83, C3, pp. 339-350, August 1961.
2. Heller, L., "Wet/Dry Hybrid Condensing System," Proceedings Dry and Wet/Dry Cooling Towers for Power Plants, ASME Winter Annual Meeting, November 1973.
3. Kals, W., "Wet Surface Aircoolers: Characteristics and Usefulness," ASME Paper 72-HT-28, 1972.
4. Kennedy, J. F., "Wet Cooling Towers," Report, Institute of Hydraulic Research, University of Iowa, Iowa City, Iowa.
5. Leung, P., and R. E. Moore, "Water Consumption Determination for Steam Power Plant Cooling Towers," ASME Paper No. 69-WA-PWR-3, Los Angeles, 1969.
6. Li, K. W., "Research Work on Dry-Type Cooling Towers for Power Plants," Report No. 23, Engineering Equipment Station Series, North Dakota State University, April 1974.
7. McKelvey, K. K., and M. Brooke, The Industrial Cooling Tower, Elsevier Publishing Company, New York, 1959.
8. Smith, K., "The Performance and Economics of a Series Hybrid Cooling Tower," Paper No. PB-227 813, Cornell University, May 1973.

APPENDIX A

EFFECT OF DRY AND WET TOWER ITDs
ON COMBINED TOWER PERFORMANCE

APPENDIX A

EFFECT OF DRY AND WET TOWER ITDs ON COMBINED TOWER PERFORMANCE

The effect of the way in which dry and wet tower ITDs are defined on combined dry/wet tower performance is illustrated here by direct comparison of System A with System B.

For convenience, assume the following for System A:

$$Q_D = Q_W = 1/2 Q_T$$

Furthermore, for both systems, it is assumed that heat capacity flows are the same for both fluids in both towers:

$$\dot{m}_{aW} c_{pa} = \dot{m}_{aD} c_{pa} = \dot{m}_{wD} c_{pw} = \dot{m}_{wW} c_{pw} = 1.$$

The major assumption now is that

$$Q = (\text{constant}) (\text{ITD})$$

for both wet and dry systems. While this is a restricting assumption, it does not affect the nature of the result. Defining μ as the ratio of ITDs for System A and System B,

$$\mu = \frac{(\text{ITD})_B}{(\text{ITD})_A}$$

then

$$(Q_W)_B = \mu (Q_W)_A$$

Total heat rejected is given by

$$(Q_T = Q_D + Q_W)$$

For System A, we have

$$(Q_T)_A = (Q_D)_A + (Q_W)_A$$

and

$$(Q_D)_A = (Q_W)_A = 1/2(Q_T)_A$$

System B can be expressed as

$$(Q_T)_B = (Q_D)_B + (Q_W)_B$$

Due to increase in ITD for the wet tower,

$$(Q_W)_B = \mu_W (Q_W)_A$$

Due to decrease in ITD for the dry tower,

$$(Q_D)_B = \mu_D (Q_D)_A$$

Hence, System B may be expressed in terms of System A as

$$(Q_T)_B = \mu_D (Q_D)_A + \mu_W (Q_W)_A = 1/2(Q_T)_A (\mu_D + \mu_W)$$

To determine the relative magnitudes of $(Q_T)_B$ and $(Q_T)_A$, we must express μ_D and μ_W in terms of dry bulb and wet bulb temperatures;

$$\mu_D = \frac{(T_{wDi} - T_{ai})_B}{(T_{wDi} - T_{ai})_A}, \quad \mu_W = \frac{(T_{wWi} - T_{ai}^*)_B}{(T_{wWi} - T_{ai}^*)_A}$$

where the (*) is used to indicate wet bulb temperatures. From Figure 1,

$$T_{wDi} = T_{wi} \text{ for System A,}$$

$$T_{wWi} = T_{wi} \text{ for System B.}$$

Also, $T_{ai}^* \leq T_{ai}$, i.e., the wet bulb temperature is less than or equal to the dry bulb air temperature. Thus

$$\mu_D = \frac{(T_{wDi})_B - T_{ai}}{T_{wi} - T_{ai}}, \quad \mu_W = \frac{T_{wi} - T_{ai}^*}{(T_{wWi})_A - T_{ai}^*} \quad (1)$$

and the only unknowns are the water temperatures into the second components of each System A and B. From water side heat balances on the dry and wet towers, letting $\dot{m}_a c_{pa} = \dot{m}_w c_{pw} = 1$,

$$\begin{aligned} (T_{wDi})_B &= (T_{wWo})_B = (T_{wWi})_B - (Q_W)_B = T_{wi} - (Q_W)_B \\ &= T_{wi} - \mu_W (Q_W)_A \end{aligned}$$

$$\begin{aligned} (T_{wWi})_A &= (T_{wDo})_A = (T_{wDi})_A - (Q_D)_A \\ &= T_{wi} - (Q_D)_A \end{aligned}$$

Substituting into equation (1),

$$\mu_D = \frac{T_{wi} - \mu_W (Q_W)_A - T_{ai}}{T_{wi} - T_{ai}} = 1 - \frac{1/2 \mu_W (Q_T)_A}{T_{wi} - T_{ai}}$$

$$\mu_W = \frac{T_{wi} - T_{ai}^*}{T_{wi} - (Q_D)_A - T_{ai}^*}$$

$$\mu_D = 1 - \left(\frac{1/2 (Q_T)_A}{T_{wi} - T_{ai}} \right) \cdot \left(\frac{T_{wi} - T_{ai}^*}{T_{wi} - T_{ai}^* - 1/2 (Q_T)_A} \right)$$

$$\mu_D + \mu_W =$$

$$1 - \left(\frac{1/2 (Q_T)_A}{T_{wi} - T_{ai}} \right) \cdot \left(\frac{T_{wi} - T_{ai}^*}{T_{wi} - T_{ai}^* - 1/2 (Q_T)_A} \right) + \frac{T_{wi} - T_{ai}^*}{T_{wi} - T_{ai}^* - 1/2 (Q_T)_A}$$

$$\begin{aligned}
\mu_D + \mu_W &= 1 + \left(\frac{T_{wi} - T_{ai}^*}{T_{wi} - T_{ai}^* - 1/2(Q_T)_A} \right) \left(1 - \frac{1/2(Q_T)_A}{T_{wi} - T_{ai}} \right) \\
\mu_D + \mu_W &= 1 + \left(\frac{T_{wi} - T_{ai}^*}{T_{wi} - T_{ai}^* - 1/2(Q_T)_A} \right) \left(\frac{T_{wi} - T_{ai} - 1/2(Q_T)_A}{T_{wi} - T_{ai}} \right) \\
&= 1 + \phi
\end{aligned} \tag{2}$$

where ϕ represents the bracketed quantities above.

Now, for $T_{ai}^* = T_{ai}$,

$$\mu_D + \mu_W = 1 + 1 = 2$$

$$\text{and } (Q_T)_B = 1/2(Q_T)_A \quad (2) = (Q_T)_A.$$

For $T_{ai}^* \leq T_{ai}$,

$$x^* = T_{wi} - T_{ai}^* \geq x = T_{wi} - T_{ai}.$$

Define

$$\phi = \left(\frac{x^*}{x^* - y} \right) \left(\frac{x - y}{x} \right),$$

where

$$y = 1/2(Q_T)_A,$$

$$\phi = \left(\frac{x^*}{x} \right) \left(\frac{x - y}{x^* - y} \right) = \frac{xx^* - x^*y}{xx^* - xy}.$$

Since

$$x^* \geq x.$$

$$\phi \leq 1.$$

Therefore

$$\mu_D + \mu_W = 1 + \phi \leq 2$$

$$\text{and } (Q_T)_B = 1/2 (Q_T)_A (1 + \phi) \leq (Q_T)_A. \quad (3)$$

Thus, the more effective combination for systems with heat rejection capacities proportional to ITD is to flow the water first through a dry system and then through a wet system as shown in Figure 1A.

DISTRIBUTION

No. of
Copies

No. of
Copies

OFFSITE

A. A. Churm Chicago Patent Group Energy Research and Development Administration 9800 South Cass Avenue Argonne, IL 60439	J. S. Joyce Allis-Chalmers Power Systems, Inc. 1135 South 70th Street West Allis, WI 53214
Office of Assistant General Counsel for Patents Energy Research and Development Administration Washington, DC 20545	E. T. Wanderer Aluminum Company of America Alcoa Technical Center Alcoa Center, OH 45069
27 ERDA Technical Information Center	H. J. Janzon American Electric Power 2 Broadway New York, NY 10004
4 W. F. Savage Advanced Concepts Evaluation Branch - F-309 ERDA Division of Nuclear Research and Application Washington, DC 20545	T. Woods Arizona Public Service Co. 2121 W. Cheryl Drive Phoenix, AZ 85021
T. Beresovski Chief, Advanced Concepts Evaluation Branch ERDA Division of Nuclear Research and Application Washington, DC 20545	M. W. Peterson Babcock & Wilcox Fossil Power Division 20 South Van Buren Barberton, OH 44203
D. C. Bauer Director, Division of Nuclear Research and Application Energy Research and Development Administration Washington, DC 20545	G. C. Creel Baltimore Gas & Electric Co. Gas & Electric Building Baltimore, MD 21203
B. R. Dickey Allied Chemical Company 550 2nd Street Idaho Falls, ID 83401	E. Schinner Baltimore Aircoil Co., Inc. P.O. Box 7322 Baltimore, MD 21227
	J. P. Budliger Battelle-Geneva 7 Route De Drizi 1227 Geneva, SWITZERLAND

No. of
Copies

P. Leung
Bechtel Corporation
P.O. Box 60860
Terminal Annex
Los Angeles, CA 90060

G. R. Retti
Bechtel Corporation
P.O. Box 3965
San Francisco, CA 94119

J. P. Rossie
R. W. Beck and Associates
400 Prudential Plaza
Denver, CO 80202

B. Westre
Black Hills Power and Light
Company
P.O. Box 1400
Rapid City, SD 57701

J. Sell
Carolina Power & Light Co.
336 Fayetteville Street
Raleigh, NC 27602

J. Morse
Catalytic Construction Corp.
P.O. Box 11402
Charlotte, NC 28029

H. H. Osborn
Combustion Engineering
Air Preheater Company
Wellsville, NY 14895

R. H. Holyoak
Commonwealth Edison
One First Plaza
P.O. Box 767
Chicago, IL 60690

C. L. Newman
Consolidated Edison Co.
of New York, Inc.
4 Irving Place
New York, NY 10003

No. of
Copies

W. A. Messner
Consolidated Edison Co.
of New York, Inc.
4 Irving Place
New York, NY 10003

L. E. Wilkinson
Consultant
6582 Ganon Street, S. E.
Salem, OR 97301

P. Sporn
Consultant Engineer
74 Trinity Pl., Suite 511
New York, NY 10006

F. K. Moore
Cornell University
Ithaca, NY 14850

R. J. Haberski
Curtiss-Wright Corp.
One Passaic Street
Wood Ridge, NJ 07075

S. K. Blackley
Duke Power Company
P.O. Box 2178
Charlotte, NC 28201

J. K. Swindt
Ecodyne Cooling Products Co.
San Rosa, CA 95403

J. Maulbetsch
Electric Power Research
Institute
3412 Hillview Avenue
P.O. Box 10412
Palo Alto, CA 94304

L. Geller
Empire State Electric Energy
Research Corp.
1250 Broadway
New York, NY 10001

No. of
Copies

F. H. Rainwater
Environmental Protection Agency
Pacific Northwest Water
Laboratory
200 S.W. 35th Street
Corvallis, OR 97330

W. Hoydisch
Environmental Sciences and
Services
44-43 37th Street
Long Island City, NY 11101

A. M. Rubin
Franklin Institute
Twentieth & Parkway
Philadelphia, PA 19103

B. Davis
GEA Airexchangers, Inc.
46 Worthington Drive
Maryland Heights, MD 63043

A. C. Eulberg
General Atomic
P.O. Box 81608
San Diego, CA 92138

E. H. Miller
General Electric Co.
Large Steam Turbine Division
300 Nott Street
Schenectady, NY 12301

R. K. Shah
Harrison Radiator Division
General Motors Corp.
Lockport, NY 14094

J. F. Sebal
Gilbert Associates, Inc.
525 Lancaster Avenue
Reading, PA 19603

J. E. Taboric
Heat Transfer Research Inc.
1000 S. Fremont Avenue
Alhambra, CA 91802

No. of
Copies

E. C. Smith
Hudson Products
6855 Horwin Drive
Houston, TX 77036

M. W. Larinoff
Hudson Products
6855 Horwin Drive
Houston, TX 77036

C. Rocco
Italimpianti - Societa
Italiana Impianti p.a.
Piazza, Piccapietra 9
18121 Genoa, ITALY

J. L. Mulloy
Los Angeles Department of
Water and Power
111 N. Hope Street
Los Angeles, CA 90012

R. J. Meyer
Louisiana Power & Light Co.
142 Delaronde Street
New Orleans, LA 70174

R. Landon
Marley Co.
5800 Fox Ridge Drive
Mission, KS 66202

L. R. Glicksman
Massachusetts Institute of
Technology
77 Massachusetts Avenue
Cambridge, MA 02139

R. Hofacher
Montana Power Co.
40 E. Broadway
Butte, MT 59701

W. G. Hoydysn
New York University
University Heights
New York, NY 10453

No. of
Copies

W. Kals
Niagara Blower
405 Lexington Avenue
New York, NY 10017

R. H. Meyer
Northeast Utilities
P.O. Box 270
Hartford, CT 06101

L. P. Davis
Department of Mechanical
Engineering
Oregon State University
Corvallis, OR 97330

C. E. Wicks
Department of Chemical
Engineering
Oregon State University
Corvallis, OR 97330

J. W. Michel
OTEC Heat Exchange Project
Activity
Oak Ridge National Laboratory
Box Y
Oak Ridge, TN 37830

F. F. Mautz
Pacific Gas & Electric
77 Beale Street
San Francisco, CA 94106

P. G. Humphries
Pacific Power & Light Co.
Public Service Bldg.
Portland, OR 97204

T. Rozenmann
PFR Engineering Systems, Inc.
Suite 832
4676 Admiralty Way
Marina del Rey, CA 90291

Prof. Ing. Carlo Roma
Piazza delle Muse 8
Rome, ITALY

No. of
Copies

A. C. Smith
Power Generation Cooling
Systems
4714 - 52nd Street S.
Seattle, WA 98118

R. F. Walker
Public Service of Colorado
5900 E. 39th Avenue
Denver, CO 80207

E. D. Kist
Public Service of New Mexico
Corporate Planning Dept.
Albuquerque, NM 87103

G. E. Collins
Research Cottrell
Hamon Cooling Tower Division
Box 750
Bound Brook, NJ 08805

R. Lindberg
Reynolds Aluminum Co.
Reynolds Metallurgical
Research Laboratory
Richmond, VA 23261

R. G. Lacy
San Diego Gas & Electric
101 Ash Street
San Diego, CA 92107

R. L. Skone
Seattle City Light
1015 Third Ave.
Seattle, WA 98104

J. Rasband
Southern California Edison
2244 Walnut Grove Avenue
Rosemead, CA 91770

A.L. London
Department of Mechanical
Engineering
Stanford University
Stanford, CA 94305

No. of
Copies

J. Y. Parce
Stearns and Rogers
700 S. Ash Street
Denver, CO 80222

D. H. Guild
Stone & Webster Engineering
Corp.
225 Franklin Street
Boston, MA 02107

H. I. Wilson
Tampa Electric Co.
P.O. Box 111
Tampa, FL 33601

W. Keel
Texas Electric Service Co.
115 W. Seventh Street
Fort Worth, TX 76102

G. J. Kidd
Union Carbide Corp.
ORGDP - K-25
P.O. Box D
Oak Ridge, TN 37803

G. A. Englessen
United Engineers &
Constructors
1401 Arch Street
Philadelphia, PA 19105

K. Simhan
University of Bremen
Bremen, WEST GERMANY

J. F. Kennedy
University of Iowa
Hydraulic Research Institute
Iowa City, IA 52240

J. H. Hutchinson
Utah Power & Light
1407 W. N. Temple
Salt Lake City, UT 84103

No. of
Copies

S. Ragone
Virginia Electric & Power Co.
700 E. Franklin Street
Richmond, VA 23261

G. H. Soule
West Associates
Sierra Pacific Power Co.
100 E. Moana Lane
Reno, NV 89510

J. A. Barnett
Executive Director
Western States Water Council
Room 1725
University Club Building
Salt Lake City, UT 84111

G. J. Silvestri
Westinghouse Electric Corp.
Steam Turbines Division
Lester Branch
Box 9175
Philadelphia, PA 19113

K. A. Oleson
Westinghouse Electric Corp.
Steam Turbines Division
Lester Branch
Box 9175
Philadelphia, PA 19113

ONSITE

2 ERDA Richland Operations
Office, Program Division

B. J. Melton
W. A. Burns

Westinghouse Hanford Company
Hanford Engineering Develop-
ment Laboratory

J. Fletcher

67 Battelle-Northwest

R. T. Alleman
P. A. Ard
J. W. Currie
D. E. Deonigi
J. G. DeSteeze
R. L. Dillon
D. W. Dragnich
R. L. Drake
D. W. Faletti
J. W. Finnigan
B. C. Fryer
C. H. Henager
A. D. Johnson
W. S. Kelly
R. S. Kemper
C. J. Knoll
W. V. Loscutoff
R. P. Marshall
D. E. Olesen
Y. Onishi
L. T. Pedersen
D. R. Pratt
H. C. Riches
J. B. Schuette
G. C. Smith
J. S. Stoakes
A. M. Sutey
D. S. Trent
M. Vagins
R. A. Walter
R. L. Watts
R. D. Widrig
L. E. Wiles
File - B. M. Johnson (30)
Technical Information (3)
Technical Publications